MECHANICAL
ENGINEER'S
POCKET-BOOK

KENT

1898

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David F. Castill

THE

ECHANICAL ENGINEER'S POCKET-BOOK.

JOHN WILEY & SONS.
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1916 1526p

The Publishers and the Author will be grateful to any of the readers of this volume who will kindly call their attention to any errors of omission or of commission that they may find therein. It is intended to make our publications standard works of study and reference, and, to that end, the greatest accuracy is sought. It rarely happens that the early editions of works of any size are free from errors; but it is the endeavor of the Publishers to see them removed immediately upon being discovered, and it is therefore desired that the Author may be aided in his task of revision, from time to time, by the kindly criticism of his readers.

JOHN WILEY & SONS.

53 East Tenth Street.

Thos. 4.

THE

MECHANICAL ENGINEER'S POCKET-BOOK.

A REFERENCE-BOOK OF RULES, TABLES, DATA, AND FORMULÆ, FOR THE USE OF ENGINEERS, MECHANICS, AND STUDENTS.

BY

WILLIAM KENT, A.M., M.E.,

Consulting Engineer,

Member Amer. Soc'y Mechl. Engrs. and Amer Inst. Mining Engrs.

THIRD EDITION, REVISED.

THIRD THOUSAND.

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PREFACE.

MORE than twenty years ago the author began to follow the advice given by Nystrom: "Every engineeer should make his own pocket-book, as he proceeds in study and practice, to suit his particular business." The manuscript pocket-book thus begun, however, soon gave place to more modern means for disposing of the accumulation of engineering facts and figures, viz., the index rerum, the scrap-book, the collection of indexed envelopes, portfolios and boxes, the card catalogue, etc. Four years ago, at the request of the publishers, the labor was begun of selecting from this accumulated mass such matter as pertained to mechanical engineering, and of condensing, digesting, and arranging it in form for publication. In addition to this, a careful examination was made of the transactions of engineering societies, and of the most important recent works on mechanical engineering, in order to fill gaps that might be left in the original collection, and insure that no important facts had been overlooked.

Some ideas have been kept in mind during the preparation of the Pocket-book that will, it is believed, cause it to differ from other works of its class. In the first place it was considered that the field of mechanical engineering was so great, and the literature of the subject so vast, that as little space as possible should be given to subjects which especially belong to civil engineering. While the mechanical engineer must continually deal with problems which belong properly to civil engineering, this latter branch is so well covered by Trautwine's "Civil Engineer's Pocket-book" that any attempt to treat it exhaustively would not only fill no "long-felt want," but would occupy space which should be given to mechanical engineering.

Another idea prominently kept in view by the author has been that he would not assume the position of an "authority" in giving rules and formulæ for designing, but only that of compiler, giving not only the name of the originator of the rule, where it was known, but also the *volume and page from which it* was taken, so that its derivation may be traced when desired. When differen formulæ for the same problem have been found they hav been given in contrast, and in many cases examples hav been calculated by each to show the difference betwee them. In some cases these differences are quite remark able, as will be seen under Safety-valves and Crank pins. Occasionally the study of these differences has led to th author's devising a new formula, in which case the deriva tion of the formula is given.

Much attention has been paid to the abstracting of dat of experiments from recent periodical literature, and numer ous references to other data are given. In this respec the present work will be found to differ from other Pocke books.

The author desires to express his obligation to the ma persons who have assisted him in the preparation of t work, to manufacturers who have furnished their ca logues and given permission for the use of their tabl and to many engineers who have contributed original d and tables. The names of these persons are mentioned their proper places in the text, and in all cases it has be endeavored to give credit to whom credit is due. T thanks of the author are also due to the following gent men who have given assistance in revising manuscript proofs of the sections named: Prof. De Volson Woo mechanics and turbines; Mr. Frank Richards, compress air; Mr. Alfred R. Wolff, windmills; Mr. Alex. Humphreys, illuminating gas; Mr. Albert E. Mitche locomotives; Prof. James E. Denton, refrigerating-m chinery; Messrs. Joseph Wetzler and Thomas W. Varle electrical engineering; and Mr. Walter S. Dix, for valu able contributions on several subjects, and suggestions to their treatment.

WM. KENT.

PASSAIC, N. J., *April*, 1895.

THIRD EDITION, APRIL, 1897.

All the typographical and other errors discovered in th first and second editions have been corrected, a few alte tions have been made in the text, and the index has be revised and enlarged.

W. K.

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ND ABBREVIATIONS OF PERIODICALS TEXT-BOOKS FREQUENTLY REFERRED TO IS WORK.

American Machinist.
ch. Appleton's Cyclopædia of Mechanics, Vols. I and II.
A. Bulletin of the American Iron and Steel Association
phia).
city and Resistance of Materials.
D. D. K. Clark's Rules, Tables, and Data for Mechanical En-

D. K. Clark's Treatise on the Steam-engine.
ineering (London).
Engineering News.
Engineer (London).
Useful Information for Engineers.
gation Canals and Flow of Water.
. W. Journal of American Charcoal Iron Workers' Association.
Journal of the Franklin Institute.
tric Transmission of Energy.
plied Mechanics.
Strength of Materials.
chanism. Supplementary volume of Appleton's Cyclopædia of
ics.
C. E. Proceedings Institution of Civil Engineers (London).
M. E. Proceedings Institution of Mechanical Engineers (Lon-

hermodynamics.
s Engineers' Club of Philadelphia.
E. Rankine's The Steam Engine and other Prime Movers.
Machinery and Millwork.
T. D. Rankine's Rules, Tables, and Data.
U. S. Test Board.
U. S. Testing Machine at Watertown, Massachusetts.
hermodynamics.
anual of Marine Engineering.
mith, Jr.'s Hydraulics.
s Indicator.
Dynamo-electric Machinery.
Manual of the Steam Engine.
Materials of Engineering.
E. E. Transactions American Institute of Electrical Engineers.
M. E. Transactions American Institute of Mining Engineers.
C. E. Transactions American Society of Civil Engineers.
M. E. Transactions American Soc'ty of Mechanical Engineers
Civil Engineer's Pocket Book.
otive (Hartford, Connecticut).
ments of Machine Design.
Mechanics of Engineering.
istance of Materials.
rmodynamics.

MATHEMATICS.

·al and Algebraical Signs and Abbreviations.

›n).

traction).

us.
.us.

by.
× *b*.
.

.

+ *b*. 15-16 = $\frac{15}{16}$.

= $\frac{2}{1000}$.
›t.

.

is, : to (proportion).
, as 2 is to 4 so is 3 to 6.
ded by.
›f 2 to 4 = 2/4.

an.

rc or thermometer.
r feet.
r inches.
to distinguish letters, as
a'''.
a_c. read *a* sub 1, *a* sub *b*,

—— vincula, denoting
e numbers enclosed are
ıken together; as,
: = $\overline{4+3} \times 5 = 35$.
ıred, *a* cubed.
o the *n*th power.
= $\sqrt{a^3}$.
$^2 = \frac{1}{a^2}$.
ıe 9th power = 1,000,000,-

sine of *a*.
he arc whose sine is *a*.
$\frac{1}{\quad}$
in. *a*.
thm.
. *log.* = *hyperbolic loga-*

∠ angle.
∟ right angle.
⊥ perpendicular to.
sin., sine.
cos., cosine.
tang., or tan., tangent.
sec., secant.
versin., versed sine.
cot., cotangent.
cosec., cosecant.
covers., co-versed sine.

In Algebra, the first letters of the alphabet, *a*, *b*, *c*, *d*, etc., are generally used to denote known quantities, and the last letters, *w*, *x*, *y*, *z*, etc., unknown quantities.

Abbreviations and Symbols commonly used.

d, differential (in calculus).
$\int$, integral (in calculus).
$\int_b^a$, integral between limits *a* and *b*.
Δ, delta, difference.
Σ, sigma, sign of summation.
π, pi, ratio of circumference of circle to diameter = 3.14159.
g, acceleration due to gravity = 32.16 ft. per sec.

Abbreviations frequently used in this Book.

L., l., length in feet and inches.
B., b., breadth in feet and inches.
D., d., depth or diameter.
H., h., height, feet and inches.
T., t., thickness or temperature.
V., v., velocity.
F., force, or factor of safety.
f., coefficient of friction.
E., coefficient of elasticity.
R., r., radius.
W., w., weight.
P., p., pressure or load.
H.P., horse-power.
I.H.P., indicated horse-power.
B.H.P., brake horse-power.
h. p., high pressure.
i. p., intermediate pressure.
l. p., low pressure.
A.W.G., American Wire Gauge (Brown & Sharpe).
B.W.G., Birmingham Wire Gauge.
r. p. m., or revs. per min., revolutions per minute.

A. RAMONEDA
INGENIERO
MAR 4 1903
MONTEREY, N. L., MEX.

denominators except its own for the new numerators, and all
tors together for the common denominator:

$$\frac{1}{2}, \frac{1}{3}, \frac{3}{7} = \frac{21}{42}, \frac{14}{42}, \frac{18}{42}.$$

fractions.—Reduce them to a common denominator, then
erators and place their sum over the common denominator:

$$\frac{1}{2} + \frac{1}{3} + \frac{3}{7} = \frac{21 + 14 + 18}{42} = \frac{53}{42} = 1\tfrac{11}{42}.$$

ract fractions.—Reduce them to a common denominator,
numerators and place the difference over the common denomi-

$$\frac{1}{2} - \frac{3}{7} = \frac{7 - 6}{14} = \frac{1}{14}.$$

DECIMALS.

decimals.—Set down the figures so that the decimal points
ve the other, then proceed as in simple addition: 18.75 + .012 =

tract decimals.—Set down the figures so that the decimal
one above the other, then proceed as in simple subtraction: 18.75
738.

ltiply decimals.—Multiply as in multiplication of whole
hen point off as many decimal places as there are in multiplier
licand taken together: $1.5 \times .02 = .030 = .03$.

ide decimals.—Divide as in whole numbers, and point off in
t as many decimal places as those in the dividend exceed those
or. Ciphers must be added to the dividend to make its decimal
ast equal those in the divisor, and as many more as it is desired
the quotient: $1.5 \div .25 = 6$. $0.1 \div 0.3 = 0.10000 \div 0.3 = 0.3333 +$

mal Equivalents of Fractions of One Inch.

015625	17-64	.265625	33-64	.515625	49-64	.765625
03125	9-32	.28125	17-32	.53125	25-32	.78125
046875	19-64	.296875	35-64	.546875	51-64	.796875
0625	**5-16**	.3125	**9-16**	.5625	**13-16**	.8125
078125	21-64	.328125	37-64	.578125	53-64	.828125
09375	11-32	.34375	19-32	.59375	27-32	.84375
109375	23-64	.359375	39-64	.609375	55-64	.859375
.125	**3-8**	.375	**5-8**	625	**7-8**	.875
.140625	25-64	.390625	41-64	.640625	57-64	.890625
.15625	13-32	.40625	21-32	.65625	29-32	.90625
.171875	27-64	.421875	43-64	.671875	59-64	.921875
.1875	**7-16**	.4375	**11-16**	.6875	**15-16**	.9375
.203125	29-64	.453125	45-64	.703125	61-64	.953125
.21875	15-32	.46875	23-32	.71875	31-32	.96875
.234375	31-64	.484375	47-64	.734375	63-64	.984375
.25	**1-2**	.50	**3-4**	.75	1	1.

nvert a common fraction into a decimal.—Divide the
r by the denominator, adding to the numerator as many ciphers
y a decimal point as are necessary to give the number of decimal
ired in the result: $\frac{1}{3} = 1.0000 \div 3 = 0.3333 +$.

nvert a decimal into a common fraction.—Set down
al as a numerator, and place as the denominator 1 with as many
nexed as there are decimal places in the numerator; erase the

ARITHMETIC.

The user of this book is supposed to have had a training in arithm well as in elementary algebra. Only those rules are given here whi apt to be easily forgotten.

GREATEST COMMON MEASURE, OR GREATE COMMON DIVISOR OF TWO NUMBERS.

Rule.—Divide the greater number by the less; then divide the by the remainder, and so on, dividing always the last divisor by t remainder, until there is no remainder, and the last divisor is the g common measure required.

LEAST COMMON MULTIPLE OF TWO OR MO NUMBERS.

Rule.—Divide the given numbers by any number that will divi greatest number of them without a remainder, and set the quotient the undivided numbers in a line beneath.

Divide the second line as before, and so on, until there are no two nu that can be divided; then the continued product of the divisors an quotients will give the multiple required.

FRACTIONS.

To reduce a common fraction to its lowest terms.— both terms by their greatest common divisor: $\frac{18}{24} = \frac{3}{4}$

To change an improper fraction to a mixed numb Divide the numerator by the denominator; the quotient is the whole nu and the remainder placed over the denominator is the fraction: $\frac{29}{4} =$

To change a mixed number to an improper fracti Multiply the whole number by the denominator of the fraction; to the uct add the numerator; place the sum over the denominator: $1\frac{7}{8} = \frac{15}{8}$

To express a whole number in the form of a fra with a given denominator.—Multiply the whole number given denominator, and place the product over that denominator: 8

To reduce a compound to a simple fraction, al multiply fractions.—Multiply the numerators together for numerator and the denominators together for a new denominator:

$$\frac{2}{3} \text{ of } \frac{4}{3} = \frac{8}{9}, \quad \text{also} \quad \frac{2}{3} \times \frac{4}{3} = \frac{8}{9}.$$

To reduce a complex to a simple fraction.—The num and denominator must each first be given the form of a simple fr then multiply the numerator of the upper fraction by the denomin the lower for the new numerator, and the denominator of the upper numerator of the lower for the new denominator:

$$\frac{\frac{2}{3}}{1\frac{1}{3}} = \frac{\frac{2}{3}}{\frac{4}{3}} = \frac{6}{12} = \frac{1}{2}.$$

To divide fractions.—Reduce both to the form of simple fra invert the divisor, and proceed as in multiplication:

$$\frac{2}{3} \div 1\frac{1}{3} = \frac{2}{3} \div \frac{4}{3} = \frac{2}{3} \times \frac{3}{4} = \frac{6}{12}.$$

Cancellation of fractions.—In compound or multiplied fra divide any numerator and any denominator by any number whi divide them both without remainder, striking out the numbers thus and setting down the quotients in their stead.

—duce ns to a common denominator.— the simple fraction; then multiply each decimal point in the numerator, and reduce the fraction thus formed to its lowest terms:

$$.25 = \frac{25}{100} = \frac{1}{4}; \quad .3333 = \frac{3333}{10000} = \frac{1}{3}, \text{ nearly.}$$

To reduce a recurring decimal to a common fraction.—Subtract the decimal figures that do not recur from the whole decimal including one set of recurring figures; set down the remainder as the numerator of the fraction, and as many nines as there are recurring figures, followed by as many ciphers as there are non-recurring figures, in the denominator. Thus:

$$\begin{array}{ll} & .79054054, \text{ the recurring figures being 054.} \\ \text{Subtract} & \underline{79} \\ & \frac{78975}{99900} = \text{(reduced to its lowest terms) } \frac{117}{148}. \end{array}$$

COMPOUND OR DENOMINATE NUMBERS.

Reduction descending.—To reduce a compound number to a lower denomination. Multiply the number by as many units of the lower denomination as makes one of the higher.

3 yards to inches: $3 \times 36 = 108$ inches.

.04 square feet to square inches: $.04 \times 144 = 5.76$ sq. in.

If the given number is in more than one denomination proceed in steps from the highest denomination to the next lower, and so on to the lowest, adding in the units of each denomination as the operation proceeds.

3 yds. 1 ft. 7 in. to inches: $3 \times 3 = 9$, $+ 1 = 10$, $10 \times 12 = 120$, $+ 7 = 127$ in.

Reduction ascending.—To express a number of a lower denomination in terms of a higher, divide the number by the number of units of the lower denomination contained in one of the next higher; the quotient is in the higher denomination, and the remainder, if any, in the lower.

127 inches to higher denomination.

$127 \div 12 = 10$ feet $+ 7$ inches; 10 feet $\div 3 = 3$ yards $+ 1$ foot.
Ans. 3 yds. 1 ft. 7 in.

To express the result in decimals of the higher denomination, divide the given number by the number of units of the given denomination contained in one of the required denomination, carrying the result to as many places of decimals as may be desired.

127 inches to yards: $127 \div 36 = 3\frac{19}{36} = 3.5277 +$ yards.

RATIO AND PROPORTION.

Ratio is the relation of one number to another, as obtained by dividing one by the other.

Ratio of 2 to 4, or $2 : 4 = 2/4 = 1/2$.
Ratio of 4 to 2, or $4 : 2 = 2$.

Proportion is the equality of two ratios. Ratio of 2 to 4 equals ratio of 3 to 6, $2/4 = 3/6$: expressed thus, $2 : 4 :: 3 : 6$; read, 2 is to 4 as 3 is to 6.

The first and fourth terms are called the extremes or outer terms, the second and third the means or inner terms.

The product of the means equals the product of the extremes:

$$2 : 4 :: 3 : 6; \quad 2 \times 6 = 12; \quad 3 \times 4 = 12.$$

Hence, given the first three terms to find the fourth, multiply the second and third terms together and divide by the first.

2 : 4 :: 3 : what number? Ans. $\frac{4 \times 3}{2} = 6$.

Product of Fractions Expressed in Decimals.

	1	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1
	.0625	.0039															
	.1250	.0078	.0156														
	.1875	.0117	.0234	.0352													
	.2500	.0156	.0313	0469	.0625												
$\frac{5}{16}$	.3125	.0195	.0391	.0586	.0781	.0977											
$\frac{3}{8}$	.3750	.0234	.0469	.0703	.0937	.1172	.1406										
$\frac{7}{16}$	.4375	.0273	.0547	.0820	.1093	.1367	.1641	.1914									
$\frac{1}{2}$	.5000	.0313	.0625	.0938	.1250	.1562	.1875	.2188	.2500								
$\frac{9}{16}$	.5625	.0352	.0703	.1055	.1406	.1758	.2109	.2461	.2813	.3164							
$\frac{5}{8}$	.6250	.0391	.0781	.1172	.1562	.1953	.2344	.2734	.3125	.3516	.3906						
$\frac{11}{16}$	.6875	.0430	.0859	.1289	.1719	.2148	.2578	.3008	.3438	.3867	.4297	.4727					
$\frac{3}{4}$	.7500	.0469	.0938	.1406	.1875	.2344	.2813	.3281	.3750	.4219	.4688	.5156	.5625				

cimal point in the numerator, and reduce the fraction thus formed to its west terms:

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dividing the index of the power by the index of the root, indicating the division by a fraction. Thus, extract the square root of the 6th power of 2:

$$\sqrt{2^6} = 2^{\frac{6}{2}} = 2^{\frac{3}{1}} = 2^3 = 8.$$

The 6th power of 2, as in the table above, is 64; $\sqrt{64} = 8$.

Difficult problems in evolution are performed by logarithms, but the square root and the cube root may be extracted directly according to the rules given below. The 4th root is the square root of the square root. The 6th root is the cube root of the square root, or the square root of the cube root; the 9th root is the cube root of the cube root; etc.

To Extract the Square Root.—Point off the given number into periods of two places each, beginning with units. If there are decimals, point these off likewise, beginning at the decimal point, and supplying as many ciphers as may be needed. Find the greatest number whose square is less than the first left-hand period, and place it as the first figure in the quotient. Subtract its square from the left-hand period, and to the remainder annex the two figures of the second period for a dividend. Double the first figure of the quotient for a partial divisor; find how many times the latter is contained in the dividend exclusive of the right-hand figure, and set the figure representing that number of times as the second figure in the quotient, and annex it to the right of the partial divisor, forming the complete divisor. Multiply this divisor by the second figure in the quotient and subtract the product from the dividend. To the remainder bring down the next period and proceed as before, in each case doubling the figures in the root already found to obtain the trial divisor. Should the product of the second figure in the root by the completed divisor be greater than the dividend, erase the second figure both from the quotient and from the divisor, and substitute the next smaller figure, or one small enough to make the product of the second figure by the divisor less than or equal to the dividend.

$$\begin{array}{r|l}
 & \dot{3}.\dot{1}4\dot{1}5\dot{9}2\dot{6}5\dot{3}6 \,|\, 1.77245+ \\
 & 1 \\
27 & 214 \\
 & 189 \\
347 & 2515 \\
 & 2429 \\
3542 & 8692 \\
 & 7084 \\
35444 & 160865 \\
 & 141776 \\
354485 & 1908936 \\
 & 1772425
\end{array}$$

To extract the square root of a fraction, extract the root of numerator and denominator separately. $\sqrt{\frac{4}{9}} = \frac{2}{3}$, or first convert the fraction into a decimal, $\sqrt{\frac{4}{9}} = \sqrt{.4444+} = .6666+$.

To Extract the Cube Root.—Point off the number into periods of 3 figures each, beginning at the right hand, or unit's place. Point off decimals in periods of 3 figures from the decimal point. Find the greatest cube that does not exceed the left-hand period; write its root as the first figure in the required root. Subtract the cube from the left-hand period, and to the remainder bring down the next period for a dividend.

Square the first figure of the root; multiply by 300, and divide the product into the dividend for a trial divisor; write the quotient after the first figure of the root as a trial second figure.

Complete the divisor by adding to 300 times the square of the first figure 30 times the product of the first by the second figure, and the square of t
econd figure. Multiply this divisor by the second figure; subtract
roduct from the remainder. (Should the product be greater than
mainder, the last figure of the root and the complete divisor are too la

ite for the last figure the next smaller number, and correct the trial iccordingly.)

; remainder bring down the next period, and proceed as before to third figure of the root—that is, square the two figures of the root found; multiply by 300 for a trial divisor, etc.

any time the trial divisor is less than the dividend, bring down an- eriod of 3 figures, and place 0 in the root and proceed.

ube root of a number will contain as many figures as there are of 3 in the number.

ter Methods of Extracting the Cube Root.—1. From **rth's Algebra:**

		$\dot{1},88\dot{1},36\dot{5},96\dot{3},62\dot{5}$ | 12345
		1
300×1^2 =	300	881
$30 \times 1 \times 2$ =	60	
2^2 =	4	
	364	728
	64	153365
300×12^2 =	43200	
$30 \times 12 \times 3$ =	1080	
3^2 =	9	
	44289	132867
	1089	20498963
300×123^2 =	4538700	
$30 \times 123 \times 4$ =	14760	
4^2 =	16	
	4553476	18213904
	14776	2285059625
300×1234^2 =	456826800	
$30 \times 1234 \times 5$ =	185100	
5^2 =	25	
	457011925	2285059625

· the first two figures of the root are found the next trial divisor is oy bringing down the sum of the 60 and 4 obtained in completing the ing divisor, then adding the three lines connected by the brace, and ng two ciphers. This method shortens the work in long examples, as in the case of the last two trial divisors, saving the labor of squaring l 1234. A further shortening of the work is made by obtaining the o figures of the root by division, the divisor employed being three he square of the part of the root already found; thus, after finding t three figures:

3×123^2 = 45387	20498963	45.1+
	181548	
	234416	
	226935	
	74813	

'or due to the remainder is not sufficient to change the fifth figure of t.

Prof. H. A. Wood (*Stevens Indicator*, July, 1890):

ving separated the number into periods of three figures each, count- n the right, divide by the square of the *nearest* root of the first or first two periods; the *nearest root is the trial* root.

) the quotient *obtained add twice the trial* root, and divide by 3. *es the root, or first approximation.*

using the first approximate root as a new trial root, and proceed- *fore, a nearer approximation is* obtained, which process may be *ntil the root has been extracted*, or the approximation carried as *ed.*

Example.—Required the cube root of 20. The nearest cube

$$3^2 = 9)20.0$$
$$2.2$$
$$6$$
$$3)8.1$$
$$2.7 \text{ 1st T. R.}$$

$$2.7^2 = 7.29)20.000$$
$$2.743$$
$$5.4$$
$$3)8.143$$
$$2.714, \text{ 1st ap. cube root.}$$

$$2.714^2 = 7.365796)20.0000000$$
$$2.7152534$$
$$5.428$$
$$3)8.1432534$$
$$2.7144178 \text{ 2d ap. cube root.}$$

Remark.—In the example it will be observed that the sec first two figures of the root, were obtained by using for trial ro the first period. Using, in like manner, these two terms for obtained four terms of the root; and these four terms for tr seven figures of the root correct. In that example the last figı 7. Should we take these eight figures for trial root we should c fifteen figures of the root correct.

To Extract a Higher Root than the Cube.—The the square root of the square root; the sixth root is the cub square root or the square root of the cube root. Other roots ε veniently found by the use of logarithms.

ALLIGATION

shows the value of a mixture of different ingredients when and value of each is known.

Let the ingredients be a, b, c, d, etc., and their respective va w, x, y, z, etc.

A = the sum of the quantities $= a + b + c + d$, etc

P = mean value or price per unit of A.

$$AP = aw + bx + cy + dz, \text{ etc.}$$

$$P = \frac{aw + bx + cy + dz}{A}.$$

PERMUTATION

shows in how many positions any number of things may be row; thus, the letters a, b, c may be arranged in six positions, *cab*, *cba*, *bac*, *bca*.

Rule.—Multiply together all the numbers used in counting the permutations of 1, 2, and $3 = 1 \times 2 \times 3 = 6$. In how many pc things in a row be placed?

$$1 \times 2 \times 3 \times 4 \times 5 \times 6 \times 7 \times 8 \times 9 = 362880.$$

COMBINATION

shows how many arrangements of a few things may be mε greater number. Rule: Set down that figure which indicate number, and after it a series of figures diminishing by 1, until set down as the number of the few things to be taken in each. Then beginning under the last one set down said number of then ... backward set down a series diminishing by 1 until a ... numbers. Multiply together all the uppe ... the lower numbers to form anothe ...ue.

How many combinations of 9 things can be made, taking 3 in ea bination ?

$$\frac{9 \times 8 \times 7}{1 \times 2 \times 3} = \frac{504}{6} = 84.$$

ARITHMETICAL PROGRESSION,

in a series of numbers, is a progressive increase or decrease in each sive number by the addition or subtraction of the same amount at ea as 1, 2, 3, 4, 5, etc., or 15, 12, 9, 6, etc. The numbers are called terms, equal increase or decrease the difference. Examples in arithmeti gression may be solved by the following formulæ:

Let a = first term, l = last term, d = common difference, n = nu terms, s = sum of the terms:

$$l = a + (n - 1)d, \qquad = -\frac{1}{2}d \pm \sqrt{2ds + \left(a - \frac{1}{2}\right.}$$

$$= \frac{2s}{n} - a, \qquad = \frac{s}{n} + \frac{(n-1)d}{2}.$$

$$s = \frac{1}{2}n[2a + (n - 1)d], \qquad = \frac{l + a}{2} + \frac{l^2 - a^2}{2d},$$

$$= (l + a)\frac{n}{2}, \qquad = \frac{1}{2}n[2l - (n - 1)d].$$

$$a = l - (n - 1)d, \qquad = \frac{s}{n} - \frac{(n-1)d}{2},$$

$$= \frac{1}{2}d \pm \sqrt{\left(l + \frac{1}{2}d\right)^2 - 2ds}, \qquad = \frac{2s}{n} - l.$$

$$d = \frac{l - a}{n - 1}, \qquad = \frac{2(s - an)}{n(n - 1)},$$

$$= \frac{l^2 - a^2}{2s - l - a}, \qquad = \frac{2(nl - s)}{n(n - 1)}.$$

$$n = \frac{l - a}{d} + 1, \qquad = \frac{d - 2a \pm \sqrt{(2a - d)^2 + 8c}}{2d}$$

$$= \frac{2s}{l + a}, \qquad = \frac{2l + d \pm \sqrt{(2l + d)^2 - 8ds}}{2d}$$

GEOMETRICAL PROGRESSION,

in a series of numbers, is a progressive increase or decrease in e cessive number by the same multiplier or divisor at each step, as 16, etc., or 243, 81, 27, 9, etc. The common multiplier is called the r

Let a = first term, l = last term, r = ratio or constant multipli number of terms, m = any term, as 1st, 2d, etc., s = sum of the term

$$l = ar^{n-1}, \qquad = \frac{a + (r - 1)s}{r}, \qquad = \frac{(r - 1)sr^{n-1}}{r^n - 1},$$

$$\log l = \log a + (n - 1)\log r, \qquad l(s - l)^{n-1} - a(s - a)^n$$

$$m = ar^{m-1}. \qquad \log m = \log a + (m - 1)\log r.$$

$$s = \frac{a(r^n - 1)}{r - 1}, \qquad = \frac{rl - a}{r - 1}, \qquad = \frac{\sqrt[n-1]{l^n} - \sqrt[n-1]{a^n}}{\sqrt[n-1]{l} - \sqrt[n-1]{a}}, \qquad =$$

$$a = \frac{l}{r^{n-1}}, \qquad = \frac{(r-1)s}{r^n - 1}. \qquad \log a = \log l - (n-1) \log$$

$$r = \sqrt[n-1]{\frac{l}{a}}, \qquad = \frac{s-a}{s-l}. \qquad \log r = \frac{\log l - \log}{n-1}$$

$$r^n - \frac{s}{a} r + \frac{s-a}{a} = 0. \qquad r^n - \frac{s}{s-l} r^{n-1} + \frac{l}{s-l} = 0.$$

$$n = \frac{\log l - \log a}{\log r} + 1, \qquad = \frac{\log [a + (r-1)s] - \log a}{\log r},$$

$$= \frac{\log l - \log a}{\log (s-a) - \log (s-l)} + 1, \qquad = \frac{\log l - \log [lr - (r-1)s]}{\log r} + 1$$

Population of the United States.

(A problem in geometrical progression.)

Year.	Population.	Increase in 10 Years, per cent.	Annual Incre per cent.
1860	31,443,321		
1870	39,818,449*	26.63	2.39
1880	50,155,783	25.96	2.33
1890	62,622,250	24.86	2.25
1895	Est. 69,733,000	——	Est. 2.174
1900	" 77,652,000	Est. 24.0	" 2.174

Estimated Population in Each Year from 1860 *to* 1899.

(Based on the above rates of increase, in even thousands.)

1860....	31,443	1870....	39,818	1880....	50,156	1890 ...	6
1861	32,195	1871....	40,748	1881....	51,281	1891....	6
1862 ...	32,964	1872....	41,699	1882 ...	52,433	1892...	6
1863....	33,752	1873. ..	42,673	1883....	53,610	1893....	6
1864....	34,558	1874...	43,670	1884 ...	54,813	1894. ..	6
1865 ..	35,384	1875....	44,690	1885....	56,043	1895....	6
1866 ...	36,229	1876....	45,373	1886 ...	57,301	1896 ...	7
1867....	37,095	1877 ..	46,800	1887....	58,588	1897....	7
1868 ...	37,981	1878 ...	47,893	1888....	59,903	1898 ...	7
1869 ..	38,889	1879....	49,011	1889 ...	61,247	1899....	7

The above table has been calculated by logarithms, as follows:

$\log r = \log l - \log a \div (n-1)$, $\qquad \log m = \log a + (m-1) \log r$

Pop. 1870 .. 39,818449 log = 7.6000841 = log l
" 1860....31,443321 log = 7.4975288 = log a

diff. = .1025558
$n = 11$, $n - 1 = 10$, diff. ÷ 10 = .01025553 = log r,
add log for 1860 7.4975288 = log a

log for 1861 = 7.50778433 No. = 32,195 . . .
add again .01025553

log for 1862 7.51803986 No. = 32,964 . . .

Compound interest is a form of geometrical progression; the r
being 1 plus the percentage.

*[illegible]0,078, estimated error of the census of 1870.

INTEREST AND DISCOUNT.

terest is money paid for the use of money for a given time; the fac are:

p, the sum loaned, or the principal;
t, the time in years;
r, the rate of interest;
i, the amount of interest for the given rate and time;
$a = p + i =$ the amount of the principal with interest at the end of the time.

ormulæ:

$$i = \text{interest} = \text{principal} \times \text{time} \times \text{rate per cent} = i = \frac{ptr}{100};$$

$$a = \text{amount} = \text{principal} + \text{interest} = p + \frac{ptr}{100};$$

$$r = \text{rate} = \frac{100i}{pt};$$

$$p = \text{principal} = \frac{100i}{tr} = a - \frac{ptr}{100};$$

$$t = \text{time} = \frac{100i}{pr}.$$

the rate is expressed decimally as a per cent,—thus, 6 per cent = .06,—formulæ become

$$i = prt; \ a = p(1 + rt); \quad r = \frac{i}{pt}; \quad t = \frac{i}{pr}; \quad p = \frac{i}{tr} = \frac{a}{1 + rt}.$$

ules for finding Interest.—Multiply the principal by the rate num divided by 100, and by the time in years and fractions of a year. the time is given in days, interest $= \dfrac{\text{principal} \times \text{rate} \times \text{no. of days}}{365 \times 100}$.

banks interest is sometimes calculated on the basis of 360 days to a , or 12 months of 30 days each.

ort rules for interest at 6 per cent, when 360 days are taken as 1 year:
ltiply the principal by number of days and divide by 6000.
ltiply the principal by number of months and divide by 200.
e interest of 1 dollar for one month is ½ cent.

terest of 100 Dollars for Different Times and Rates.

Time.	2%	3%	4%	5%	6%	8%	10%
r	\$2.00	\$3.00	\$4.00	\$5.00	\$6.00	\$8.00	\$10 00
nth	.16⅔	.25	.33⅓	.41⅔	.50	.66⅔	.83⅓
$= \frac{1}{360}$ year	.0055$\frac{5}{9}$	.0083⅓	.0111⅑	.0138$\frac{8}{9}$	.0166⅔	.0222$\frac{2}{9}$	.0277$\frac{7}{9}$
$= \frac{1}{365}$ year	.005479	.008219	.010959	.013699	.016438	.0219178	.0273973

scount is interest deducted for payment of money before it is due.
ue discount is the difference between the amount of a debt pay- at a future date without interest and its present worth. The present h is that sum which put at interest at the legal rate will amount to the when it is due.

find the present worth of an amount due at future date, divide the unt by the amount of \$1 placed at interest for the given time. The dis- t equals the amount minus the present worth.

at discount should be allowed on \$103 paid six months before it is due, est being 6 per cent per annum?

$$\frac{103}{1 + 1 \times .06 \times \frac{1}{2}} = \$100 \text{ present worth, discount} = 3.00.$$

nk discount is the amount deducted by a bank as interest on ey loaned on promissory notes. It is interest calculated not on the act- sum loaned, but on the gross amount of the note, from which the dis- t is deducted in advance. It is also calculated on the basis of 360 days e year, and for 3 (in some banks 4) days more than the time specified in note. These are called days of grace, and the note is not payable til ast of these days. In some States days of grace have been abolis

What discount will be deducted by a bank in discounting a payable 6 months hence? Six months = 182 days, add 3 day days' $\frac{103 \times 185}{6000} = \$3.176$.

Compound Interest.—In compound interest the interes the principal at the end of each year, (or shorter period if agre

Let p = the principal, r = the rate expressed decimally, n = and a the amount:

$$a = \text{amount} = p\,(1 + r)^n;\ r = \text{rate} = \sqrt[n]{\frac{a}{p}} - 1,$$

$$p = \text{principal}, = \frac{a}{(1 + r)^n},\ \text{no of years} = n, = \frac{\log a - \text{lo}}{\log (1 + }$$

Compound Interest Table.

(Value of one dollar at compound interest, compounded yearl 3, 4, 5, and 6 per cent, from 1 to 50 years.)

Years.	3%	4%	5%	6%	Years.	3%	4%	5%
1	1.03	1.04	1.05	1.06	16	1 6047	1 8730	2.1829
2	1.0609	1.0816	1.1025	1.1236	17	1.6528	1.9479	2.2920
3	1.0927	1.1249	1.1576	1.1910	18	1.7024	2.0258	2.4066
4	1.1255	1.1699	1.2155	1.2625	19	1 7535	2.1068	2 5269
5	1.1593	1.2166	1.2763	1.3382	20	1.8061	2.1911	2.6533
6	1.1941	1.2653	1.3401	1.4185	21	1.8603	2.2787	2.7859
7	1.2299	1.3159	1.4071	1.5036	22	1.9161	2.3699	2 9252
8	1.2668	1.3686	1.4774	1.5938	23	1.9736	2 4647	3.0715
9	1.3048	1.4233	1.5513	1.6895	24	2.0328	2.5633	3.2251
10	1.3439	1.4802	1.6289	1.7908	25	2.0937	2.6658	3.3864
11	1.3842	1.5394	1.7103	1.8983	30	2.4272	3.2434	4.3219
12	1.4258	1.6010	1.7958	2.0122	35	2.8138	3.9460	5.5166
13	1.4685	1.6651	1 8856	2 1329	40	3.2620	4.8009	7 0100
14	1.5126	1.7317	1.9799	2.2609	45	3.7815	5.8410	8.9850
15	1.5580	1.8009	2.0789	2.3965	50	4.3838	7.1064	11.6792

At compound interest at 3 per cent money will double itself in 23½ y at 4 per cent in 17⅔ years, at 5 per cent in 14.2 years, and at 6 per ce 11.9 years.

EQUATION OF PAYMENTS.

By equation of payments we find the equivalent or average time in w ne payment should be made to cancel a number of obligations due a rent dates; also the number of days upon which to calculate inter scount upon a gross sum which is composed of several smaller sums le at different dates.

Rule.—Multiply each item by the time of its maturity in days fr d date, taken as a standard, and divide the sum of the products b i of the items: the result is the average time in days from the sta

owes B \$100 due in 30 days, \$200 due in 60 days, and \$300 due in 90 ow many days may the whole be paid in one sum of \$600?

$100 \times 30 + 200 \times 60 + 300 \times 90 = 42{,}000;\ \ 42{,}000 \div 600 = 70$ days, *ans.*

wes B \$100, \$200, and \$300, which amounts are overdue respective d 90 d- ... w pays the whole amount, \$600, how many ... hat sum? *Ans.* 70 days.

PARTIAL PAYMENTS.

ıpute interest on notes and bonds when partial payments have been

ıd **States Rule.**—Find the amount of the principal to the time st payment, and, subtracting the payment from it, find the amount mainder as a new principal to the time of the next payment. payment is less than the interest, find the amount of the principal ne when the sum of the payments equals or exceeds the interest subtract the sum of the payments from this amount. d in this manner till the time of settlement.

—The principles upon which the preceding rule is founded are: ıat payments must be applied first to discharge accrued interest, the remainder, if any, toward the discharge of the principal. at only unpaid principal can draw interest.

antile Method.—When partial payments are made on short interest accounts, business men commonly employ the following

he amount of the whole debt to the time of settlement; also find ınt of each payment from the time it was made to the time of set- Subtract the amount of payments from the amount of the debt; ıinder will be the balance due.

ANNUITIES.

ınuity is a fixed sum of money paid yearly, or at other equal times ıpon. The values of annuities are calculated by the principles of ıd interest.

i denote interest on $1 for a year, then at the end of a year the will be $1+i$. At the end of n years it will be $(1+i)^n$.

sum which in n years will amount to 1 is $\frac{1}{(1+i)^n}$ or $(1+i)^{-n}$, or the value of 1 due in n years.

amount of an annuity of 1 in any number of years n is $\frac{(1+i)^n-1}{i}$.

present value of an annuity of 1 for any number of years n is $i)^{-n}$ ——.

annuity which 1 will purchase for any number of years n is —— $i)^{-n}$.

annuity which would amount to 1 in n years is $\frac{i}{(1+i)^n-1}$.

nounts, Present Values, etc., at 5% Interest.

(1) $(1+i)^n$	(2) $(1+i)^{-n}$	(3) $\frac{(1+i)^n-1}{i}$	(4) $\frac{1-(1+i)^{-n}}{i}$	(5) $\frac{i}{1-(1+i)^{-n}}$	(6) $\frac{i}{(1+i)^n-1}$
1.05	.952381	1.	.952381	1.05	1.
1.1025	.907029	2.05	1.859410	.537805	.487805
1.157625	.863838	3.1525	2.723248	.367209	.317209
1.215506	.822702	4.310125	3.545951	.282012	.232012
1.276282	.783526	5.525631	4.329477	.230975	.180975
1.340096	.746215	6.801913	5.075692	.197017	.147018
1.407100	.710681	8.142008	5.786373	.172820	.122820
.477455	.676839	9.549109	6.463213	.154722	.104722
551328	.644609	11.026564	7.107822	.140690	.090690
28895	.613913	12.577893	7.721735	.129505	.079505

Table I.—Annuity Required to Redeem $1000 in from 1 to 50 Years.

Years to run.	Rate of Interest, per cent.												
	2	2¼	2½	2¾	3	3¼	3½	3¾	4	4½	5	5½	6
2	495.05	494.50	493.78	493.23	492.69	492.05	491.42	490.81	490.20	489.00	487.80	486.62	485.43
3	326.72	325.94	325.14	324.35	323.56	322.75	321.94	321.13	320.36	318.77	317.21	315.63	314.10
4	242.63	241.74	240.84	239.93	239.02	238.14	237.26	236.38	235.50	233.74	232.01	230.29	228.60
5	192.16	191.18	190.24	189.30	188.35	187.42	186.49	185.56	184.63	182.79	180.98	179.18	177.39
6	158.53	157.53	156.56	155.58	154.61	153.64	152.67	151.73	150.79	148.88	147.02	145.18	143.36
7	134.52	133.51	132.49	131.50	130.51	129.54	128.57	127.59	126.61	124.67	122.82	120.96	119.18
8	116.51	115.48	114.47	113.46	112.46	111.47	110.48	109.50	108.53	106.60	104.72	102.86	101.03
9	102.52	101.48	100.46	99.45	98.44	97.44	96.44	95.46	94.49	92.57	90.69	88.83	87.02
10	91.33	90.29	89.25	88.24	87.24	86.24	85.24	84.26	83.29	81.38	79.50	77.67	75.87
11	82.18	81.14	80.11	79.09	78.07	77.08	76.09	75.12	74.15	72.25	70.39	68.57	66.79
12	74.56	73.52	72.49	71.47	70.46	69.47	68.48	67.51	66.55	64.67	62.83	61.08	59.28
13	68.12	67.08	66.05	65.04	64.03	63.05	62.06	61.10	60.14	58.27	56.45	54.68	52.96
14	62.60	61.56	60.54	59.53	58.53	57.55	56.57	55.62	54.67	52.82	51.02	49.28	47.58
15	57.83	56.79	55.77	54.77	53.77	52.79	51.82	50.88	49.94	48.11	46.34	44.62	42.96
16	53.65	52.62	51.60	50.60	49.61	48.64	47.68	46.70	45.82	44.01	42.27	40.58	38.95
17	49.97	48.94	47.93	46.94	45.95	44.99	44.04	43.12	42.20	40.42	38.70	37.04	35.44
18	46.70	45.67	44.67	43.69	42.71	41.76	40.82	39.90	38.99	37.24	35.54	33.92	32.36
19	43.78	42.76	41.76	40.78	39.81	38.87	37.94	37.04	36.14	34.40	32.75	31.15	29.62
20	41.15	40.14	39.14	38.18	37.22	36.29	35.36	34.47	33.58	31.87	30.24	28.68	27.18
25	31.22	30.24	29.27	28.35	27.43	26.55	25.67	24.84	24.01	22.44	20.95	19.55	18.23
30	24.65	23.70	22.78	21.90	21.02	20.19	19.37	18.60	17.83	16.39	15.05	13.80	12.65
35	20.00	19.09	18.20	17.37	16.54	15.77	15.00	14.29	13.58	12.27	11.07	9.97	8.97
40	16.55	15.68	14.84	14.05	13.26	12.54	11.88	11.17	10.52	9.34	8.28	7.32	6.46
45	13.91	13.07	12.27	11.52	10.78	10.12	9.45	8.85	8.26	7.20	6.26	5.43	4.70
50	11.82	11.02	10.26	9.56	8.87	8.25	7.68	7.09	6.55	5.60	4.78	4.06	3.44

TABLES FOR CALCULATING SINKING-FUNDS AND PRESENT VALUES.

Engineers and others connected with municipal work and industrial enterprises often find it necessary to calculate payments to sinking-funds which will provide a sum of money sufficient to pay off a bond issue or other debt at the end of a given period, or to determine the present value of certain annual charges. The accompanying tables were computed by Mr. John W. Hill, of Cincinnati, *Eng'g News*, Jan. 25, 1894.

Table I (opposite page) shows the annual sum at various rates of interest required to net $1000 in from 2 to 50 years, and Tab'e II shows the present value at various rates of interest of an annual charge of $1000 for from 5 to 50 years. at five-year intervals and for 100 years.

Table II.—Capitalization of Annuity of $1000 for from 5 to 100 Years.

Years.	Rate of Interest, per cent.							
	2½	3	3½	4	4½	5	5½	6
5	4,645 88	4,579.60	4.514.92	4,451.68	4,389.91	4,329.45	4,268.09	4,212.40
10	8,752.17	8,530.13	8.316.45	8,110.74	7,912.67	7,721.73	7,537.54	7,360.19
15	12,381.41	11,937.80	11,517.23	11,118.06	10,739.42	10,379.53	10,037.48	9.712.30
20	15.589.215	14,877.27	14,212.12	13,590.21	13,007.88	12,462.13	11,950.26	11,469.96
25	18,424.67	17,413.01	16,481.28	15,621.93	14,828.12	14,093.86	13,413.82	12,783.38
30	20,930.59	19,600.21	18,391.85	17,291.86	16,288.77	15,372.36	14,533.63	13.764.85
35	23,145.31	21,487.04	20,000.43	18,664.37	17,460.89	16,374.36	15,390.48	14.488.65
40	25,103.53	23,114.36	21,354.83	19,792.65	18,401.49	17,159.01	16.044.92	15,046.31
45	26,833.15	24,518.49	22.495.23	20,719.89	19,156.24	17,773.99	16,547.65	15.455.85
50	28,362.48	25.729.58	23,455.21	21,482.08	19,761 93	18.255.86	16,931.97	15,761.87
100	36,614.21	31,598.81	27,655.36	24,504.96	21,949.21	19,847.90	18,095.83	16,612.64

WEIGHTS AND MEASURES.

Long Measure.—Measures of Length.

12 inches	= 1 foot.
3 feet	= 1 yard.
5½ yards, or 16½ feet	= 1 rod, pole, or perch.
40 poles, or 220 yards	= 1 furlong.
8 furlongs, or 1760 yards, or 5280 feet	= 1 mile.
3 miles	= league.

Additional measures of length in occasional use: 1000 mils = 1 inch: 4 inches = 1 hand; 9 inches = 1 span; 2½ feet = 1 military pace; 2 yards = 1 fathom.

Old Land Measure.—7.92 inches = 1 link; 100 links, or 66 feet, or 4 poles = 1 chain; 10 chains = 1 furlong; 8 furlongs = 1 mile; 10 square chains = 1 acre.

Nautical Measure.

6080.26 feet, or 1.15156 statute miles	= 1 nautical mile, or knot.*
3 nautical miles	= 1 league.
60 nautical miles, or 69.168 statute miles	= 1 degree (at the equator).
360 degrees	= circumference of the earth at the equator.

**The British Admiralty takes the round figure of 6080 ft. which is the length of the "measured mile" used in trials of vessels.* The value varies *from 6080.26 to 6088.44 ft. according to different measures of the earth's diameter. There is a difference of opinion among writers as to the use of the word "knot" to mean length or a distance*—some holding that it should

Square Measure.—Measures of Surface.

144 square inches, or 183.35 circular inches	= 1 square foot.
9 square feet	= 1 square yard.
30¼ square yards, or 272¼ square feet	= 1 square rod, pole, or perc[h]
40 square poles	= 1 rood.
4 roods, or 10 sq. chains, or 160 sq. poles, or 4840 sq. yards, or 43560 sq. feet,	= 1 acre.
640 acres	= 1 square mile.

An acre equals a square whose side is 208.71 feet.
A circular inch is the area of a circle 1 inch in diameter = 0.7854 squ[are] inch.
1 square inch = 1.2732 circular inches.
A circular mil is the area of a circle 1 mil, or .001 inch in diame[ter]. 1000^2 or 1,000,000 circular mils = 1 circular inch.
1 square inch = 1,273,239 circular mils.
The mil, and circular mil are used in electrical calculations invol[ving] the diameter and area of wires.

Solid or Cubic Measure.—Measures of Volume.

1728 cubic inches = 1 cubic foot.
27 cubic feet = 1 cubic yard.
1 cord of wood = a pile, 4 × 4 × 8 feet = 128 cubic feet.
1 perch of masonry = 16½ × 1½ × 1 foot = 24¾ cubic feet.

Liquid Measure.

4 gills	= 1 pint.
2 pints	= 1 quart.
4 quarts	= 1 gallon { U. S. 231 cubic inches. Eng. 277.274 cubic inches.
31½ gallons	= 1 barrel.
42 gallons	= 1 tierce.
2 barrels, or 63 gallons	= 1 hogshead.
84 gallons, or 2 tierces	= 1 puncheon.
2 hogsheads, or 126 gallons	= 1 pipe or butt.
2 pipes, or 3 puncheons	= 1 tun.

The U. S. gallon contains 231 cubic inches; 7.4805 gallons = 1 cubic f[oot]. A cylinder 7 in. diam. and 6 in. high contains 1 gallon, very nearly, or [2]... cubic inches. The British Imperial gallon contains 277.274 cubic in[ches] = 1.20032 U. S. gallon.

The Miner's Inch.—(Western U. S. for measuring flow of a str[eam] of water).
The term Miner's Inch is more or less indefinite, for the reason that C[ali]fornia water companies do not all use the same head above the centr[e of] the aperture, and the inch varies from 1.36 to 1.78 cubic feet per mi[nute] each; but the most common measurement is through an aperture 2 in[ches] high and whatever length is required, and through a plank 1¼ inches th[ick]. The lower edge of the aperture should be 2 inches above the bottom of [the] measuring-box, and the plank 5 inches high above the aperture, thus m[ak]ing a 6-inch head above the centre of the stream. Each square inch of [the] opening represents a miner's inch, which is equal to a flow of 1½ cubic [feet] per minute.

Apothecaries' Fluid Measure.

60 minims = 1 fluid drachm.
8 drachms, or 437½ grains, or 1.732 cubic inches = 1 fluid ounce.

Dry Measure, U. S.

2 pints = 1 quart.
8 quarts = 1 peck.
4 pecks = 1 bushel.

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used only to denote a rate of speed. The length between knots on the line is $\frac{1}{120}$ of a n... ...le or 50.7 ft. when a half-minute glass is used that a speed c... ...d to 10 nautical miles per hour.

The standard U. S. bushel is the Winchester bushel, which is in cylinder form, 18½ inches diameter and 8 inches deep, and contains 2150.42 cubic inches.

A struck bushel contains 2150.42 cubic inches = 1.2445 cu. ft.: 1 cubic foot = 0.80356 struck bushel. A heaped bushel is a cylinder 18½ inches diameter and 8 inches deep, with a heaped cone not less than 6 inches high. It is equal to 1¼ struck bushels.

The British Imperial bushel is based on the Imperial gallon, and contains 8 such gallons, or 2218.192 cubic inches = 1.2837 cubic feet. The English quarter = 8 Imperial bushels.

Capacity of a cylinder in U. S. gallons = square of diameter, in inches × height in inches × .0034. (Accurate within 1 part in 100,000.)

Capacity of a cylinder in U. S. bushels = square of diameter in inches × height in inches × .0003652.

Shipping Measure.

Register Ton.—For register tonnage or for measurement of the entire internal capacity of a vessel:

100 cubic feet = 1 register ton.

This number is arbitrarily assumed to facilitate computation.

Shipping Ton.—For the measurement of cargo:

40 cubic feet =	1 U. S. shipping ton. 31.16 Imp. bushels. 32.143 U. S. "
42 cubic feet =	1 British shipping ton. 32.719 Imp. bushels. 33.75 U. S. "

Carpenter's Rule.—Weight a vessel will carry = length of keel × breadth at main beam × depth of hold in feet ÷ 95 (the cubic feet allowed for a ton). The result will be the tonnage. For a double-decker instead of the depth of the hold take half the breadth of the beam.

Measures of Weight.—Avoirdupois, or Commercial Weight.

16 drachms, or 437.5 grains	= 1 ounce, oz.
16 ounces, or 7000 grains	= 1 pound, lb.
28 pounds	= 1 quarter, qr.
4 quarters	= 1 hundredweight, cwt = 112 lbs.
20 hundred weight	= 1 ton of 2240 pounds, or long ton.
2000 pounds	= 1 net, or short ton.
2204.6 pounds	= 1 metric ton.

1 stone = 14 pounds ; 1 quintal = 100 pounds.

Troy Weight.

24 grains	= 1 pennyweight, dwt.
20 pennyweights	= 1 ounce, oz. = 480 grains.
12 ounces	= 1 pound, lb. = 5760 grains.

Troy weight is used for weighing gold and silver. The grain is the same in Avoirdupois, Troy, and Apothecaries' weights. A carat, used in weighing diamonds = 3.168 grains = .205 gramme.

Apothecaries' Weight.

20 grains	= 1 scruple, ℈	
3 scruples	= 1 drachm, ʒ	= 60 grains.
8 drachms	= 1 ounce, ℥	= 480 grains.
12 ounces	= 1 pound, lb.	= 5760 grains.

To determine whether a balance has unequal arms.—After weighing an article and obtaining equilibrium, transpose the article and the weights. *If the balance is true, it will remain in equilibrium;* if untrue, the pan suspended from the longer arm will descend.

To weigh correctly on an incorrect balance.—First, by substitution. *Put the article to be weighed* in one pan of the balance a counterpoise it by any convenient heavy articles placed on the other pan. Remove the article to be weighed and substitute for it standard weights until equipoise is again established. The amount of these weights is the weight of the article.

Second, by transposition. Determine the apparent weight of the article as usual, then its apparent weight after transposing the article and the weights. If the difference is small, add half the difference to the smaller of the apparent weights to obtain the true weight. If the difference is 2 per cent the error of this method is 1 part in 10,000. For larger differences, or to obtain a perfectly accurate result, multiply the two apparent weights together and extract the square root of the product.

Circular Measure.

60 seconds, ″ = 1 minute, ′.
60 minutes, ′ = 1 degree, °.
90 degrees = 1 quadrant.
360 " = circumference.

Time.

60 seconds = 1 minute.
60 minutes = 1 hour.
24 hours = 1 day.
7 days = 1 week.
365 days, 5 hours, 48 minutes 48 seconds = 1 year.

By the Gregorian Calendar every year whose number is divisible by 4 is a leap year, and contains 366 days, the other years containing 365 days, except that the centesimal years are leap years only when the number of the year is divisible by 400.

The comparative values of mean solar and sidereal time are shown by the following relations according to Bessel:

$$365.24222 \text{ mean solar days} = 366.24222 \text{ sidereal days, whence}$$
$$1 \text{ mean solar day} = 1.00273791 \text{ sidereal days;}$$
$$1 \text{ sidereal day} = 0.99726957 \text{ mean solar day;}$$
$$24 \text{ hours mean solar time} = 24^h\ 3^m\ 56^s.555 \text{ sidereal time;}$$
$$24 \text{ hours sidereal time} = 23^h\ 56^m\ 4^s.091 \text{ mean solar time,}$$

whence 1 mean solar day is $3^m\ 55^s.91$ longer than a sidereal day, reckoned in mean solar time.

BOARD AND TIMBER MEASURE.

Board Measure.

In board measure boards are assumed to be one inch in thickness. To obtain the number of feet board measure (B. M.) of a board or stick of square timber, multiply together the length in feet, the breadth in feet, and the thickness in inches.

To compute the measure or surface in square feet.—When all dimensions are in feet, multiply the length by the breadth, and the product will give the surface required.

When either of the dimensions are in inches, multiply as above and divide the product by 12.

When all dimensions are in inches, multiply as before and divide product by 144.

Timber Measure.

To compute the volume of round timber.—When all dimensions are in feet, multiply the length by one quarter of the product of the mean girth and diameter, and the product will give the measurement in cubic feet. When length is given in feet and girth and diameter in inches, divide the product by 144; when all the dimensions are in inches, divide by 1728.

To compute the volume of square timber.—When all dimensions are in feet, multiply together the length, breadth, and depth; the product will be the volume in cubic feet. When one dimension is given in inches, divide by 12; when two dimensions are in inches, divide by 144; when all three dimensions are in inches, divide by 1728.

Contents in Feet of Joists, Scantling, and Timber.

Length in Feet.

Size.	12	14	16	18	20	22	24	26	28	30
					Feet Board Measure.					
2 × 4	8	9	11	12	13	15	16	17	19	20
2 × 6	12	14	16	18	20	22	24	26	28	30
2 × 8	16	19	21	24	27	29	32	35	37	40
2 × 10	20	23	27	30	33	37	40	43	47	50
2 × 12	24	28	32	36	40	44	48	52	56	60
2 × 14	28	33	37	42	47	51	56	61	65	70
3 × 8	24	28	32	36	40	44	48	52	56	60
3 × 10	30	35	40	45	50	55	60	65	70	75
3 × 12	36	42	48	54	60	66	72	78	84	90
3 × 14	42	49	56	63	70	77	84	91	98	105
4 × 4	16	19	21	24	27	29	32	35	37	40
4 × 6	24	28	32	36	40	44	48	52	56	60
4 × 8	32	37	43	48	53	59	64	69	75	80
4 × 10	40	47	53	60	67	73	80	87	93	100
4 × 12	48	56	64	72	80	88	96	104	112	120
4 × 14	56	65	75	84	93	103	112	121	131	140
6 × 6	36	42	48	54	60	66	72	78	84	90
6 × 8	48	56	64	72	80	88	96	104	112	120
6 × 10	60	70	80	90	100	110	120	130	140	150
6 × 12	72	84	96	108	120	132	144	156	168	180
6 × 14	84	98	112	126	140	154	168	182	196	210
8 × 8	64	75	85	96	107	117	128	139	149	160
8 × 10	80	93	107	120	133	147	160	173	187	200
8 × 12	96	112	128	144	160	176	192	208	224	240
8 × 14	112	131	149	168	187	205	224	243	261	280
10 × 10	100	117	133	150	167	183	200	217	233	250
10 × 12	120	140	160	180	200	220	240	260	280	300
10 × 14	140	163	187	210	233	257	280	303	327	350
12 × 12	144	168	192	216	240	264	288	312	336	360
12 × 14	168	196	224	252	280	308	336	364	392	420
14 × 14	196	229	261	294	327	359	392	425	457	490

FRENCH OR METRIC MEASURES.

The metric unit of length is the metre = 39.37 inches.
The metric unit of weight is the gram = 15.432 grains.
The following prefixes are used for subdivisions and multiples; Milli = $\frac{1}{1000}$, Centi = $\frac{1}{100}$, Deci = $\frac{1}{10}$, Deca = 10, Hecto = 100, Kilo = 1000, Myria = 10,000.

FRENCH AND BRITISH (AND AMERICAN) EQUIVALENT MEASURES.

Measures of Length.

French.	British and U. S.
1 metre	= 39.37 inches, or 3.28083 feet, or 1.09361 yards.
.3048 metre	= 1 foot.
1 centimetre	= .3937 inch.
2.54 centimetres	= 1 inch.
1 millimetre	= .03937 inch, or 1/25 inch, nearly.
25.4 millimetres	= 1 inch.
1 kilometre	= 1093.61 yards, or 0.62137 mile.

Measures of Surface.

FRENCH.		BRITISH.
1 square metre	=	10.764 square feet, 1.196 square yards.
.836 square metre	=	1 square yard.
.0929 square metre	=	1 square foot.
1 square centimetre	=	.155 square inch.
6.452 square centimetres	=	1 square inch.
1 square millimetre	=	.00155 square inch.
645.2 square millimetres	=	1 square inch.
1 centiare = 1 sq. metre	=	10.764 square feet.
1 are = 1 sq. decametre	=	1076.41 " "
1 hectare = 100 ares	=	107641 " " = 2.4711 acres.
1 sq. kilometre	=	.386109 sq. miles = 247.11 "
1 sq. myriametre	=	38.6109 " "

Of Volume.

FRENCH.		BRITISH and U. S.
1 cubic metre	=	35.314 cubic feet, 1.308 cubic yards.
.7645 cubic metre	=	1 cubic yard.
.02832 cubic metre	=	1 cubic foot.
1 cubic decimetre	=	61.023 cubic inches, .0353 cubic foot.
28.32 cubic decimetres	=	1 cubic foot.
1 cubic centimetre	=	.061 cubic inch.
16.387 cubic centimetres	=	1 cubic inch.
cubic centimetre = 1 millilitre	=	.061 cubic inch.
centilitre =	=	.610 " "
decilitre =	=	6.102 " "
litre = 1 cubic decimetre	=	61.023 " " = 1.05671 quart
hectolitre or decistere	=	3.314 cubic feet = 2.8375 bush
stere, kilolitre, or cubic metre	=	1.308 cubic yards = 28.37 bush

Of Capacity.

FRENCH.		BRITISH and U. S.
1 litre (= 1 cubic decimetre)	=	61.023 cubic inches, .03531 cubic foot, .2642 gallon (American), 2.202 pounds of water at 62°
28.317 litres	=	1 cubic foot.
4.543 litres	=	1 gallon (British).
3.785 litres	=	1 gallon (American).

Of Weight.

FRENCH.		BRITISH and U. S.
1 gramme	=	15.432 grains.
.0648 gramme	=	1 grain.
28.35 gramme	=	1 ounce avoirdupois.
1 kilogramme	=	2.2046 pounds.
.4536 kilogramme	=	1 pound.
1 tonne or metric ton = 1000 kilogrammes	=	.9842 ton of 2240 pounds, 19.68 cwts., 2204.6 pounds.
1.016 metric tons = 1016 kilogrammes	=	1 ton of 2240 pounds.

Ir. O. H. Titmann, in Bulletin No. 9 of the U. S. Coast and Geode
. discusses the work of various authorities who have compared t
l the metre, and by referring all the observations to a common s
s succeeded in reconciling the discrepancies within very narrow
e following are his results for the number of inches in a metre ac
the comparisons of the authorities named:

1817.	Hassler	39.36994 inches.
1818.	Kater	39.36990 "
1835.	Baily	39.36973 "
[illegible]	Clarke	39.36970 "
	[illegible]	39.36984 "
	…se is	39.36982 "

METRIC CONVERSION TABLES.

The following tables, with the subjoined memoranda, were published in 1890 by the United States Coast and Geodetic Survey, office of standard weights and measures, T. C. Mendenhall, Superintendent.

Tables for Converting U. S. Weights and Measures—Customary to Metric.

LINEAR.

	Inches to Millimetres.	Feet to Metres.	Yards to Metres.	Miles to Kilometres.
1 =	25.4001	0.304801	0.914402	1.60935
2 =	50.8001	0.609601	1.828804	3.21869
3 =	76.2002	0.914402	2.743205	4.82804
4 =	101.6002	1.219202	3.657607	6.43739
5 =	127.0003	1.524003	4.572009	8.04674
6 =	152.4003	1.828804	5.486411	9.65608
7 =	177.8004	2.133604	6.400813	11.26543
8 =	203.2004	2.438405	7.315215	12.87478
9 =	228.6005	2.743205	8.229616	14.48412

SQUARE.

	Square Inches to Square Centimetres.	Square Feet to Square Decimetres.	Square Yards to Square Metres.	Acres to Hectares.
1 =	6.452	9.290	0.836	0.4047
2 =	12.903	18.581	1.672	0.8094
3 =	19.355	27.871	2.508	1.2141
4 =	25.807	37.161	3.344	1.6187
5 =	32.258	46.452	4.181	2.0234
6 =	38.710	55.742	5.017	2.4281
7 =	45.161	65.032	5.853	2.8328
8 =	51.613	74.323	6.689	3.2375
9 =	58.065	83.613	7.525	3.6422

CUBIC.

	Cubic Inches to Cubic Centimetres.	Cubic Feet to Cubic Metres.	Cubic Yards to Cubic Metres.	Bushels to Hectolitres.
1 =	16.387	0.02832	0.765	0.35242
2 =	32.774	0.05663	1.529	0.70485
3 =	49.161	0.08495	2.294	1.05727
4 =	65.549	0.11327	3.058	1.40969
5 =	81.936	0.14158	3.823	1.76211
6 =	98.323	0.16990	4.587	2.11454
7 =	114.710	0.19822	5.352	2.46696
8 =	131.097	0.22654	6.116	2.81938
9 =	147.484	0.25485	6.881	3.17181

WIRE AND SHEET-METAL GAUGES COMPARI

Number of Gauge.	Birmingham Wire Gauge.	American or Brown and Sharpe Gauge.	Roebling's and Washburn & Moen's Gauge.	Trenton Iron Co.'s Wire Gauge.	British Imperial Standard Wire Gauge. (Legal Standard in Great Britain since March 1, 1884.)		U. S. Standard Gauge for Sheet and Plate Iron and Steel. (Legal Standard since July 1, 1893.)
	inch.	inch.	inch.	inch.	inch.	millim.	inch.
0000000			.49		.500	12.7	.5
000000			.46		.464	11.78	.469
00000			.43	.45	.432	10.97	.438
0000	.454	.46	.393	.40	.4	10.16	.406
000	.425	.40964	.362	.36	.372	9.45	.375
00	.38	.3648	.331	.33	.348	8.84	.344
0	.34	.32486	.307	.305	.324	8.23	.313
1	.3	.2893	.283	.285	.3	7.62	.281
2	.284	.25763	.263	.265	.276	7.01	.266
3	.259	.22942	.244	.245	.252	6.4	.25
4	.238	.20431	.225	.225	.232	5.89	.234
5	.22	.18194	.207	.205	.212	5.38	.219
6	.203	.16202	.192	.19	.192	4.88	.203
7	.18	.14428	.177	.175	.176	4.47	.188
8	.165	.12849	.162	.16	.16	4.06	.172
9	.148	.11443	.148	.145	.144	3.66	.156
10	.134	.10189	.135	.13	.128	3.26	.141
11	.12	.09074	.12	.1175	.116	2.95	.125
12	.109	.08081	.105	.105	.104	2.64	.109
13	.095	.07196	.092	.0925	.092	2.34	.094
14	.083	.06408	.08	.08	.08	2.03	.078
15	.072	.05707	.072	.07	.072	1.83	.07
16	.065	.05082	.063	.061	.064	1.63	.0625
17	.058	.04526	.054	.0525	.056	1.42	.0563
18	.049	.0403	.047	.045	.048	1.22	.05
19	.042	.03589	.041	.04	.04	1.01	.0438
20	.035	.03196	.035	.035	.036	.91	.0375
21	.032	.02846	.032	.031	.032	.81	.0344
22	.028	.02535	.028	.028	.028	.71	.0313
23	.025	.02257	.025	.025	.024	.61	.0281
24	.022	.0201	.023	.0225	.022	.56	.025
25	.02	.0179	.02	.02	.02	.51	.0219
26	.018	.01594	.018	.018	.018	.45	.0188
27	.016	.01419	.017	.017	.0164	.42	.0172
28	.014	.01264	.016	.016	.0148	.38	.0156
29	.013	.01126	.015	.015	.0136	.35	.0141
30	.012	.01002	.014	.014	.0124	.31	.0125
31	.01	.00893	.0135	.013	.0116	.29	.0109
32	.009	.00795	.013	.012	.0108	.27	.0101
33	.008	.00708	.011	.011	.01	.25	.0094
34	.007	.0063	.01	.01	.0092	.23	.0086
35	.005	.00561	.0095	.0095	.0084	.21	.0078
36	.004	.005	.009	.009	.0076	.19	.007
37		.00445	.0085	.0085	.0068	.17	.0066
38		.00396	.008	.008	.006	.15	.0063
39		.00353	.0075	.0075	.0052	.13	
40		.00314	.007	.007	.0048	.12	
41					.0044	.11	
42					.004	.10	
43					.0036	.09	
44					.0032	.08	
45					.0028	.07	
46					.0024	.06	
47					.002	.05	
48					.0016	.04	
49					.0012	.03	
50					.001	.025	

EDISON, OR CIRCULAR MIL GAUGE, FOR ELECTRICAL WIRES.

Gauge Number.	Circular Mils.	Diameter in Mils.	Gauge Number.	Circular Mils.	Diameter in Mils.	Gauge Number.	Circular Mils.	Diameter in Mils.
3	3,000	54.78	70	70,000	264.58	190	190,000	435.89
5	5,000	70.72	75	75,000	273.87	200	200,000	447.22
8	8,000	89.45	80	80,000	282.85	220	220,000	469.05
12	12,000	109.55	85	85,000	291.55	240	240,000	489.90
15	15,000	122.48	90	90,000	300.00	260	260,000	509.91
20	20,000	141.43	95	95,000	308.23	280	280,000	529.16
25	25,000	158.12	100	100,000	316.23	300	300,000	547.73
30	30,000	173.21	110	110,000	331.67	320	320,000	565.69
35	35,000	187.09	120	120,000	346.42	340	340,000	583.10
40	40,000	200.00	130	130,000	360.56	360	360,000	600.00
45	45,000	212.14	140	140,000	374.17			
50	50,000	223.61	150	150,000	387.30			
55	55,000	234.58	160	160,000	400.00			
60	60,000	244.95	170	170,000	412.32			
65	65,000	254.96	180	180,000	424.27			

TWIST DRILL AND STEEL WIRE GAUGE.

(Morse Twist Drill and Machine Co.)

No.	Size.	No.	Size.	No.	Size.	No.	Size.
	inch.		inch.		inch.		inch.
1	.2280	16	.1770	31	.1200	46	.0810
2	.2210	17	.1730	32	.1160	47	.0785
3	.2130	18	.1695	33	.1130	48	.0760
4	.2090	19	.1660	34	.1110	49	.0730
5	.2055	20	.1610	35	.1100	50	.0700
6	.2040	21	.1590	36	.1065	51	.0670
7	.2010	22	.1570	37	.1040	52	.0635
8	.1990	23	.1540	38	.1015	53	.0595
9	.1960	24	.1520	39	.0995	54	.0550
10	.1935	25	.1495	40	.0980	55	.0520
11	.1910	26	.1470	41	.0960	56	.0465
12	.1890	27	.1440	42	.0935	57	.0430
13	.1850	28	.1405	43	.0890	58	.0420
14	.1820	29	.1360	44	.0860	59	.04[illegible]0
15	.1800	30	.1285	45	.0820	60	.0400

STEEL MUSIC-WIRE GAUGE.

(Washburn & Moen Mfg. Co.)

No.	Size.	No.	Size.	No.	Size.	No.	Size.
	inch.		inch.		inch.		inch.
12	.0295	17	.0378	21	.0461	25	.0585
13	.0311	18	.0395	22	.0181	26	.0626
14	.0325	19	.0414	23	.0506	27	.0663
15	.0343	20	.043	24	.0347	28	.0719
16	.0359						

THE EDISON OR CIRCULAR MIL WIRE GAUGE.

(For table of copper wires by this gauge, giving weights, electrical resistances, etc., see Copper Wire.)

Mr. C. J. Field (*Stevens Indicator*, July, 1887) thus describes the origin of the Edison gauge:

The Edison company experienced inconvenience and loss by not having a wide enough range nor sufficient number of sizes in the existing gauges. This was felt more particularly in the central-station work in making electrical determinations for the street system. They were compelled to make use of two of the existing gauges at least, thereby introducing a complication that was liable to lead to mistakes by the contractors and linemen.

In the incandescent system an even distribution throughout the entire system and a uniform pressure at the point of delivery are obtained by calculating for a given maximum percentage of loss from the potential as delivered from the dynamo. In carrying this out, on account of lack of regular sizes, it was often necessary to use larger sizes than the occasion demanded, and even to assume new sizes for large underground conductors. It was also found that nearly all manufacturers based their calculation for the conductivity of their wire on a variety of units, and that not one used the latest unit as adopted by the British Association and determined from Dr. Matthiessen's experiments; and as this was the unit employed in the manufacture of the Edison lamps, there was a further reason for constructing a new gauge. The engineering department of the Edison company, knowing the requirements, have designed a gauge that has the widest range obtainable and a large number of sizes which increase in a regular and uniform manner. The basis of the graduation is the sectional area, and the number of the wire corresponds. A wire of 100,000 circular mils area is No. 100; a wire of one half the size will be No. 50; twice the size No. 200.

In the older gauges, as the number increased the size decreased. With this gauge, however, the number increases with the wire, and the number multiplied by 1000 will give the circular mils.

The weight per mil-foot, 0.00000302705 pounds, agrees with a specific gravity of 8.889, which is the latest figure given for copper. The ampere capacity which is given was deduced from experiments made in the company's laboratory, and is based on a rise of temperature of 50° F. in the wire.

In 1893 Mr. Field writes, concerning gauges in use by electrical engineers:

The B. and S. gauge seems to be in general use for the smaller sizes up to 100,000 c. m., and in some cases a little larger. From between one and two hundred thousand circular mils upwards, the Edison gauge or its equivalent is practically in use, and there is a general tendency to designate all sizes above this in circular mils, specifying a wire as 200,000, 400,000, 500,000, or 1,000,000 c. m.

In the electrical business there is a large use of copper wire and rod and other materials of these large sizes, and in ordering them, speaking of them, specifying, and in every other use, the general method is to simply specify the circular milage. I think it is going to be the only system in the future for the designation of wires, and the attaining of it means practically the adoption of the Edison gauge or the method and basis of this gauge as the correct one for wire sizes.

THE U. S. STANDARD GAUGE FOR SHEET AND PLATE IRON AND STEEL, 1893.

The Committee on Coinage, Weights, and Measures of the House of Representatives in 1893, in introducing the bill establishing the new sheet and plate gauge, made a report from which we take the following:

The purpose of this bill is to establish an authoritative standard gauge for the measurement of sheet and plate iron.

There is in this country no uniform or standard gauge, and the same numbers in different gauges represent different thicknesses of sheets or plates. This has given rise to much misunderstanding and friction between employers and workmen and mistakes and fraud between dealers and consumers.

The practice of describing the different thicknesses of sheet and plate iron by gauge numbers has been so long established and become so universal both here and in Great Britain that it is not deemed advisable to change this mode of designation: but these descriptive gauge num-

. S. STANDARD GAUGE FOR SHEET AND PLATE IRON AND STEEL, 1893.

Gauge.	Approximate Thickness in Fractions of an Inch.	Approximate Thickness in Decimal Parts of an Inch.	Approximate Thickness in Millimeters.	Weight per Square Foot in Ounces Avoirdupois.	Weight per Square Foot in Pounds Avoirdupois.	Weight per Square Foot in Kilograms.	Weight per Square Meter in Kilograms.	Weight per Square Meter in Pounds Avoirdupois
0000000	1-2	0.5	12.7	320	20.	9.072	97.65	215.28
000000	15-32	0.46875	11.90625	300	18.75	8.505	91.55	201.82
00000	7-16	0.4375	11.1125	280	17.50	7.938	85.44	188.37
0000	13-32	0.40625	10.31875	260	16.25	7.371	79.33	174.91
000	3-8	0.375	9.525	240	15.	6.804	73.24	161.46
00	11-32	0.34375	8.73125	220	13.75	6.237	67.13	148.00
0	5-16	0.3125	7.9375	200	12.50	5.67	61.03	134.55
1	9-32	0.28125	7.14375	180	11.25	5.103	54.93	121.09
2	17-64	0.265625	6.746875	170	10.625	4.819	51.88	114.37
3	1-4	0.25	6.35	160	10.	4.536	48.82	107.64
4	15-64	0.234375	5.953125	150	9.375	4.252	45.77	100.91
5	7-32	0.21875	5.55625	140	8.75	3.969	42.72	94.18
6	13-64	0.203125	5.159375	130	8.125	3.685	39.67	87.45
7	3-16	0.1875	4.7625	120	7.5	3.402	36.62	80.72
8	11-64	0.171875	4.365625	110	6.875	3.118	33.57	74.00
9	5-32	0.15625	3.96875	100	6.25	2.835	30.52	67.27
10	9-64	0.140625	3.571875	90	5.625	2.552	27.46	60.55
11	1-8	0.125	3.175	80	5.	2.268	24.41	53.82
12	7-64	0.109375	2.778125	70	4.375	1.984	21.36	47.09
13	3-32	0.09375	2.38125	60	3.75	1.701	18.31	40.36
14	5-64	0.078125	1.984375	50	3.125	1.417	15.26	33.64
15	9-128	0.0703125	1.7859375	45	2.8125	1.276	13.73	30.27
16	1-16	0.0625	1.5875	40	2.5	1.134	12.21	26.91
17	9-160	0.05625	1.42875	36	2.25	1.021	10.99	24.22
18	1-20	0.05	1.27	32	2.	0.9072	9.765	21.53
19	7-160	0.04375	1.11125	28	1.75	0.7938	8.544	18.84
20	3-80	0.0375	0.9525	24	1.50	0.6804	7.324	16.15
21	11-320	0.034375	0.873125	22	1.375	0.6237	6.713	14.80
22	1-32	0.03125	0.793750	20	1.25	0.567	6.103	13.46
23	9-320	0.028125	0.714375	18	1.125	0.5103	5.493	12.11
24	1-40	0.025	0.635	16	1.	0.4536	4.882	10.76
25	7-320	0.021875	0.555625	14	0.875	0.3969	4.272	9.42
26	3-160	0.01875	0.47625	12	0.75	0.3402	3.662	8.07
27	11-640	0.0171875	0.4365625	11	0.6875	0.3119	3.357	7.40
28	1-64	0.015625	0.396875	10	0.625	0.2835	3.052	6.73
29	9-640	0.0140625	0.3571875	9	0.5625	0.2551	2.746	6.05
30	1-80	0.0125	0.3175	8	0.5	0.2268	2.441	5.38
31	7-640	0.0109375	0.2778125	7	0.4375	0.1984	2.136	4.71
32	13-1280	0.01015625	0.25796875	6½	0.40625	0.1843	1.983	4.37
33	3-320	0.009375	0.238125	6	0.375	0.1701	1.831	4.04
34	11-1280	0.00859375	0.21828125	5½	0.34375	0.1559	1.678	3.7
35	5-640	0.0078125	0.1984375	5	0.3125	0.1417	1.526	3.3
36	9-1280	0.00703125	0.17859375	4½	0.28125	0.1276	1.373	3.
37	17-2560	0.006640625	0.168671875	4¼	0.265625	0.1205	1.297	2
38	1-160	0.00625	0.15875	4	0.25	0.1134	1.221	2

WIRE AND SHEET-METAL GAUGES COMPAR

Number of Gauge.	Birmingham Wire Gauge.	American or Brown and Sharpe Gauge.	Roebling's and Washburn & Moen's Gauge.	Trenton Iron Co.'s Wire Gauge.	British Imperial Standard Wire Gauge. (Legal Standard in Great Britain since March 1, 1884.)		U. S. Standard Gauge for Sheet and Plate Iron and Steel. (Legal Standard since July 1, 1893.)
	inch.	inch.	inch.	inch.	inch.	millim.	inch.
0000000			.49		.500	12.7	.5
000000			.46		.464	11.78	.469
00000			.43	.45	.432	10.97	.438
0000	.454	.46	.393	.40	.4	10.16	.406
000	.425	.40964	.362	.36	.372	9.45	.375
00	.38	3648	.331	.33	.348	8.84	.344
0	.34	.32486	.307	.305	.324	8.23	.313
1	.3	.2893	.283	.285	.3	7.62	.281
2	.284	.25763	.263	.265	.276	7.01	.266
3	.259	.22942	.244	.245	.252	6.4	.25
4	.238	.20431	.225	.225	.232	5.89	.234
5	.22	.18194	.207	.205	.212	5.38	.219
6	.203	.16202	.192	.19	.192	4.88	.203
7	.18	.14428	.177	.175	.176	4.47	.188
8	.165	.12849	.162	.16	.16	4.06	.172
9	.148	.11443	.148	.145	.144	3.66	.156
10	.134	.10189	.135	.13	.128	3.26	.141
11	.12	.09074	.12	.1175	.116	2.95	.125
12	.109	.08081	.105	.105	.104	2.64	.109
13	.095	.07196	.092	.0925	.092	2.34	.094
14	.083	.06408	.08	.08	.08	2.03	.078
15	.072	.05707	.072	.07	.072	1.83	.07
16	.065	.05082	.063	.061	.064	1.63	.0625
17	.058	.04526	.054	.0525	.056	1.42	.0563
18	.049	.0403	.047	.045	.048	1.22	.05
19	.042	.03589	.041	.04	.04	1.01	.0438
20	.035	.03196	.035	.035	.036	.91	.0375
21	.032	.02846	.032	.031	.032	.81	.0344
22	.028	.02535	.028	.028	.028	.71	.0313
23	.025	.02257	.025	.025	.024	.61	.0281
24	.022	.0201	.023	.0225	.022	.56	.025
25	.02	.0179	.02	.02	.02	.51	.0219
26	.018	.01594	.018	.018	.018	.45	.0188
27	.016	.01419	.017	.017	.0164	.42	.0172
28	.014	.01264	.016	.016	.0148	.38	.0156
29	.013	.01126	.015	.015	.0136	.35	.0141
30	.012	.01002	.014	.014	.0124	.31	.0125
31	.01	.00893	.0135	.013	.0116	.29	.0109
32	.009	.00795	.013	.012	.0108	.27	.0101
33	.008	.00708	.011	.011	.01	.25	.0094
34	.007	.0063	.01	.01	.0092	.23	.0086
35	.005	.00561	.0095	.0095	.0084	.21	.0078
36	.004	.005	.009	.009	.0076	.19	.007
37		.00445	.0085	.0085	.0068	.17	.0066
38		.00396	.008	.008	.006	.15	.0063
39		.00353	.0075	.0075	.0052	.13	
40		.00314	.007	.007	.0048	.12	
41					.0044	.11	
42					.004	.10	
43					.0036	.09	
44					.0032	.08	
45					.0028	.07	
46					.0024	.06	
47					.002	.05	
48					.0016	.04	
49					.0012	.03	
50					.001	.025	

EDISON, OR CIRCULAR MIL GAUGE, FOR ELECTRICAL WIRES.

Gauge Number.	Circular Mils.	Diameter in Mils.	Gauge Number.	Circular Mils.	Diameter in Mils.	Gauge Number.	Circular Mils.	Diameter in Mils.
3	3,000	54.78	70	70,000	264.58	190	190,000	435.8
5	5,000	70.72	75	75,000	273.87	200	200,000	447.
8	8,000	89.45	80	80,000	282.85	220	220,000	469.
12	12,000	109.55	85	85,000	291.55	240	240,000	489.9
15	15,000	122.48	90	90,000	300.00	260	260,000	509.9
20	20,000	141.43	95	95,000	308.23	280	280,000	529.1
25	25,000	158.12	100	100,000	316.23	300	300,000	547.7
30	30,000	173.21	110	110,000	331.67	320	320,000	565.
35	35,000	187.09	120	120,000	346.42	340	340,000	583.1
40	40,000	200.00	130	130,000	360.56	360	360,000	600.
45	45,000	212.14	140	140,000	374.17			
50	50,000	223.61	150	150,000	387.30			
55	55,000	234.53	160	160,000	400.00			
60	60,000	244.95	170	170,000	412.32			
65	65,000	254.96	180	180,000	424.27			

TWIST DRILL AND STEEL WIRE GAUGE.

(Morse Twist Drill and Machine Co.)

No.	Size.	No.	Size.	No.	Size.	No.	Size.
	inch.		inch.		inch.		inch.
1	.2280	16	.1770	31	.1200	46	.081
2	.2210	17	.1730	32	.1160	47	.0785
3	.2130	18	.1695	33	.1130	48	.076
4	.2090	19	.1660	34	.1110	49	.073
5	.2055	20	.1610	35	.1100	50	.070
6	.2040	21	.1590	36	.1065	51	.067
7	.2010	22	.1570	37	.1040	52	.0635
8	.1990	23	.1540	38	.1015	53	.0595
9	.1960	24	.1520	39	.0995	54	.055
10	.1935	25	.1495	40	.0980	55	.052
11	.1910	26	.1470	41	.0960	56	.0465
12	.1890	27	.1440	42	.0935	57	.0430
13	.1850	28	.1405	43	.0890	58	.042
14	.1820	29	.1360	44	.0860	59	.041
15	.1800	30	.1285	45	.0820	60	.0400

STEEL MUSIC-WIRE GAUGE.

(Washburn & Moen Mfg. Co.)

No.	Size.	No.	Size.	No.	Size.	No.	Size.
	inch.		inch.		inch.		inch.
12	.0295	17	.0378	21	.0461	25	.0585
13	.0311	18	.0395	22	.0481	26	[illegible]
14	.0325	19	.0414	23	.0506	27	[illegible]
15	.0343	20	.043	24	.0547	28	[illegible]
16	.0359						

Parentheses.—When a parenthesis is preceded by a plus sign it ma[y be] removed without changing the value of the expression: $a + b + (a + $ ... $2a + 2b$. When a parenthesis is preceded by a minus sign it may be rem[oved] if we change the signs of all the terms within the parenthesis: $1 - (a$ ... $- c) = 1 - a + b + c$. When a parenthesis is within a parenthesis re[move] the inner one first: $a - \left[b - \left\{c - (d - e)\right\}\right] = a - \left[b - \left\{c - d + \right.\right.$ ... $= a - [b - c + d - e] = a - b + c - d + e$.

A multiplication sign, $\times$, has the effect of a parenthesis, in that the o[per]ation indicated by it must be performed before the operations of add[ition] or subtraction. $a + b \times a + b = a + ab + b$; while $(a + b) \times (a +$ ... $a^2 + 2ab + b^2$, and $(a + b) \times a + b = a^2 + ab + b$.

Division.—The quotient is positive when the dividend and di[visor] have like signs, and negative when they have unlike signs: $abc \div b =$ ... $abc \div - b = - ac$.

To divide a monomial by a monomial, write the dividend over the div[isor] with a line between them. If the expressions have common factors, re[move] the common factors:

$$a^2bx \div aby = \frac{a^2bx}{aby} = \frac{ax}{y}; \quad \frac{a^4}{a^3} = a; \quad \frac{a^3}{a^5} = \frac{1}{a^2} = a^{-2}.$$

To divide a polynomial by a monomial, divide each term of the polyno[mial] by the monomial: $(8ab - 12ac) \div 4a = 2b - 3c$.

To divide a polynomial by a polynomial, arrange both dividend and [divi]sor in the order of the ascending or descending powers of some com[mon] letter and keep this arrangement throughout the operation

Divide the first term of the dividend by the first term of the divisor, [and] write the result as the first term of the quotient.

Multiply all the terms of the divisor by the first term of the quotient [and] subtract the product from the dividend. If there be a remainder, cons[ider] it as a new dividend and proceed as before: $(a^2 - b^2) \div (a + b)$.

$$\begin{array}{l|l} a^2 - b^2 & a + b. \\ \underline{a^2 + ab} & \overline{a - b.} \\ \quad - ab - b^2. & \\ \quad \underline{- ab - b^2.} & \end{array}$$

The difference of two equal odd powers of any two numbers is divi[sible] by their difference and also by their sum:

$(a^3 - b^3) \div (a - b) = a^2 + ab + b^2$; $(a^3 - b^3) \div (a + b) = a^2 - ab +$ ...

The difference of two equal even powers of two numbers is divisibl[e by] their difference and also by their sum: $(a^2 - b^2) \div (a - b) = a + b$.

The sum of two equal even powers of two numbers is not divisibl[e by] either the difference or the sum of the numbers; but when the expo[nent] of each of the two equal powers is composed of an odd and an even fa[ctor,] the sum of the given power is divisible by the sum of the powers expre[ssed] by the even factor. Thus $x^6 + y^6$ is not divisible by $x + y$ or by $x - y$, b[ut is] divisible by $x^2 + y^2$.

Simple equations.—An equation is a statement of equality bet[ween] two expressions; as, $a + b = c + d$.

A simple equation, or equation of the first degree, is one which con[tains] only the first power of the unknown quantity. If equal changes be m[ade] (by addition, subtraction, multiplication, or division) in both sides o[f an] equation, the results will be equal.

Any term may be changed from one side of an equation to another, [pro]vided its sign be changed: $a + b = c + d$; $a = c + d - b$. To solv[e an] equation having one unknown quantity, transpose all the terms invol[ving] the unknown quantity to one side of the equation, and all the other t[erms] to the other side; combine like terms, and divide both sides by the coeffi[cient] of the unknown quantity.

Solve $8x - 29 = 26 - 3x$. $8x + 3x = 29 + 26$; $11x = 55$; $x = 5$, ans.

Simple algebraic problems containing one unknown quantity are so[lved] by making $x =$ the unknown quantity, and stating the conditions o[f the] problem in the form of an algebraic equation, and then solving the e[qua]tion. What two numbers are those whose sum is 48 and difference 14? [Let] $x =$ the smaller number, $x + 14$ the greater. $x + x + 14 = 48$. $2x =$ ... $= 17$; $x + 14 = 31$, ans.

Find a number whose treble exceeds 50 as much as its double falls [short] *of 40.* Let $x =$ the number. $3x - 50 = 40 - 2x$; $5x = 90$; $x = 18$, ans. $54 - 50 =$ ...

Equations containing two unknown quantities.—If one equation contains two unknown quantities, x and y, an indefinite number of pairs of values of x and y may be found that will satisfy the equation, but if a second equation be given only one pair of values can be found that will satisfy both equations. Simultaneous equations, or those that may be satisfied by the same values of the unknown quantities, are solved by combining the equations so as to obtain a single equation containing only one unknown quantity. This process is called elimination.

Elimination by addition or subtraction.—Multiply the equation by such numbers as will make the coefficients of one of the unknown quantities equal in the resulting equation. Add or subtract the resulting equations according as they have unlike or like signs.

Solve $\begin{cases} 2x + 3y = 7. \\ 4x - 5y = 3. \end{cases}$ Multiply by 2: $4x + 6y = 14$; Subtract: $4x - 5y = 3$; $11y = 11$; $y = 1$.

Substituting value of y in first equation, $2x + 3 = 7$; $x = 2$.

Elimination by substitution.—From one of the equations obtain the value of one of the unknown quantities in terms of the other. Substitute for this unknown quantity its value in the other equation and reduce the resulting equations.

Solve $\begin{cases} 2x + 3y = 8. & (1). \\ 3x + 7y = 7. & (2). \end{cases}$ From (1) we find $x = \dfrac{8 - 3y}{2}$.

Substitute this value in (2): $3\left(\dfrac{8 - 3y}{2}\right) + 7y = 7$; $= 24 - 9y + 14y = 14$, whence $y = -2$. Substitute this value in (1): $2x - 6 = 8$; $x = 7$.

Elimination by comparison.—From each equation obtain the value of one of the unknown quantities in terms of the other. Form an equation from these equal values, and reduce this equation.

Solve $\begin{cases} 2x - 9y = 11. & (1). \text{ From (1) we find } x = \dfrac{11 + 9y}{2}. \\ 3x - 4y = 7. & (2). \text{ From (2) we find } x = \dfrac{7 + 4y}{3}. \end{cases}$

Equating these values of x, $\dfrac{11 + 9y}{2} = \dfrac{7 + 4y}{3}$; $19y = -19$; $y = -1$.

Substitute this value of y in (1): $2x + 9 = 11$; $x = 1$.

If three simultaneous equations are given containing three unknown quantities, one of the unknown quantities must be eliminated between two pairs of the equations; then a second between the two resulting equations.

Quadratic equations.—A quadratic equation contains the square of the unknown quantity, but no higher power. A pure quadratic contains the square only; an affected quadratic both the square and the first power.

To solve a pure quadratic, collect the unknown quantities on one side, and the known quantities on the other; divide by the coefficient of the unknown quantity and extract the square root of each side of the resulting equation.

Solve $3x^2 - 15 = 0$. $3x^2 = 15$; $x^2 = 5$; $x = \sqrt{5}$

A root like $\sqrt{5}$, which is indicated, but which can be found only approximately, is called a *surd*.

Solve $3x^2 + 15 = 0$. $3x^2 = -15$; $x^2 = -5$; $x = \sqrt{-5}$.

The square root of -5 cannot be found even approximately, for the square of any number positive or negative is positive; therefore a root which is indicated, but cannot be found even approximately, is called *imaginary*.

To solve an affected quadratic.—1. Convert the equation into the form $a^2x^2 \pm 2abx = c$, multiplying or dividing the equation if necessary, so as to make the coefficient of x^2 a square number.

2. Complete the square of the first member of the equation, so as to convert it to the form of $a^2x^2 \pm 2abx + b^2$, which is the square of the binomial $ax \pm b$, as follows: *add to each side of the equation the square of the quotient obtained by dividing the second term by twice the square root of the first term.*

3. *Extract the square root of each side of the resulting equation.*

Solve $3x^2 - 4x = 32$. To make the coefficient of x^2 a square number multiply by 3: $9x^2 - 12x = 96$; $12x \div (2 \times 3x) = 2$; $2^2 = 4$.

Complete the square: $9x^2 - 12x + 4 = 100$. Extract the root: $3x - 2 =$

10, whence $x = 4$ or $-2\ 2/3$. The square root of 100 is either $+10$ or $-$ since the square of -10 as well as $+10^2 = 100$.

Problems involving quadratic equations have apparently two solutions, a quadratic has two roots. Sometimes both will be true solutions, but g erally one only will be a solution and the other be inconsistent with conditions of the problem.

The sum of the squares of two consecutive positive numbers is 481. F the numbers.

Let x = one number, $x + 1$ the other. $x^2 + (x + 1)^2 = 481$. $2x^2 + 2x -$ $= 481$.

$x^2 + x = 240$. Completing the square, $x^2 + x + 0.25 = 240.25$. Extracti the root we obtain $x + 0.5 = \pm 15.5$; $x = 15$ or -16.

The positive root gives for the numbers 15 and 16. The negative root 16 is inconsistent with the conditions of the problem.

Quadratic equations containing two unknown quantities require differ methods for their solution, according to the form of the equations. F these methods reference must be made to works on algebra.

Theory of exponents.—$\sqrt[n]{a}$ when n is a positive integer is one of equal factors of a. $\sqrt[n]{a^m}$ means a is to be raised to the mth power and t nth root extracted.

$(\sqrt[n]{a})^m$ means that the nth root of a is to be taken and the res raised to the mth power.

$\sqrt[n]{a^m} = (\sqrt[n]{a})^m = a^{\frac{m}{n}}$. When the exponent is a fraction, the numer tor indicates a power, and the denominator a root. $a^{\frac{6}{2}} = \sqrt{a^6} = a^3$; $a^{\frac{3}{2}}$ $\sqrt{a^3} = a^{1.5}$.

To extract the root of a quantity raised to an indicated power, divi the exponent by the index of the required root; as,

$$\sqrt[n]{a^m} = a^{\frac{m}{n}}; \qquad \sqrt[3]{a^6} = a^{\frac{6}{3}} = a^2.$$

Subtracting 1 from the exponent of a is equivalent to dividing by a:

$$a^{2-1} = a^1 = a;\ a^{1-1} = a^0 = \frac{a}{a} = 1;\ a^{0-1} = a^{-1} = \frac{1}{a};\ a^{-1-1} = a^{-2} =$$

A number with a negative exponent denotes the reciprocal of the numb with the corresponding positive exponent.

A factor under the radical sign whose root can be taken may, by havi the root taken, be removed from under the radical sign:

$$\sqrt{a^2b} = \sqrt{a^2} \times \sqrt{b} = a\sqrt{b}.$$

A factor outside the radical sign may be raised to the correspondi power and placed under it:

$$\sqrt{\frac{a}{b}} = \sqrt{\frac{ab}{b^2}} = \sqrt{ab \times \frac{1}{b^2}} = \frac{1}{b}\sqrt{ab}; \qquad \sqrt{\frac{a}{b^2}} = \frac{1}{b}\sqrt{a}$$

Binomial Theorem.—To obtain any power, as the nth, of an e pression of the form $x + a$

$$(a + x)^n = a^n + na^{n-1}x + \frac{n(n-1)a^{n-2}}{1.2}x^2 + \frac{n(n-1)(n-2)a^{n-3}}{1.2.3.}x^3$$

etc.

The following laws hold for any term in the expansion of $(a + x)^n$.

The exponent of x is less by one than the number of terms.

The exponent of a is n minus the exponent of x.

The last factor of the numerator is greater by one than the exponent of

The last factor of the denominator is the same as the exponent of x.

In the rth term the exponent of x will be $r - 1$.

The exponent of a will be $n - (r - 1)$, or $n - r + 1$.

The last factor of the numerator will be $n - r + 2$.

The last factor of the denominator will be $= r - 1$.

$$r\text{th} \qquad \frac{n(n-1)(n-2)\ldots(n-r+2)}{1.2.3\ldots(r-1)}a^{n-r+1}x^{r-1}$$

GEOMETRICAL PROBLEMS.

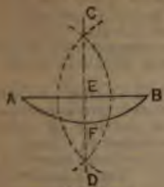


FIG. 1.

1. To bisect a straight line, or an arc of a circle (Fig. 1).—From the ends *A*, *B*, as centres, describe arcs intersecting at *C* and *D*, and draw a line through *C* and *D* which will bisect the line at *E* or the arc at *F*.

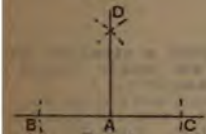


FIG. 2.

2. To draw a perpendicular to a straight line, or a radial line to a circular arc.—Same as in Problem 1. *C D* is perpendicular to the line *A B*, and also radial to the arc.

3. To draw a perpendicular to a straight line from a given point in that line (Fig. 2).—With any radius, from the given point *A* in the line *B C*, cut the line at *B* and *C*. With a longer radius describe arcs from *B* and *C*, cutting each other at *D*, and draw the perpendicular *D A*.

FIG. 3.

4. From the end *A* of a given line *A D* to erect a perpendicular *A E* (Fig. 3).—From any centre *F*, above *A D*, describe a circle passing through the given point *A*, and cutting the given line at *D*. Draw *D F* and produce it to cut the circle at *E*, and draw the perpendicular *A E*.

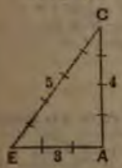


FIG. 4.

Second Method (Fig. 4).—From the given point *A* set off a distance *A E* equal to three parts, by any scale; and on the centres *A* and *E*, with radii of four and five parts respectively, describe arcs intersecting at *C*. Draw the perpendicular *A C*.

NOTE.—This method is most useful on very large scales, where straight edges are inapplicable. Any multiples of the numbers 3, 4, 5 may be taken with the same effect as 6, 8, 10, or 9, 12, 15.

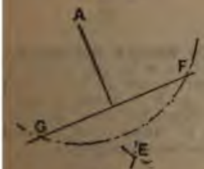


FIG. 5.

5. To draw a perpendicular to a straight line from any point without it (Fig. 5).—From the point *A*, with a sufficient radius cut the given line at *F* and *G*, and from these points describe arcs cutting at *E*. Draw the perpendicular *A E*.

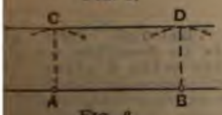


FIG. 6.

6. To draw a straight line parallel to a given line, at a given distance apart (Fig. 6).—From the centres A, B, in the given line, with the given distance as radius, describe arcs C, D, and draw the parallel lines C D touching the arcs.

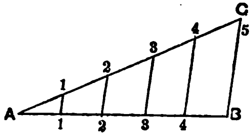


FIG. 7.

7. To divide a straig[h]
into a number of equa[l]
(Fig. 7).—To divide the line
say, five parts, draw the lin[e]
an angle from A; set off fi[ve]
parts; draw B 5 and draw pa[rallels]
it from the other points of di[vision]
A C. These parallels divide
required.

NOTE.—By a similar proces[s]
may be divided into a numb[er]
equal parts; setting off divi[sions]
A C, proportional by a scale t
quired divisions, and drawing
cutting A B. The triangles
A33, etc., are *similar triangl[es]*

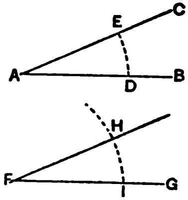


FIG. 8.

8. Upon a straight l
draw an angle equa[l]
given angle (Fig. 8).—Let
given angle and F G the line
the point A with any radius
the arc D E. From F with t
radius describe I H. Set off
I H equal to D E and draw F
angle F is equal to A, as requi[red]

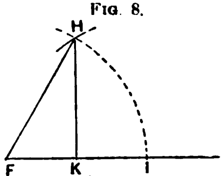


FIG. 9.

9. To draw angles
and 30° (Fig. 9).—From
any radius F I, describe an
and from I, with the same ra[dius]
the arc at H and draw F H
the required angle I F H. D[raw]
perpendicular H K to the bas[e]
form the angle of 30° F H K.

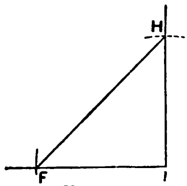


FIG. 10.

10. To draw an angle
(Fig. 10).—Set off the distar[nce]
draw the perpendicular I H
I F, and join H F to form the
F. The angle at H is also 45°.

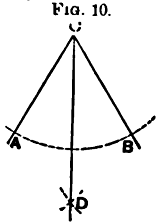
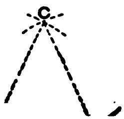


FIG. 11.

11. To bisect an angl[e]
11).—Let A C B be the angle;
as a centre draw an arc cutt[ing]
sides at A, B. From A an[d]
centres, describe arcs cutti[ng]
other at D. Draw C D, divi[ding]
angle into two equal parts.



12. Through two
points to describe an
a circle with a given r
(Fig. 12).—From the points A
as centres, with the given ra[dius]
scribe arcs cutting at C, a
? with the same radius de
ro A B.

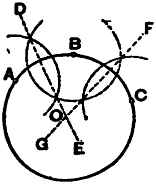


FIG. 13.

13. To find the centre of a circle or of an arc of a circle (Fig. 13).—Select three points, *A*, *B*, *C*, in the circumference, well apart; with the same radius describe arcs from these three points, cutting each other, and draw the two lines, *D E*, *F G*, through their intersections. The point *O*, where they cut, is the centre of the circle or arc.

To describe a circle passing through three given points.—Let *A*, *B*, *C* be the given points, and proceed as in last problem to find the centre *O*, from which the circle may be described.

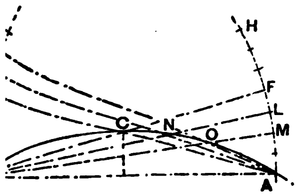


FIG. 14.

14. To describe an arc of a circle passing through three given points when the centre is not available (Fig. 14).—From the extreme points *A*, *B*, as centres, describe arcs *A H*, *B G*. Through the third point *C* draw *A E*, *B F*, cutting the arcs. Divide *A F* and *B E* into any number of equal parts, and set off a series of equal parts of the same length on the upper portions of the arcs beyond the points *E F*. Draw straight lines, *B L*, *B M*, etc., to the divisions in *A F*, and *A I*, *A K*, etc., to the divisions in *E G*. The successive intersections *N*, *O*, etc., of these li es are points in the circle required between the given points *A* and *C*, which may be drawn in; similarly the remaining part of the curve *B C* may be described. (See also Problem 54.)

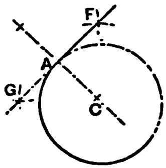


FIG. 15.

15. To draw a tangent to a circle from a given point in the circumference (Fig. 15).—Through the given point *A*, draw the radial line *A C*, and a perpendicular to it, *F G*, which is the tangent required.

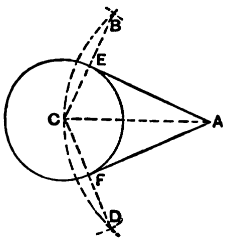


FIG. 16.

16. To draw tangents to a circle from a point without it (Fig. 16).—From *A*, with the radius *A C*, describe an arc *B C D*, and from *C*, with a radius equal to the diameter of the circle, cut the arc at *B D*. Join *B C*, *C D*, cutting the circle at *E F*, and draw *A E*, *A F*, the tangents.

NOTE.—When a tangent is already drawn, the exact point of contact may be found by drawing a perpendicular to it from the centre.

Between two inclined lines to draw a series of cir-
ching these lines and **touching each other** (Fig. 17)
he inclination of the given lines A B, *C D*, by the line *N O*. From
in this line draw the perpendicular *P B* to the line *A B*, an

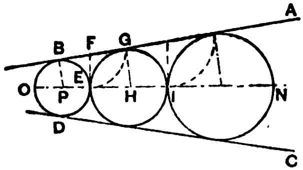


FIG. 17.

on *P* describe the circle *B D*, to the lines and cutting the cent at *E*. From *E* draw *E F* perpen to the centre line, cutting *A B* and from *F* describe an arc *E* ting *A B* at *G*. Draw *G H* para *B P*, giving *H*, the centre of th circle, to be described with the *H E*, and so on for the next circ
Inversely, the largest circle i described first, and the smalle in succession. This problem is quent use in scroll work.

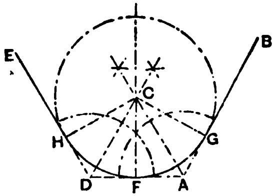


FIG. 18.

18. Between two inc lines to draw a circula ment tangent to the line passing through a poi on the line *F C* which bi the angle of the lines (F —Through *F* draw *D A* at right to *F C*; bisect the angles *A* an in Problem 11, by lines cuttin and from *C* with radius *C F* d arc *H F G* required.

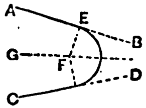


FIG 19.

19. To draw a circula that will be tangent to given lines *A B* and *C* clined to one another, tangential point *E* given (Fig. 19).—Draw the line *G F*. From *E* draw *E F* a to angles *A B*: then *F* is the of the circle required.

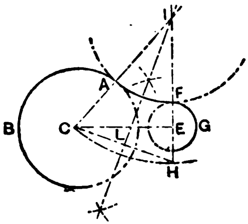


FIG. 20.

20. To describe a cir arc joining two circles touching one of them given point (Fig. 20).—To jo circles *A B*, *F G*, by an arc to one of them at *F*, draw the radii and produce it both ways. Set o equal to the radius *A C* of the circle; join *C H* and bisect it w perpendicul r *L I*, cutting *E*. On the centre *I*, with radius *I* scribe the arc *F A* as required.

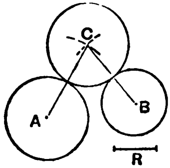


FIG. 21.

21. To draw a circle w given radius *R* that wi tangent to two given ci *A* and *B* (Fig. 21)—From of circle *A* with radius equal radius of *A*, and from centre of radius equal to *R* + radius of *B* two arcs cutting each other in *C* will be the centre of the cir quired.

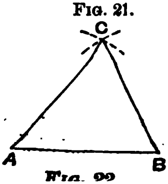


FIG. 22.

22. To construct an lateral triangle, the being given (Fig. 22).—On of one side, A, B, with A B a describe arcs cutting at C, A C, C B.

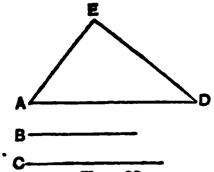


FIG. 23.

23. To construct a triangle of unequal sides (Fig. 23).—On either end of the base *A D*, with the side *B* as radius, describe an arc; and with the side *C* as radius, on the other end of the base as a centre, cut the arc at *E*. Join *A E*, *D E*.

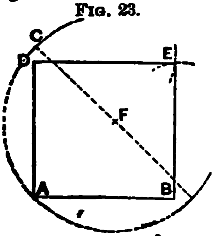


FIG. 24.

24. To construct a square on a given straight line *A B* (Fig. 24).—At *A* erect a perpendicular *A C*, as in Problem 4. Lay off *A D* equal to *A B*; from *D* and *B* as centres with radius equal *A B*, describe arcs cutting each other in *E*. Join *D E* and *B E*.

FIG. 25.

25. To construct a rectangle with given base *E F* **and height** *E H* (Fig. 25).—On the base *E F* draw the perpendiculars *E H*, *F G* equal to the height, and join *G H*.

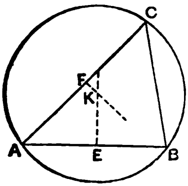


FIG. 26.

26. To describe a circle about a triangle (Fig. 26).—Bisect two sides *A B*, *A C* of the triangle at *E F*, and from these points draw perpendiculars cutting at *K*. On the centre *K*, with the radius *K A*, draw the circle *A B C*.

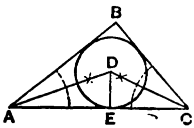


FIG. 27.

27. To inscribe a circle in a triangle (Fig 27).—Bisect two of the angles *A*, *C*, of the triangle by lines cutting at *D*; from *D* draw a perpendicular *D E* to any side, and with *D E* as radius describe a circle.

When the triangle is equilateral, draw a perpendicular from one of the angles to the opposite side, and from the side set off one third of the perpendicular.

28. To describe a circle about a square, and to inscribe a square in a circle (Fig. 28).—To describe the circle, draw the diagonals *A B*, *C D* of the square, cutting at *E*. On the centre *E*, with the radius *A E*, describe the circle.

To inscribe the square.—Draw the two diameters, *A B*, C D, a' right angles, and join the points *A*, B *C D*, to form the square.

NOTE.—In the same way a circle m be described about a rectangle.

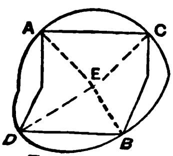


FIG. 28.

Fig. 29.

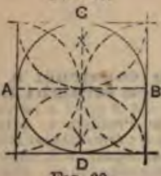


Fig. 30.

Fig. 31.

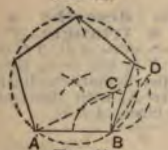


Fig. 32.

Fig. 33.

Fig. 34.

29. To inscribe a circle in a square (Fig. 29).—To inscribe the circle, draw the diagonals *A B*, *C D* of the square, cutting at *E*; draw the perpendicular *E F* to one side, and with the radius *E F* describe the circle.

30. To describe a square about a circle (Fig. 30).—Draw two diameters *A B*, *C D* at right angles. With the radius of the circle and *A*, *B*, *C* and *D* as centres, draw the four half circles which cross one another in the corners of the square.

31. To inscribe a pentagon in a circle (Fig. 31).—Draw diameters *A C*, *B D* at right angles, cutting at *o*. Bisect *A o* at *E*, and from *E*, with radius *E B*, cut *A C* at *F*; from *B*, with radius *B F*, cut the circumference at *G*, *H*, and with the same radius step round the circle to *I* and *K*; join the points so found to form the pentagon.

32. To construct a pentagon on a given line *A B* (Fig. 32).—From *B* erect a perpendicular *B C* half the length of *A B*; join *A C* and prolong it to *D*, making *C D* = *B C*. Then *B D* is the radius of the circle circumscribing the pentagon. From *A* and *B* as centres, with *B D* as radius, draw arcs cutting each other in *O*, which is the centre of the circle.

33. To construct a hexagon upon a given straight line (Fig. 33).—From *A* and *B*, the ends of the given line, with radius *A B*, describe arcs cutting at *g*; from *g*, with the radius *g A*, describe a circle; with the same radius set off the arcs *A G*, *G F*, and *B D*, *D E*. Join the points so found to form the hexagon. The side of a hexagon = radius of its circumscribed circle.

34. To inscribe a hexagon in a circle (Fig. 34).—Draw a diameter *A C B*. From *A* and *B* as centres, with the radius of the circle *A C*, cut the circumference at *D*, *E*, *F*, *G*, and draw *A D*, *D E*, etc., to form the hexagon. The radius of the circle is equal to the side of the hexagon; therefore the points *D*, *E*, etc., may also be found by stepping the radius six times round the circle. The angle between the diameter and the sides of a hexagon and also the exterior angle between a side and an adjacent side prolonged is 60 degrees; therefore a hexagon may conveniently be drawn by the use of a 60-degree triangle.

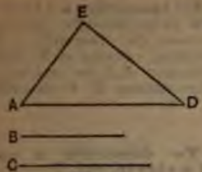


FIG. 23.

23. To construct a triangle of unequal sides (Fig. 23).—On either end of the base *A D*, with the side *B* as radius, describe an arc; and with the side *C* as radius, on the other end of the base as a centre, cut the arc at *E*. Join *A E*, *D E*.

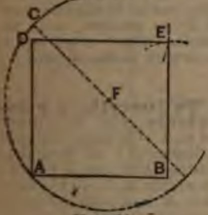


FIG. 24.

24. To construct a square on a given straight line *A B* (Fig. 24).—At *A* erect a perpendicular *A C*, as in Problem 4. Lay off *A D* equal to *A B*; from *D* and *B* as centres with radius equal *A B*, describe arcs cutting each other in *E*. Join *D E* and *B E*.

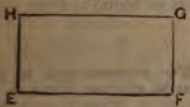


FIG. 25.

25. To construct a rectangle with given base *E F* and height *E H* (Fig. 25).—On the base *E F* draw the perpendiculars *E H*, *F G* equal to the height, and join *G H*.

FIG. 26.

26. To describe a circle about a triangle (Fig. 26).—Bisect two sides *A B*, *A C* of the triangle at *E F*, and from these points draw perpendiculars cutting at *K*. On the centre *K*, with the radius *K A*, draw the circle *A B C*.

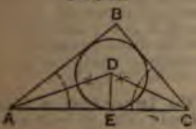


FIG. 27.

27. To inscribe a circle in a triangle (Fig. 27).—Bisect two of the angles *A*, *C*, of the triangle by lines cutting at *D*; from *D* draw a perpendicular *D E* to any side, and with *D E* as radius describe a circle.

When the triangle is equilateral, draw a perpendicular from one of the angles to the opposite side, and from the side set off one third of the perpendicular.

28. To describe a circle about a square, and to inscribe a square in a circle (Fig. 28).—To describe the circle, draw the diagonals *A B*, *C D* of the square, cutting at *E*. On the centre *E*, with the radius *A E*, describe the circle.

To inscribe the square.—Draw the two diameters, A B, C D, at right angles, and join the points A, B, C D, to form the square.

NOTE.—In the same way a circl[e] be described about a rectang[le]

FIG. 28.

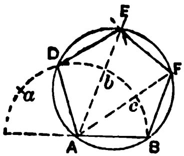


FIG. 40.

with the radius $A\ B$, describe a semicircle; divide the semi-circumference into as many equal parts as there are to be sides in the polygon—say, in this example, five sides. Draw lines from A through the divisional points D, b, and c, omitting one point a; and on the centres B, D, with the radius $A\ B$, cut $A\ b$ at E and $A\ c$ at F. Draw $D\ E$, $E\ F$, $F\ B$ to complete the polygon.

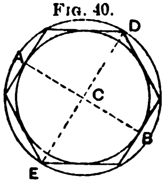


FIG. 41.

41. To inscribe a circle within a polygon (Figs. 41, 42).—When the polygon has an even number of sides (Fig. 41), bisect two opposite sides at A and B; draw $A\ B$, and bisect it at C by a diagonal $D\ E$, and with the radius $C\ A$ describe the circle.

When the number of sides is odd (Fig. 42), bisect two of the sides at A and B, and draw lines $A\ E$, $B\ D$ to the opposite angles, intersecting at C; from C, with the radius $C\ A$, describe the circle.

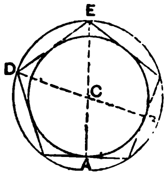


FIG. 42.

42. To describe a circle without a polygon (Figs. 41, 42).—Find the centre C as before, and with the radius $C\ D$ describe the circle.

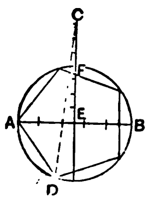


FIG. 43.

43. To inscribe a polygon of any number of sides within a circle (Fig. 43).—Draw the diameter $A\ B$ and through the centre E draw the perpendicular $E\ C$, cutting the circle at F. Divide $E\ F$ into four equal parts, and set off three parts equal to those from F to C. Divide the diameter $A\ B$ into as many equal parts as the polygon is to have sides; and from C draw $C\ D$, through the second point of division, cutting the circle at D. Then $A\ D$ is equal to one side of the polygon, and by stepping round the circumference with the length $A\ D$ the polygon may be completed.

TABLE OF POLYGONAL ANGLES.

Number of Sides.	Angle at Centre.	Number of Sides.	Angle at Centre.	Number of Sides.	Angle at Centre.
No.	Degrees.	No.	Degrees.	No.	Degrees.
8	120	9	40	15	24
4	90	10	36	16	$22\frac{1}{2}$
5	72	11	$32\frac{8}{11}$	17	$21\frac{3}{17}$
6	60	12	30	18	20
7	$51\frac{3}{7}$	13	$27\frac{9}{13}$	19	19
8	45	14	$25\frac{5}{7}$	20	18

Fig. 35.

35. To describe a hexagon about a circle (Fig. 35).—Draw a diameter $A\,D\,B$, and with the radius $A\,D$, on the centre A, cut the circumference at C; join $A\,C$, and bisect it with the radius $D\,E$; through E draw $F\,G$, parallel to $A\,C$, cutting the diameter at F, and with the radius $D\,F$ describe the circumscribing circle $F\,H$. Within this circle describe a hexagon by the preceding problem. A more convenient method is by use of a 60-degree triangle. Four of the sides make angles of 60 degrees with the diameter, and the other two are parallel to the diameter.

Fig. 36.

36. To describe an octagon on a given straight line (Fig. 36).—Produce the given line $A\,B$ both ways, and draw perpendiculars $A\,E$, $B\,F$; bisect the external angles A and B by the lines $A\,H$, $B\,C$, which make equal to $A\,B$. Draw $C\,D$ and $H\,G$ parallel to $A\,E$, and equal to $A\,B$; from the centres G, D, with the radius $A\,B$, cut the perpendiculars at E, F, and draw $E\,F$ to complete the octagon.

Fig. 37.

37. To convert a square into an octagon (Fig. 37).—Draw the diagonals of the square cutting at e; from the corners A, B, C, D, with $A\,e$ as radius, describe arcs cutting the sides at gn, fk, hm, and ol, and join the points so found to form the octagon. Adjacent sides of an octagon make an angle of 135 degrees.

Fig. 38.

38. To inscribe an octagon in a circle (Fig. 38).—Draw two diameters, $A\,C$, $B\,D$ at right angles; bisect the arcs $A\,B$, $B\,C$, etc., at $e\,f$, etc., and join $A\,e$, $e\,B$, etc., to form the octagon.

Fig. 39.

39. To describe an octagon about a circle (Fig. 39).—Describe a square about the given circle $A\,B$; draw perpendiculars $h\,k$, etc., to the diagonals, touching the circle to form the octagon.

. To describe a polygon of any number of sides upo
ven straight line (Fig. 40).—Produce the given line A B, and o

ints 1, 2, 3, etc. With the radius $A\ I$ on F and F' as centres, describe es, and with the radius $B\ I$ on the same centres cut these arcs as shown. Repeat the operation for the other divisions of the transverse axis. The series of intersections thus made are points in the curve, through which the curve may be traced.

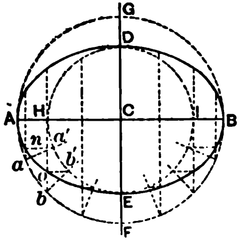


FIG. 48.

5th Method (Fig. 48).—On the two axes $A\ B$, $D\ E$ as diameters, on centre C, describe circles; from a number of points a, b, etc., in the circumference $A\ F\ B$, draw radii cutting the inner circle at a', b', etc. From a, b, etc., draw perpendiculars to $A\ B$; and from a', b', etc., draw parallels to $A\ B$, cutting the respective perpendiculars at n, o, etc. The intersections are points in the curve, through which the curve may be traced.

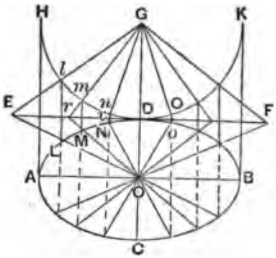


FIG 49.

6th Method (Fig. 49). — When the transverse and conjugate diameters are given, $A\ B$, $C\ D$, draw the tangent $E\ F$ parallel to $A\ B$. Produce $C\ D$, and on the centre G with the radius of half $A\ B$, describe a semicircle $H\ D\ K$; from the centre G draw any number of straight lines to the points E, r, etc., in the line $E\ F$, cutting the circumference at l, m, n, etc.; from the centre O of the ellipse draw straight lines to the points E, r, etc.; and from the points l, m, n, etc., draw parallels to $G\ C$, cutting the lines $O\ E$, $O\ r$, etc., at L, M, N, etc. These are points in the circumference of the ellipse, and the curve may be traced through them. Points in the other half of the ellipse are formed by extending the intersecting lines as indicated in the figure.

45. To describe an ellipse approximately by means of circular arcs.—*First.*—With arcs of two radii (Fig. 50).—Find the difference of the semi-axes, and set it off from the centre O to a and c on $O\ A$ and $O\ C$; draw $a\ c$, and set off half $a\ c$ to d; draw $d\ i$ parallel to $a\ c$; set off $O\ e$ equal to $O\ d$; join $e\ i$, and draw the parallels $e\ m$, $d\ m$. From m, with radius $m\ C$, describe an arc through C; and from i describe an arc through D; from d and e describe arcs through A and B. The four arcs form the ellipse approximately.

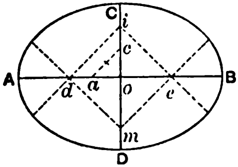


FIG. 50.

NOTE.—This method does not apply satisfactorily when the conjugate axis is less than two thirds of the transverse axis.

2d Method (by Carl G. Barth, Fig. 51).- In Fig. 51 $a\ b$ is the major and $c\ d$ the minor axis of the ellipse to be approximated. Lay off $b\ e$ equal to the semi-minor axis $c\ O$, and use $a\ e$ as radius for the arc at each extremity of the minor axis. Bisect $e\ o$ at f and lay off $e\ g$ equal to $e\ f$, and use $g\ b$ as radius for the arc at each extremity of the major axis.

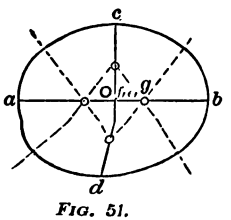


FIG. 51.

GEOMETRICAL PROPOSITIONS.

a right-angled triangle the square on the hypothenuse is equal to the of the squares on the other two sides.

triangle is equilateral, it is equiangular, and *vice versa*.

straight line from the vertex of an isosceles triangle bisects the base, ects the vertical angle and is perpendicular to the base.

ne side of a triangle is produced, the exterior angle is equal to the sum e two interior and opposite angles.

two triangles are mutually equiangular, they are similar and their sponding sides are proportional.

the sides of a polygon are produced in the same order, the sum of the rior angles equals four right angles.

a quadrilateral, the sum of the interior angles equals four right angles.

a parallelogram, the opposite sides are equal; the opposite angles equal; it is bisected by its diagonal; and its diagonals bisect each r.

three points are not in the same straight line, a circle may be passed ugh them.

two arcs are intercepted on the same circle, they are proportional to corresponding angles at the centre.

two arcs are similar, they are proportional to their radii.

e areas of two circles are proportional to the squares of their radii.

a radius is perpendicular to a chord, it bisects the chord and it bisects arc subtended by the chord.

straight line tangent to a circle meets it in only one point, and it is endicular to the radius drawn to that point.

from a point without a circle tangents are drawn to touch the circle, e are but two; they are equal, and they make equal angles with the d joining the tangent points.

two lines are parallel chords or a tangent and parallel chord, they cept equal arcs of a circle.

an angle at the circumference of a circle, between two chords, is subed by the same arc as an angle at the centre, between two radii, the e at the circumference is equal to half the angle at the centre.

a triangle is inscribed in a semicircle, it is right-angled.

an angle is formed by a tangent and chord, it is measured by one half e arc intercepted by the chord; that is, it is equal to half the angle at entre subtended by the chord.

two chords intersect each other in a circle, the rectangle of the segs of the one equals the rectangle of the segments of the other.

d if one chord is a diameter and the other perpendicular to it, the ngle of the segments of the diameter is equal to the square on half the chord, and the half chord is a mean proportional between the segs of the diameter.

MENSURATION.

PLANE SURFACES.

Quadrilateral.—A four-sided figure.

Parallelogram.—A quadrilateral with opposite sides parallel.

Varieties.—Square: four sides equal, all angles right angles. Rectangle: opposite sides equal, all angles right angles. Rhombus: four sides equal, opposite angles equal, angles not right angles. Rhomboid: opposite sides equal, opposite angles equal, angles not right angles.

Trapezium.—A quadrilateral with unequal sides.

Trapezoid.—A quadrilateral with only one pair of opposite sides parallel.

Diagonal of a square = $\sqrt{2 \times \text{side}^2} = 1.4142 \times$ side.

Diag. of a rectangle = $\sqrt{\text{sum of squares of two adjacent sides}}$.

Area of any parallelogram = base × altitude.

Area of rhombus or rhomboid = product of two adjacent sides × sine of angle included between them.

Area of a trapezium = half the product of the diagonal by the sum of the perpendiculars let fall on it from opposite angles.

Area of a trapezoid = product of half the sum of the two parallel sides by the perpendicular distance between them.

To find the area of any quadrilateral figure.—Divide the quadrilateral into two triangles; the sum of the areas of the triangles is the area.

Or, multiply half the product of the two diagonals by the sine of the angle at their intersection.

To find the area of a quadrilateral inscribed in a circle.—From half the sum of the four sides subtract each side severally; multiply the four remainders together; the square root of the product is the area.

Triangle.—A three-sided plane figure.

Varieties.—Right-angled, having one right angle; obtuse-angled, having one obtuse angle; isosceles, having two equal angles and two equal sides; equilateral, having three equal sides and equal angles.

The sum of the three angles of every triangle = 180°.

The two acute angles of a right-angled triangle are complements of each other.

Hypothenuse of a right-angled triangle, the side opposite the right angle = $\sqrt{\text{sum of the squares of the other two sides}}$.

To find the area of a triangle:

RULE 1. Multiply the base by half the altitude.

RULE 2. Multiply half the product of two sides by the sine of the included angle.

RULE 3. From half the sum of the three sides subtract each side severally; multiply together the half sum and the three remainders, and extract the square root of the product.

The area of an equilateral triangle is equal to one fourth the square of one of its sides multiplied by the square root of 3, $= \frac{a^2\sqrt{3}}{4}$, a being the side; or $a^2 \times .433013$.

Hypothenuse and one side of right-angled triangle given, to find other side. Required side $= \sqrt{\text{hyp}^2 - \text{given side}^2}$.

If the two sides are equal, side = hyp ÷ 1.4142; or hyp × .7071.

Area of a triangle given, to find base: Base = twice area ÷ perpendicular height.

Area of a triangle given, to find height: Height = twice area ÷ base.

Two sides and base given, to find perpendicular height (in a triangle in which both of the angles at the base are acute).

RULE.—As the base is to the sum of the sides, so is the difference of the sides to the difference of the divisions of the base made by drawing the perpendicular. Half this difference being added to or subtracted from half the base will give the two divisions thereof. As each side and its op-

ı of the base constitutes a right-angled triangle, the perpendicular is .ined by the rule perpendicular $= \sqrt{\text{hyp}^2 - \text{base}^2}$.

ygon. — A plane figure having three or more sides. Regular or ar, according as the sides or angles are equal or unequal. Polygons ned from the number of their sides and angles.

find the area of an irregular polygon.—Draw diagonals ıg the polygon into triangles, and find the sum of the areas of these es.

find the area of a regular polygon:

ᴇ.—Multiply the length of a side by the perpendicular distance to the ·; multiply the product by the number of sides, and divide it by 2. altiply half the perimeter by the perpendicular let fall from the centre : of the sides.

perpendicular from the centre is equal to half of one of the sides of olygon multiplied by the cotangent of the angle subtended by the half

ı angle at the centre = 360° divided by the number of sides.

TABLE OF REGULAR POLYGONS.

Name of Polygon.	Area, Side = 1.	Radius of Circumscribed Circle. Perpen. from Centre = 1.	Radius of Circumscribed Circle. Side = 1.	Radius of Inscribed Circle, Side = 1.	Length of Side, Radius of Circumsc. Circle = 1.	Angle at Centre.	Angle between Adjacent Sides.
Triangle	.4330127	2.	.5773	.2887	1.732	120°	60°
Square	1.	1.414	.7071	.5	1.4142	90	90
Pentagon	1.7204774	1.238	.8506	.6882	1.1756	72	108
Hexagon	2.5980762	1.156	1.	.866	1.	60	120
Heptagon	3.6339124	1.11	1.1524	1.0383	.8677	51 26′	128 4-7
Octagon	4.8284271	1.083	1.3066	1.2071	.7653	45	135
Nonagon	6.1818242	1.064	1.4619	1.3737	.684	40	140
Decagon	7.6942088	1.051	1.618	1.5388	.618	36	144
Undecagon	9.3656399	1.042	1.7747	1.7028	.5634	32 43′	147 3-11
Dodecagon	11.1961524	1.037	1.9319	1.866	.5176	30	150

o find the area of a regular polygon, when the length ı side only is given:

ʟᴇ.—Multiply the square of the side by the multiplier opposite to the e of the polygon in the table.

o find the area of an irular figure (Fig. 69).—Draw or.tes across its breadth at equal ınces apart, the first and the last nate each being one half space ı the ends of the figure. Find the ·age breadth by adding together lengths of these lines included be·ən the boundaries of the figure, divide by the number of the lines ·d; multiply this mean breadth by length. The greater *the number* nes *the nearer the approximation.*

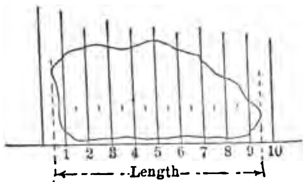


FIG. 69.

a figure of very irregular outline, as an indicator-diagram from a high-*steam-engine, mean lines may be* substituted for the actual lines of the *being so traced as to intersect the* undulations, so that the total area *spaces cut off may be compensated* by that of the extra spaces in-

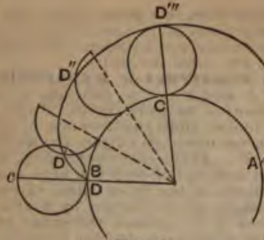


FIG. 61.

49. The Epicycloi generated by a point D i $D\,C$ rolling upon the circu another circle $A\ C\ B$, inst flat surface or line; the f the generating circle, an the fundamental circle. ing circle is shown in four which the generating poi sively marked D, D', D'', I is the epicycloid.

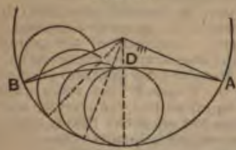


FIG. 62.

50. The Hypocycl is generated by a point i ating circle rolling on the fundamental circle.

When the generating ci of the other circle, the becomes a straight line.

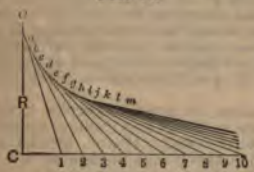


FIG. 63.

51. The Tract Schiele's anti-fricti (Fig. 63).—R is the radius C, 1, 2, etc., the axis. Fr on R a small distance, $o\,a$ R and centre a cut the a a 1, and set off a like sm $a\,b$; from b with radius I 2, join b 2, and so on. points o, a, b, c, d, etc., th the curve is to be drawn.

52. The Spiral.—The spiral is a curve described by a moves along a straight line according to any given law, the line time having a uniform angular motion. The line is called the ra

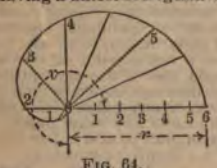


FIG. 64.

If the radius vector incre as the measuring angle, or parts described in each thus gradually increasin tance from each other, known as the spiral of (Fig. 64).

This curve is common cams. To describe it dra vector in several differen around the centre, with between them; set off the distances 1, 2, 3, 4, etc., corresponding upon which the curve is drawn, as shown in Fig. 64.

In the common spiral (Fig. 64) the pitch is uniform; that is, th equidistant. Such a spiral is made by rolling up a belt of unifor

To construct a sp four centres (Fig. 65) pitch of the spiral, constr about the centre, with th four sides equal to the pit the sides in one directio the corners are the cent arc of the external angle quadrant of a spire.

ind the diameter of a circle into which a certain of rings will fit on its inside (*Fig. 66*).—For instance, diameter of a circle into which twelve ½-inch rings will fit. as Assume that we have found the diameter of the required circle, and have drawn the rings inside of it. Join the centres of the rings by straight lines, as shown: we then obtain a regular polygon with 12 sides, each side being equal to the diameter of a given ring. We have now to find the diameter of a circle circumscribed about this polygon, and add the diameter of one ring to it; the sum will be the diameter of the circle into which the rings will fit. Through the centres A and D of two adjacent rings draw the radii CA and CD; since the polygon has twelve sides the angle $ACD = 30°$ and $ACB = 15°$. One half of the side AD is equal to AB. We now give the following proportion: The sine of the angle ACB is to AB as 1 is to the required radius. From this we get the following

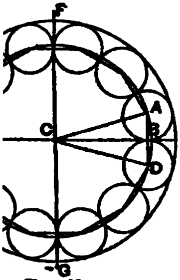


FIG. 66.

AB by the sine of the angle ACB; the quotient will be the circumscribed circle; add to the corresponding diameter the one ring; the sum will be the required diameter FG.

lescribe an arc of a circle which is too large to by a beam compass, by means of points in the as being given.—Suppose the radius is 20 feet and it is btain five points in an arc whose half chord is 4 feet. Draw a the half chord, full size, or on a smaller scale if more conerect a perpendicular at one end, thus making rectangular dinates. Erect perpendiculars at points 1, 2, 3, and 4 feet from pendicular. Find values of y in the formula of the circle. by substituting for x the values 0, 1, 2, 3, and 4, etc. and for R^2 of the radius, or 400. The values will be $y = \sqrt{R^2 - x^2} = \sqrt{400}$, $\sqrt{391}$. $\sqrt{384}$; = 20, 19.975, 19.90, 19.774, 19.596.

smallest,
ving 0.404, 0.379, 0.304, 0.178, 0 feet.

e distances on the five perpendiculars, as ordinates from the and the positions of five points on the arc will be found. Through these the c rve may be drawn. (See also Problem 14.)

55. The Catenary is the curve assumed by a perfectly flexible cord when its ends are fastened at two points, the weight of a unit length being constant.

The equation of the catenary is

$y = \frac{a}{2}\left(e^{\frac{x}{a}} + e^{-\frac{x}{a}}\right)$, in which e is the base of the Naperian system of logarithms.

To plot the catenary.—Let o (Fig. 67) be the origin of coördinates. Assigning to a any value as 3, the equation becomes

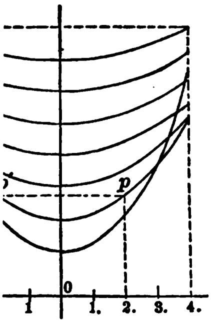


FIG. 67.

$$y = \frac{3}{2}\left(e^{\frac{x}{3}} + e^{-\frac{x}{3}}\right).$$

lowest point of the curve.

$$x = 0; \therefore y = \frac{3}{2}\left(e^{0} + e^{-0}\right) = \frac{3}{2}(1 + 1) = 3.$$

5. The Involute

Method of plotting angles without using

By means of

less than the requir

scale for each degr

GEOMETRICAL PROPOSITIONS.

ht-angled triangle the square on the hypothenuse is equal to the he squares on the other two sides.

angle is equilateral, it is equiangular, and *vice versa*.

aight line from the vertex of an isosceles triangle bisects the base, the vertical angle and is perpendicular to the base.

side of a triangle is produced, the exterior angle is equal to the sum o interior and opposite angles.

triangles are mutually equiangular, they are similar and their nding sides are proportional.

sides of a polygon are produced in the same order, the sum of the angles equals four right angles.

uadrilateral, the sum of the interior angles equals four right angles.

parallelogram, the opposite sides are equal; the opposite angles al; it is bisected by its diagonal; and its diagonals bisect each

ee points are not in the same straight line, a circle may be passed them.

arcs are intercepted on the same circle, they are proportional to esponding angles at the centre.

arcs are similar, they are proportional to their radii.

reas of two circles are proportional to the squares of their radii.

adius is perpendicular to a chord, it bisects the chord and it bisects subtended by the chord.

aight line tangent to a circle meets it in only one point, and it is licular to the radius drawn to that point.

m a point without a circle tangents are drawn to touch the circle, e but two; they are equal, and they make equal angles with the oining the tangent points.

o lines are parallel chords or a tangent and parallel chord, they pt equal arcs of a circle.

angle at the circumference of a circle, between two chords, is sub- by the same arc as an angle at the centre, between two radii, the the circumference is equal to half the angle at the centre.

riangle is inscribed in a semicircle, it is right-angled.

angle is formed by a tangent and chord, it is measured by one half arc intercepted by the chord; that is, it is equal to half the angle at tre subtended by the chord.

o chords intersect each other in a circle, the rectangle of the seg- of the one equals the rectangle of the segments of the other.

If one chord is a diameter and the other perpendicular to it, the le of the segments of the diameter is equal to the square on half the hord, and the half chord is a mean proportional between the seg- of the diameter.

Relations of Arc, Chord, Chord of Half the Ar Versed Sine, etc.

Let R = radius, D = diameter, Arc = length of arc,
Cd = chord of the arc, ch = chord of half the arc.
V = versed sine, $D - V$ = diam. minus ver. sin.,

$$Arc = \frac{8ch - Cd}{3} \text{ (very nearly)}, \quad = \frac{\sqrt{Cd^2 + 4V^2} \times 10V^2}{15Cd^2 + 33V^2} + 2ch, \text{ nea}$$

$$Arc = \frac{2ch \times 10V}{60D - 27V} + 2ch, \text{ nearly.}$$

Chord of the arc $= 2\sqrt{ch^2 - V^2}$; $= \sqrt{D^2 - (D - 2V)^2}$; $= 8ch - 3Arc$.
$= 2\sqrt{R^2 - (R - V)^2}$; $= 2\sqrt{(D - V) \times V}$.

Chord of half the arc, $ch = \frac{1}{2}\sqrt{Cd^2 + 4V^2}$; $= \sqrt{D \times V}$; $= \frac{3Arc + Cd}{8}$.

Diameter $= \frac{ch^2}{V}$; $= \frac{\left(\frac{1}{2}Cd\right)^2 + V^2}{V}$;

Versed sine $= \frac{ch^2}{D}$; $= \frac{1}{2}(D - \sqrt{D^2 - Cd^2})$

(or $\frac{1}{2}(D + \sqrt{D^2 - Cd^2})$, if V is greater than

$= \sqrt{ch^2 - \frac{Cd^2}{4}}$.

Half the chord of the arc is a mean proportional between the ver and diameter minus versed sine:

$$\frac{1}{2}Cd = \sqrt{V \times (D - V)}.$$

Length of a Circular Arc.—Huyghens's Approxim

Let C represent the length of the chord of the arc and c the lengt chord of half the arc; the length of the arc

$$L = \frac{8c - C}{3}.$$

Professor Williamson shows that when the arc subtends an angle of radius being 100.000 feet (nearly 19 miles), the error by this formula two inches, or 1/600000 part of the radius. When the length of th equal to the radius, i.e., when it subtends an angle of 57°.3, the err than 1/7680 part of the radius. Therefore, if the radius is 100.000 f error is less than $\frac{100000}{7680} = 13$ feet. The error increases rapidly v increase of the angle subtended.

In the measurement of an arc which is described with a short ra error is so small that it may be neglected. Describing an arc with of 12 inches subtending an angle of 30°, the error is 1/50000 of an in 57°.3 the error is less than 0″.0015.

In order to measure an arc when it subtends a large angle, bise measure each half as before—in this case making B = length of the half the arc, and b = length of the chord of one fourth the arc; the

$$L = \frac{16b - 2B}{3}.$$

Relation of the Circle to its Equal, Inscribed, an cumscribed Squares.

Diameter of circle × .88623 } = side of equal square.
Circle × .28209 }
cle × 1.1284 = perimeter of equal squa

Diameter of circle ×	.7071	= side of inscribed square.
Circumference of circle ×	.22508	= side of inscribed square.
Area of circle × .90031 ÷ diameter		= side of inscribed square.
Area of circle ×	1.2732	= area of circumscribed square.
Area of circle ×	.63662	= area of inscribed square.
Side of square ×	1.4142	= diam. of circumscribed circle.
" " ×	4.4428	= circum. " " "
" " ×	1.1284	= diam. of equal circle.
" " ×	3.5449	= circum. " "
Perimeter of square ×	0.88623	= " " "
Square inches ×	1.2732	= circular inches.

Sectors and Segments.—*To find the area of a sector of a circle.* Rule 1. Multiply the arc of the sector by half its radius.

Rule 2. As 360 is to the number of degrees in the arc, so is the area of circle to the area of the sector.

Rule 3. Multiply the number of degrees in the arc by the square of the radius and by .008727.

To find the area of a segment of a circle: Find the area of the sector which has the same arc, and also the area of the triangle formed by the chord of the segment and the radii of the sector.

Then take the sum of these areas, if the segment is greater than a semicircle, but take their difference if it is less.

Another Method: Area of segment $= \frac{R^2}{2}$ (arc − sin A) in which A is the central angle, R the radius, and arc the length of arc to radius 1.

To find the area of a segment of a circle when its chord and height or versed sine only are given. First find radius, as follows:

$$\text{radius} = \frac{1}{2}\left[\frac{\text{square of half the chord}}{\text{height}} + \text{height}\right].$$

2. Find the angle subtended by the arc, as follows: $\frac{\text{half chord}}{\text{radius}}$ = sine of half the angle. Take the corresponding angle from a table of sines, and double it to get the angle of the arc.

3. Find area of the sector of which the segment is a part;

$$\text{area of sector} = \text{area of circle} \times \frac{\text{degrees of arc}}{360}.$$

4. Subtract area of triangle under the segment:

$$\text{Area of triangle} = \frac{\text{chord}}{2} \times (\text{radius} - \text{height of segment}).$$

The remainder is the area of the segment.

When the chord, arc, and diameter are given, to find the area. From the length of the arc subtract the length of the chord. Multiply the remainder by the radius or one-half diameter; to the product add the chord multiplied by the height, and divide the sum by 2.

Another rule: Multiply the chord by the height and this product by .6834 plus one tenth of the square of the height divided by the radius.

To find the chord: From the diameter subtract the height; multiply the remainder by four times the height and extract the square root.

When the chords of the arc and of half the arc and the versed sine are given: To the chord of the arc add four thirds of the chord of half the arc; multiply the sum by the versed sine and the product by .40426 (approximate).

Circular Ring.—*To find the area of a ring included between the circumferences of two concentric circles:* Take the difference between the areas of the two circles; or, subtract the square of the less radius from the square of the greater, and multiply their difference by 3.14159.

The area of the greater circle is equal to πR^2;
and the area of the smaller, πr^2.

Their difference, or the area of the ring, is $\pi(R^2 - r^2)$.

The Ellipse.—Area of an ellipse = product of its semi-axes × 3.14159
= product of its axes × .785398.

The Ellipse.—Circumference (approximate) $= 3.1416\sqrt{\frac{D^2 + d^2}{2}}$, D and d being the two axes.

Trautwine gives the following as more accurate: When the longer axis D is not more than five times the length of the shorter axis, d,

$$\text{Circumference} = 3.1416\sqrt{\frac{D^2+d^2}{2} - \frac{(D-d)^2}{8.8}}.$$

When D is more than $5d$, the divisor 8.8 is to be replaced by the fo
divisors:

$\frac{D}{d}$	= 6,	7,	8,	9,	10,	12,	14,	16,	18,	20,	30,	4
Divisor =	9,	9.2,	9.3,	9.35,	9.4,	9.5,	9.6,	9.68,	9.75,	9.8,	9.92,	9.

Reuleaux gives: Circumference $= \pi(a+b)\left(1+\frac{n^2}{4}+\frac{n^4}{64}+\frac{n^6}{256}+\right.$

which $n = \frac{a-b}{a+b}$, a and b being the semi-axes.

Area of a segment of an ellipse the base of which is parallel to the axes of the ellipse. Divide the height of the segment by the which it is part, and find the area of a circular segment, in a table o lar segments, of which the height is equal to the quotient; multiply t thus found by the product of the two axes of the ellipse.

Cycloid.—A curve generated by the rolling of a circle on a plane

Length of a cycloidal curve = 4 × diameter of the generating cir
Length of the base = circumference of the generating c
Area of a cycloid = 3 × area of generating circle.

Helix (Screw).—A line generated by the progressive rotatio point around an axis and equidistant from its centre.

Length of a helix.—To the square of the circumference described generating-point add the square of the distance advanced in one rev and take the square root of their sum multiplied by the number of tions of the generating point. Or,

$$\sqrt{(c^2+h^2)}n = \text{length}, \ n \text{ being number of revolutions.}$$

Spirals.—Lines generated by the progressive rotation of a point a fixed axis, with a constantly increasing distance from the axis.

A *plane spiral* is when the point rotates in one plane.

A *conical spiral* is when the point rotates around an axis at a prog distance from its centre, and advancing in the direction of the axis, as a cone.

Length of a plane spiral line.—When the distance between the uniform.

RULE.—Add together the greater and less diameters; divide their 2; multiply the quotient by 3.1416, and again by the number of revol Or, take the mean of the length of the greater and less circumferen multiply it by the number of revolutions. Or,

$$\text{length} = \pi n\frac{d+d'}{2}, \ d \text{ and } d' \text{ being the inner and outer diamet}$$

Length of a conical spiral line.—Add together the greater and les eters; divide their sum by 2 and multiply the quotient by 3.1416. square of the product of this circumference and the number of revo of the spiral add the square of the height of its axis and take the root of the sum.

$$\text{Or, length} = \sqrt{\left(\pi n\frac{d+d'}{2}\right)^2 + h^2}.$$

SOLID BODIES.

The Prism.—*To find the surface of a right prism*: Multiply the eter of the base by the altitude for the convex surface. To this a areas of the two ends when the entire surface is required.

Volume of a prism = area of its base × its altitude.

The pyramid.—Convex surface of a regular pyramid = p *its base × half the* slant height. To this add area of the base *surface is required.*

Vol amid = area of base × one third of the a *d the surface of a frustum of a regular pyramid:* Multiply half the ight by the sum of the perimeters of the two bases for the convex To this add the areas of the two bases when the entire surface is .

l the volume of a frustum of a pyramid: Add together the areas of bases and a mean proportional between them, and multiply the one third of the altitude. (Mean proportional between two numbers root of their product.)

ge.—A wedge is a solid bounded by five planes, viz.: a rectangular o trapezoids, or two rectangles, meeting in an edge, and two triends. The altitude is the perpendicular drawn from any point in to the plane of the base.

l the volume of a wedge: Add the length of the edge to twice the the base, and multiply the sum by one sixth of the product of the the wedge and the breadth of the base.

angular prismoid.—A rectangular prismoid is a solid bounded lanes, of which the two bases are rectangles, having their corresides parallel, and the four upright sides of the solids are trape-

l the volume of a rectangular prismoid: Add together the areas of bases and four times the area of a parallel section equally distant bases, and multiply the sum by one sixth of the altitude.

der.—Convex surface of a cylinder = perimeter of base × altitude. dd the areas of the two ends when the entire surface is required.

Volume of a cylinder = area of base × altitude.

.—Convex surface of a cone = circumference of base × half the slant this add the area of the base when the entire surface is required.

$$\text{Volume of a cone} = \text{area of base} \times \frac{1}{3}\text{ altitude.}$$

l the surface of a frustum of a cone: Multiply half the side by the e circumferences of the two bases for the convex surface; to this reas of the two bases when the entire surface is required.

the volume of a frustum of a cone: Add together the areas of the s and a mean proportional between them, and multiply the sum ird of the altitude.

re.—*To find the surface of a sphere:* Multiply the diameter by the rence of a great circle; or, multiply the square of the diameter by

ce of sphere = 4 × area of its great circle.
" " = convex surface of its circumscribing cylinder.

es of spheres are to each other as the squares of their diameters.

the volume of a sphere: Multiply the surface by one third of the r. multiply the cube of the diameter by $1/6\pi$; that is, by 0.5236.

of $\frac{1}{6}\pi$ to 10 decimal places = .5235987756.

lume of a sphere = 2/3 the volume of its circumscribing cylinder.

es of spheres are to each other as the cubes of their diameters.

rical triangle.—*To find the area of a spherical triangle:* Comsurface of the quadrantal triangle, or one eighth of the surface of re. From the sum of the three angles subtract two right angles; e remainder by 90, and multiply the quotient by the area of the tal triangle.

rical polygon.—*To find the area of a spherical polygon:* Comsurface of the quadrantal triangle. From the sum of all the angles the product of two right angles by the number of sides less two; e remainder by 90 and multiply the quotient by the area of the tal triangle.

prismoid.—The prismoid is a solid having parallel end areas, and composed of any combination of prisms, cylinders, wedges, pyracones or frustums of the same, whose bases and apices lie in the s.

uch as cylinders and cones are but special forms of prisms and s, and warped surface solids may be divided into elementary forms and since frustums may also be subdivided into the elementary is sufficient to say that all prismoids may be decomposed into dges, and pyramids. If a formula can be found which is equally o all of these forms, then it will apply to any combination of s formula is called

The Prismoidal Formula.

Let A = area of the base of a prism, wedge, or pyramid;
A_1, A_2, Am = the two end and the middle areas of a prismoid, or o
its elementary solids;

h = altitude of the prismoid or elementary solid;
V = its volume;

$$V = \frac{h}{6}(A_1 + 4A_m + A_2).$$

For a prism A_1, Am and A_2 are equal, $= A$; $V = \frac{h}{6} \times 6A = hA$.

For a wedge with parallel ends, $A_2 = 0$, $Am = \frac{1}{2}A_1$; $V = \frac{h}{6}(A_1 + 2A$

For a cone or pyramid, $A_2 = 0$, $Am = \frac{1}{4}A_1$; $V = \frac{h}{6}(A_1 + A_1) = \frac{hA}{3}$

The prismoidal formula is a rigid formula for all prismoids. T
approximation involved in its use is in the assumption that the gi
may be generated by a right line moving over the boundaries of
areas.

The area of the middle section is never the mean of the two end
the prismoid contains any pyramids or cones among its elementar
When the three sections are similar in form the *dimensions* of th
area are always the means of the corresponding end dimensions.
often enables the dimensions, and hence the area of the middle se
be computed from the end areas.

Polyedrons.—A polyedron is a solid bounded by plane poly
regular polyedron is one whose sides are all equal regular polygon

To find the surface of a regular polyedron.—Multiply the area o
the faces by the number of faces; or, multiply the square of on
edges by the surface of a similar solid whose edge is unity.

A TABLE OF THE REGULAR POLYEDRONS WHOSE EDGES ARE UN

Names.	No. of Faces.	Surface.
Tetraedron	4	1.7320508
Hexaedron	6	6.0000000
Octaedron	8	3.4641016
Dodecaedron	12	20.6457288
Icosaedron	20	8.6602540

To find the volume of a regular polyedron.—Mul
surface by one third of the perpendicular let fall from the centre o
the faces; or, multiply the cube of one of the edges by the soli
similar polyedron whose edge is unity.

Solid of revolution.—The volume of any solid of revo
equal to the product of the area of its generating surface by the
the path of the centre of gravity of that surface.

The convex surface of any solid of revolution is equal to the p
the perimeter of its generating surface by the length of path of i
of gravity.

Cylindrical ring.—Let d = outer diameter; d' = inner d
$\frac{1}{2}(d - d')$ = thickness $= t$; $\frac{1}{4}\pi t^2$ = sectional area; $\frac{1}{2}(d + d')$ = me
eter $= M$; πt = circumference of section; πM = mean circumfe
ring; surface $= \pi t \times \pi M$; $= \frac{1}{4}\pi^2(d^2 - d'^2)$; $= 9.86965\, t\, M$; $= 2.46741\,($

volume $= \frac{1}{4}\pi t^2 M \pi$; $= 2.46741 t^2 M$.

Spherical zone.—*Surface of a spherical zone* or segment of
= its altitude × the circumference of a great circle of the sphere.
circle is one whose plane passes through the centre of the sphere.

Volume of a zone of a sphere.—To the sum of the squares of
the ends add one third of the square of the height; multiply
height and by 1.5708.

Spherical segment.—*Volume of a spherical segment with* the height of the segment by the area of the base, and the ight by .5236 and add the two products. Or, from three times of the sphere subtract twice the height of the segment; multi- ence by the square of the height and by .5236. Or, to three are of the radius of the base of the segment add the square of l multiply the sum by the height and by .5236.

or ellipsoid.—When the revolution of the spheroid is about e diameter it is *prolate*, and when about the conjugate it is

face of a segment of a spheroid.—Square the diameters of the take the square root of half their sum ; then, as the diameter the segment is cut is to this root so is the height of the e proportionate height of the segment to the mean diameter. product of the other diameter and 3.1416 by the proportionate

face of a frustum or zone of a spheroid.—Proceed as by for the surface of a segment, and obtain the proportionate frustum. Multiply the product of the diameter parallel to the ustum and 3.1416 by the proportionate height of the frustum.

spheroid is equal to the product of the square of the revolving ed axis and by .5236. The volume of a spheroid is two thirds circumscribing cylinder.

a segment of a spheroid.—1. When the base is parallel to the s, multiply the difference between three times the fixed axis height of the segment, by the square of the height and by ly the product by the square of the revolving axis, and divide of the fixed axis.

e base is perpendicular to the revolving axis, multiply the tween three times the revolving axis and twice the height of by the square of the height and by .5236. Multiply the e length of the fixed axis, and divide by the length of the s.

the middle frustum of a spheroid.—1. When the ends are parallel to the revolving axis: To twice the square of the ter add the square of the diameter of one end ; multiply the ngth of the frustum and by .2618.

e ends are elliptical, or perpendicular to the revolving axis: product of the transverse and conjugate diameters of the n add the product of the transverse and conjugate diameters nultiply the sum by the length of the frustum and by .2618.

.—Figures generated by the revolution of a plane area, when evolved about a chord perpendicular to its axis, or about its ate. They are designated by the name of the arc or curve ey are generated, as Circular, Elliptic, Parabolic, etc., etc.

face of a circular spindle, zone, or segment of it.—Rule: Mul- h by the radius of the revolving arc; multiply this arc by the ce, or distance between the centre of the spindle and centre ing arc ; subtract this product from the former, double the d multiply it by 3.1416

circular spindle.—Multiply the central distance by half the volving segment; subtract the product from one third of the he length, and multiply the remainder by 12 5664.

frustum or zone of a circular spindle.—From the square of h of the whole spindle take one third of the square of half the frustum, and multiply the remainder by the said half length n ; multiply the central distance by the revolving area which frustum ; subtract this product from the former, and multi- nder by 6 2832.

a segment of a circular spindle.—Subtract the length of the the half length of the spindle ; double the remainder and volume of a middle frustum of this length ; subtract the e volume of the whole spindle and halve the remainder.

cycloidal spindle = five eighths of the volume of the circum- *er.—Multiply the product of the* square of twice the diameter *g circle and 3.927 by its circumference*, and divide this pro-

onoid.—*Volume of a parabolic conoid* (generated by the *rabola on its axis).—Multiply* the area of the base by half

The tangent of the supplement is equal to the tangent of the arc, b a contrary sign. Tang. $B\,D\,F = B\,M$.

The secant of the supplement is equal to the secant of the arc, but contrary sign. Sec. $B\,D\,F = C\,M$.

Signs of the functions in the four quadrants.— divide a circle into four quadrants by a vertical and a horizontal ter, the upper right-hand quadrant is called the first, the upper left t ond, the lower left the third, and the lower right the fourth. The si the functions in the four quadrants are as follows:

	First quad.	*Second* quad.	*Third* quad.	*Fourth*
Sine and cosecant,	+	+	−	
Cosine and secant,	+	−	−	
Tangent and cotangent,	+	−	+	

The values of the functions are as follows for the angles specified:

Angle	$0°$	$30°$	$45°$	$60°$	$90°$	$120°$	$135°$	$150°$	$180°$
Sine	0	$\frac{1}{2}$	$\frac{1}{\sqrt{2}}$	$\frac{\sqrt{3}}{2}$	1	$\frac{\sqrt{3}}{2}$	$\frac{1}{\sqrt{2}}$	$\frac{1}{2}$	0
Cosine	1	$\frac{\sqrt{3}}{2}$	$\frac{1}{\sqrt{2}}$	$\frac{1}{2}$	0	$-\frac{1}{2}$	$-\frac{1}{\sqrt{2}}$	$-\frac{\sqrt{3}}{2}$	-1
Tangent	0	$\frac{1}{\sqrt{3}}$	1	$\sqrt{3}$	∞	$-\sqrt{3}$	-1	$-\frac{1}{\sqrt{3}}$	0
Cotangent	∞	$\sqrt{3}$	1	$\frac{1}{\sqrt{3}}$	0	$-\frac{1}{\sqrt{3}}$	-1	$-\sqrt{3}$	∞
Secant	1	$\frac{2}{\sqrt{3}}$	$\sqrt{2}$	2	∞	-2	$-\sqrt{2}$	$-\frac{2}{\sqrt{3}}$	-1
Cosecant	∞	2	$\sqrt{2}$	$\frac{2}{\sqrt{3}}$	1	$\frac{2}{\sqrt{3}}$	$\sqrt{2}$	2	∞
Versed sine	0	$\frac{2-\sqrt{3}}{2}$	$\frac{\sqrt{2}-1}{\sqrt{2}}$	$\frac{1}{2}$	1	$\frac{3}{2}$	$\frac{\sqrt{2}+1}{\sqrt{2}}$	$\frac{2+\sqrt{3}}{2}$	2

TRIGONOMETRICAL FORMULÆ.

The following relations are deduced from the properties of sim angles (Radius = 1):

$$\cos A : \sin A :: 1 : \tan A, \text{ whence } \tan A = \frac{\sin A}{\cos A};$$

$$\sin A : \cos A :: 1 : \cot A, \quad \text{"} \quad \text{cotan } A = \frac{\cos A}{\sin A};$$

$$\cos A : 1 \quad :: 1 : \sec A, \quad \text{"} \quad \sec A = \frac{1}{\cos A};$$

$$\sin A : 1 \quad :: 1 : \text{cosec } A, \quad \text{"} \quad \text{cosec } A = \frac{1}{\sin A};$$

$$\tan A : 1 \quad :: 1 : \cot A \quad \text{"} \quad \tan A = \frac{1}{\cot A}.$$

The sum of the square of the sine of an arc and the square of its equals unity. $\text{Sin}^2 A + \cos^2 A = 1$.

Formulæ for the functions of the sum and differe two angles:

Let the two angles be denoted by A and B, their sum $A + B = C$, a *difference* $A - B$ by D.

$$+ B) = \sin A \cos B + \cos A \sin B; \ldots$$

$$\cos(A+B) = \cos A \cos B - \sin A \sin B; \quad . \; . \; . \; . \; . \; . \quad (2)$$
$$\sin(A-B) = \sin A \cos B - \cos A \sin B; \quad . \; . \; . \; . \; . \; . \quad (3)$$
$$\cos(A-B) = \cos A \cos B + \sin A \sin B. \quad . \; . \; . \; . \; . \; . \quad (4)$$

om these four formulæ by addition and subtraction we obtain

$$\sin(A+B) + \sin(A-B) = 2 \sin A \cos B; \quad . \; . \; . \; . \; . \; . \quad (5)$$
$$\sin(A+B) - \sin(A-B) = 2 \cos A \sin B; \quad . \; . \; . \; . \; . \; . \quad (6)$$
$$\cos(A+B) + \cos(A-B) = 2 \cos A \cos B; \quad . \; . \; . \; . \; . \; . \quad (7)$$
$$\cos(A-B) - \cos(A+B) = 2 \sin A \sin B. \quad . \; . \; . \; . \; . \; . \quad (8)$$

f we put $A + B = C$, and $A - B = D$, then $A = \frac{1}{2}(C + D)$ and $B = \frac{1}{2}(C -$ and we have

$$\sin C + \sin D = 2 \sin \tfrac{1}{2}(C+D) \cos \tfrac{1}{2}(C-D); \quad . \; . \; . \; . \quad (9)$$
$$\sin C - \sin D = 2 \cos \tfrac{1}{2}(C+D) \sin \tfrac{1}{2}(C-D); \quad . \; . \; . \; . \quad (10)$$
$$\cos C + \cos D = 2 \cos \tfrac{1}{2}(C+D) \cos \tfrac{1}{2}(C-D); \quad . \; . \; . \; . \quad (11)$$
$$\cos D - \cos C = 2 \sin \tfrac{1}{2}(C+D) \sin \tfrac{1}{2}(C-D). \quad . \; . \; . \; . \; . \quad (12)$$

Equation (9) may be enunciated thus: The sum of the sines of any two angles is equal to twice the sine of half the sum of the angles multiplied by the cosine of half their difference. These formulæ enable us to transform sum or difference into a product.

The sum of the sines of two angles is to their difference as the tangent of half the sum of those angles is to the tangent of half their difference.

$$\frac{\sin A + \sin B}{\sin A - \sin B} = \frac{2 \sin \frac{1}{2}(A+B) \cos \frac{1}{2}(A-B)}{2 \cos \frac{1}{2}(A+B) \sin \frac{1}{2}(A-B)} = \frac{\tan \frac{1}{2}(A+B)}{\tan \frac{1}{2}(A-B)}. \qquad (13)$$

The sum of the cosines of two angles is to their difference as the cotangent of half the sum of those angles is to the tangent of half their difference.

$$\frac{\cos A + \cos B}{\cos B - \cos A} = \frac{2 \cos \frac{1}{2}(A+B) \cos \frac{1}{2}(A-B)}{2 \sin \frac{1}{2}(A+B) \sin \frac{1}{2}(A-B)} = \frac{\cot \frac{1}{2}(A+B)}{\tan \frac{1}{2}(A-B)}. \; . \; . \qquad (14)$$

The sine of the sum of two angles is to the sine of their difference as the sum of the tangents of those angles is to the difference of the tangents.

$$\frac{\sin(A+B)}{\sin(A-B)} = \frac{\tan A + \tan B}{\tan A - \tan B}; \quad . \; . \; . \; . \; . \; . \; . \; . \quad (15)$$

$$\frac{\sin(A+B)}{\cos A \cos B} = \tan A + \tan B;$$
$$\frac{\sin(A-B)}{\cos A \cos B} = \tan A - \tan B;$$
$$\frac{\cos(A+B)}{\cos A \cos B} = 1 - \tan A \tan B;$$
$$\frac{\cos(A-B)}{\cos A \cos B} = 1 + \tan A \tan B;$$

$$\tan(A+B) = \frac{\tan A + \tan B}{1 - \tan A \tan B};$$
$$\tan(A-B) = \frac{\tan A - \tan B}{1 + \tan A \tan B};$$
$$\cot(A+B) = \frac{\cot A \cot B - 1}{\cot B + \cot A};$$
$$\cot(A-B) = \frac{\cot A \cot B + 1}{\cot B - \cot A}.$$

Solution of Plane Right-angled Triangles.

Let A and B be the two acute angles and C the right angle, and a, b, a c the sides opposite these angles, respectively, then we have

$$1.\ \sin A = \cos B = \frac{a}{c}; \quad 3.\ \tan A = \cot B = \frac{a}{b};$$

$$2.\ \cos A = \sin B = \frac{b}{c}; \quad 4.\ \cot A = \tan B = \frac{b}{a}.$$

1. In any plane right-angled triangle the sine of either of the acute an is equal to the quotient of the opposite leg divided by the hypothenuse.
2. The cosine of either of the acute angles is equal to the quotient of adjacent leg divided by the hypothenuse.
3. The tangent of either of the acute angles is equal to the quotient of opposite leg divided by the adjacent leg.
4. The cotangent of either of the acute angles is equal to the quotien the adjacent leg divided by the opposite leg.
5. The square of the hypothenuse equals the sum of the squares of other two sides.

Solution of Oblique-angled Triangles.

The following propositions are proved in works on plane trigonometry. any plane triangle—

Theorem 1. The sines of the angles are proportional to the opposite si

Theorem 2. The sum of any two sides is to their difference as the tan of half the sum of the opposite angles is to the tangent of half their di ence.

Theorem 3. If from any angle of a triangle a perpendicular be draw the opposite side or base, the whole base will be to the sum of the other sides as the difference of those two sides is to the difference of the segm of the base.

CASE I. Given two angles and a side, to find the third angle and the ot two sides. 1. The third angle = 180° − sum of the two angles. 2. The s may be found by the following proportion:

The sine of the angle opposite the given side is to the sine of the angle posite the required side as the given side is to the required side.

CASE II. Given two sides and an angle opposite one of them, to find third side and the remaining angles.

The side opposite the given angle is to the side opposite the required an as the sine of the given angle is to the sine of the required angle.

The third angle is found by subtracting the sum of the other two from 1 and the third side is found as in Case I.

CASE III. Given two sides and the included angle, to find the third side the remaining angles.

The sum of the required angles is found by subtracting the given an from 180°. The difference of the required angles is then found by Theor II. Half the difference added to half the sum gives the greater angle, half the difference subtracted from half the sum gives the less angle. third side is then found by Theorem I.

Another method:

Given the sides c, b, and the included angle A, to find the remaining si and the remaining angles B and C.

From either of the unknown angles, as B, draw a perpendicular Be to opposite side.

Then

$$Ae = c \cos A, \quad Be = c \sin A, \quad eC = b - Ae, \quad Be \div eC = \tan C.$$

Or, in other words, solve Be, Ae and BeC as right-angled triangles.

CASE IV. Given the three sides, to find the angles.

Let fall a perpendicular upon the longest side from the opposite an dividing the given triangle into two right-angled triangles. The two s ments of the base may be found by Theorem III. There will then be gi the hypothenuse and one side of a right-angled triangle, to find the angle

For areas of triangles, see Mensuration.

ANALYTICAL GEOMETRY.

alytical geometry is that branch of Mathematics which has for ect the determination of the forms and magnitudes of geometrical tudes by means of analysis.

linates and abscissas.—In analytical geometry two intersecting lines YY', XX' are used as *coördinate axes*, XX' being the axis of abscissas or axis of X, and YY' the axis of ordinates or axis of Y. A, the intersection, is called the origin of coördinates. The distance of any point P from the axis of Y measured parallel to the axis of X is called the *abscissa* of the point, as AD or CP, Fig. 71. Its distance from the axis of X, measured parallel to the axis of Y, is called the *ordinate*, as AC or PD. The abscissa and ordinate taken together are called the coördinates of the point P. The angle of intersection is usually taken as a right angle, in which case the axes of X and Y are called *rectangular coördinates*.

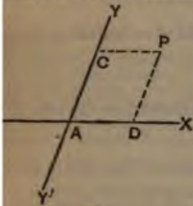


FIG. 71.

abscissa of a point is designated by the letter x and the ordinate by y. *equations* of a point are the equations which express the distances of oint from the axis. Thus $x = a$, $y = b$ are the equations of the point P.

uations referred to rectangular coördinates.—The equa- f a line expresses the relation which exists between the coördinates of point of the line.

ation of a straight line, $y = ax \pm b$, in which a is the tangent of the the line makes with the axis of X, and b the distance above A in which ne cuts the axis of Y.

ry equation of the first degree between two variables is the equation of ight line, as $Ay + Bx + C = 0$, which can be reduced to the form $y =$ b.

ation of the distance between two points:

$$D = \sqrt{(x'' - x')^2 + (y'' - y')^2},$$

ich $x'y'$, $x''y''$ are the coördinates of the two points.

ation of a line passing through a given point:

$$y - y' = a(x - x'),$$

ich $x'y'$ are the coördinates of the given point, a, the tangent of the the line makes with the axis of x, being undetermined, since any num- lines may be drawn through a given point.

ation of a line passing through two given points:

$$y - y' = \frac{y'' - y'}{x'' - x'}(x - x').$$

ation of a line parallel to a given line and through a given point:

$$y - y' = a(x - x').$$

ation of an angle V included between two given lines:

$$\text{tang } V = \frac{a' - a}{1 + a'a},$$

ch a and a' are the tangents of the angles the lines make with the f abscissas.

e lines are at right angles to each other tang $V = \infty$, and

$$1 + a'a = 0.$$

tion of an intersection of two lines, whose equations are

$$y = ax + b, \quad \text{and} \quad y = a'x + b',$$

$$x = -\frac{b - b'}{a - a'}, \quad \text{and} \quad y = \frac{ab' - a'b}{a - a'}.$$

Equation of a perpendicular from a given point to a given line:

$$y - y' = -\frac{1}{a}(x - x').$$

Equation of the length of the perpendicular P:

$$P = \frac{y' - ax' - b}{\sqrt{1 \times a^2}}.$$

The circle.—Equation of a circle, the origin of coördinates being at the centre, and radius $= R$:

$$x^2 + y^2 = R^2.$$

If the origin is at the left extremity of the diameter, on the axis of X:

$$y^2 = 2Rx - x^2.$$

If the origin is at any point, and the coördinates of the centre are $x'y'$:

$$(x - x')^2 + (y - y')^2 = R^2.$$

Equation of a tangent to a circle, the coördinates of the point of tangency being $x''y''$ and the origin at the centre,

$$yy'' + xx'' = R^2.$$

The ellipse.—Equation of an ellipse, referred to rectangular coördinates with axis at the centre:

$$A^2y^2 + B^2x^2 = A^2B^2,$$

in which A is half the transverse axis and B half the conjugate axis.

Equation of the ellipse when the origin is at the vertex of the transverse axis:

$$y^2 = \frac{B^2}{A^2}(2Ax - x^2).$$

The eccentricity of an ellipse is the distance from the centre to either focus, divided by the semi-transverse axis, or

$$e = \frac{\sqrt{A^2 - B^2}}{A}.$$

The parameter of an ellipse is the double ordinate passing through the focus. It is a third proportional to the transverse axis and its conjugate, or

$$2A : 2B :: 2B : \text{parameter; or parameter} = \frac{2B^2}{A}.$$

Any ordinate of a circle circumscribing an ellipse is to the corresponding ordinate of the ellipse as the semi-transverse axis to the semi-conjugate. Any ordinate of a circle inscribed in an ellipse is to the corresponding ordinate of the ellipse as the semi-conjugate axis to the semi-transverse.

Equation of the tangent to an ellipse, origin of axes at the centre:

$$A^2yy'' + B^2xx'' = A^2B^2,$$

$y''x''$ being the coördinates of the point of tangency.

Equation of the normal, passing through the point of tangency, and perpendicular to the tangent:

$$y - y'' \, xx \, \frac{A^2y''}{B^2x''}(x - x'').$$

The normal bisects the angle of the two lines drawn from the point of tangency to the foci.

The lines drawn from the foci make equal angles with the tangent.

The parabola.—Equation of the parabola referred to rectangular coördinates, the origin being at the vertex of its axis, $y^2 = 2px$, in which p is the parameter or double ordinate through the focus.

rmal, or projection of the normal on the axis, is constant, and lf the parameter.
ent at any point makes equal angles with the axis and with the from the point of tangency to the focus.

perbola.—Equation of the hyperbola referred to rectangular , origin at the centre:

$$A^2y^2 - B^2x^2 = -A^2B^2,$$

is the semi-transverse axis and B the semi-conjugate axis.
when the origin is at the vertex of the transverse axis:

$$y^2 = \frac{B^2}{A^2}(2Ax + x^2).$$

ate and equilateral hyperbolas.—If on the conjugate ransverse, and a focal distance equal to $\sqrt{A^2 + B^2}$, we construct anches of a hyperbola, the two hyperbolas thus constructed are jugate hyperbolas. If the transverse and conjugate axes are hyperbolas are called equilateral, in which case $y^2 - x^2 = -A^2$ the transverse axis, and $x^2 - y^2 = -B^2$ when B is the trans-

meter of the transverse axis is a third proportional to the trans- and its conjugate.

$$2A : 2B :: 2B : \text{parameter}.$$

ent to a hyperbola bisects the angle of the two lines drawn from f tangency to the foci.

ymptotes of a hyperbola are the diagonals of the rectangle on the axes, indefinitely produced in both directions.
uilateral hyperbola the asymptotes make equal angles with the axis, and are at right angles to each other.
nptotes continually approach the hyperbola, and become tangent infinite distance from the centre.

sections.—Every equation of the second degree between two will represent either a circle, an ellipse, a parabola or a hyperbola.
ces are those which are obtained by intersecting the surface of a

DIFFERENTIAL CALCULUS.

The differential of a variable quantity is the difference between an of its consecutive values; hence it is indefinitely small. It is expresse writing d before the quantity, as dx, which is read differential of x.

The term $\frac{dy}{dx}$ is called the differential coefficient of y regarded as a tion of x.

The differential of a function is equal to its differential coefficient tiplied by the differential of the independent variable; thus, $\frac{dy}{dx}dx = d$

The *limit* of a variable quantity is that value to which it conti approaches, so as at last to differ from it by less than any assignable tity.

The differential coefficient is the limit of the ratio of the increment independent variable to the increment of the function.

The differential of a constant quantity is equal to 0.

The differential of a product of a constant by a variable is equal constant multiplied by the differential of the variable.

$$\text{If} \quad u = Av, \quad du = Adv.$$

In any curve whose equation is $y = f(x)$, the differential coeff $\frac{dy}{dx} = \tan a$; hence, the rate of increase of the function, or the ascens the curve at any point, is equal to the tangent of the angle which the ta line makes with the axis of abscissas.

All the operations of the Differential Calculus comprise but two obje

1. To find the rate of change in a function when it passes from one of value to another, consecutive with it.

2. To find the actual change in the function: The rate of change differential coefficient, and the actual change the differential.

Differentials of algebraic functions.—The differential o sum or difference of any number of functions, dependent on the variable, is equal to the sum or difference of their differentials taken rately:

$$\text{If} \quad u = y + z - w, \quad du = dy + dz - dw.$$

The differential of a product of two functions dependent on the variable is equal to the sum of the products of each by the differenti the other:

$$d(uv) = vdu + udv. \quad \frac{d(uv)}{uv} = \frac{du}{u} + \frac{dv}{v}.$$

The differential of the product of any number of functions is equal t sum of the products which arise by multiplying the differential of function by the product of all the others:

$$d(uts) = tsdu + usdt + utds.$$

The differential of a fraction equals the denominator into the differe of the numerator minus the numerator into the differential of the de inator, divided by the square of the denominator:

$$dt = d\left(\frac{u}{v}\right) = \frac{vdu - udv}{v^2}.$$

If the denominator is constant, $dv = 0$, and $dt = \frac{vdu}{v^2} = \frac{du}{v}$.

If the numerator is constant, $du = 0$, and $dt = -\frac{udv}{v^2}$

The differential of the square root of a quantity is equal to the diff tial of the quantity divided by twice the square root of the quantity:

$$\text{or} \quad v = \sqrt{u}, \quad dv = \frac{du}{2\sqrt{u}}; = \frac{1}{2}u^{-\frac{1}{2}}du.$$

ial of any power of a function is equal to the exponent multi-
iction raised to a power less one, multiplied by the differen-
tion, $d(u^n) = nu^{n-1}du$.

for differentiating algebraic functions.

	6. $d\left(\frac{x}{y}\right) = \frac{ydx - xdy}{y^2}$.
lx.	7. $d(x^m) = mx^{m-1}dx$.
$dx + dy$.	8. $d(\sqrt{x}) = \frac{dx}{2\sqrt{x}}$.
$dx - dy$.	
$y + ydx$.	9. $d\left(x^{-\frac{r}{s}}\right) = -\frac{r}{s}x^{-\frac{r}{s}-1}dx$.

ifferential of the form $u = (a + bx^n)^m$:
exponent of the parenthesis into the exponent of the varia-
parenthesis, into the coefficient of the variable, into the bi-
to a power less 1, into the variable within the parenthesis
er less 1, into the differential of the variable.

$$u = d(a + bx^n)^m = mnb(a + bx^n)^{m-1}x^{n-1}dx.$$

ate of change for a given value of the variable:
erential coefficient, and substitute the value of the variable in
nber of the equation.

x is the side of a cube and u its volume, $u = x^3$, $\frac{du}{dx} = 3x^2$.
of change in the volume is three times the square of the
ige is denoted by 1, the rate of change is 3.
The coefficient of expansion by heat of the volume of a body
he linear coefficient of expansion. Thus if the side of a cube
ch, its volume expands .003 cubic inch. $1.001^3 = 1.003003001$.

differential coefficient is the differential coefficient of
vo or more variables under the supposition that only one of
ged its value.
erential is the differential of a function of two or more vari-
supposition that only one of them has changed its value.
erential of a function of any number of variables is equal to
partial differentials.

the partial differentials are $\frac{du}{dx}dx$, $\frac{du}{dy}dy$.

$$^3 - z,\ du = \frac{du}{dx}dx + \frac{du}{dy}dy + \frac{du}{dz}dz; \ = 2xdx + 3y^2dy - dz.$$

—An integral is a functional expression derived from a
ntegration is the operation of finding the primitive function
ential function. It is indicated by the sign $\int$, which is read
f." Thus $\int 2xdx = x^2$; read, the integral of $2xdx$ equals x^2.
an expression of the form $mx^{m-1}dx$ or $x^m dx$, add 1 to the
e variable, and divide by the new exponent and by the differ-
ariable: $\int 3x^2dx = x^3$. (Applicable in all cases except when
$\int x^{-1}dx$ see formula 2 page 78.)

of the product of a constant by the differential of a vari-
the constant multiplied by the integral of the differential:

$$\int ax^m dx = a\int x^m dx = a\frac{1}{m+1}x^{m+1}.$$

of the algebraic sum of any number of differentials is equal to
um of their integrals:

$$= 2ax^2dx - bydy - z^2dz; \ \int du = \frac{2}{3}ax^3 - \frac{b}{2}y^2 - \frac{z^3}{3}.$$

ential of a constant is 0, a constant connected with a vari-
+ or − disappears in the differentiation; thus $d(a + x^m) =$
Hence in integrating a differential expression we must

annex to the integral obtained a constant represented by C to com for the term which may have been lost in differentiation. Thus if w $dy = adx$; $\int dy = a\int dx$. Integrating,

$$y = ax \pm C.$$

The constant C, which is added to the first integral, must have value as to render the functional equation true for every possible va may be attributed to the variable. Hence, after having found t integral equation and added the constant C, if we then make the v equal to zero, the value which the function assumes will be the tr of C.

An indefinite integral is the first integral obtained before the valu constant C is determined.

A particular integral is the integral after the value of C has been f

A definite integral is the integral corresponding to a given valu variable.

Integration between limits.—Having found the indefin gral and the particular integral, the next step is to find the definite i and then the definite integral between given limits of the variable.

The integral of a function, taken between two limits, indicated b values of x, is equal to the difference of the definite integrals cor ing to those limits. The expression

$$\int_{x'}^{x''} dy = a \int dx$$

is read: Integral of the differential of y, taken between the limits x' the least limit, or the limit corresponding to the subtractive integr placed below.

Integrate $du = 9x^2dx$ between the limits $x = 1$ and $x = 3$, u being 81 when $x = 0$. $\int du = \int 9x^2dx = 3x^3 + C$; $C = 81$ when $x = 0$, then

$$\int_{x=1}^{x=3} du = 3(3)^3 + 81, \text{ minus } 3(1)^3 + 81 = 78.$$

Integration of particular forms.

To integrate a differential of the form $du = (a + bx^n)^m x^{n-1}dx$.

1. If there is a constant factor, place it without the sign of the and omit the power of the variable without the parenthesis and t ential;

2. Augment the exponent of the parenthesis by 1, and then di quantity, with the exponent so increased, by the exponent of th thesis, into the exponent of the variable within the parenthesis, int efficient of the variable. Whence

$$\int du = \frac{(a + bx^n)^{m+1}}{(m + 1)nb} = C.$$

The differential of an arc is the hypothenuse of a right-angle tr which the base is dx and the perpendicular dy.

If z is an arc, $dz = \sqrt{dx^2 + dy^2}$ $\quad z = \int \sqrt{dx^2 + dy^2}$.

Quadrature of a plane figure.

The differential of the area of a plane surface is equal to the ordi the differential of the abscissa.

$$ds = ydx.$$

To apply the principle enunciated in the last equation, in finding of any particular plane surface:

Find the value of y in terms of x, from the equation of the boun substitute this value in the differential equation, and then integrate the required limits of x.

Area of the parabola.—Find the area of any portion of mon parabola whose equation is

$$y^2 = 2px; \quad \text{whence } y = \sqrt{2px}.$$

noting the particular integral by s', $s' = \frac{2}{3} xy$.

is, the area of any portion of the parabola, estimated from the ver- equal to ⅔ of the rectangle of the abscissa and ordinate of the extreme The curve is therefore quadrable.

drature of surfaces of revolution.—The differential of a of revolution is equal to the circumference of a circle perpendicular axis into the differential of the arc of the meridian curve.

$$ds = 2\pi y \sqrt{dx^2 + dy^2};$$

ch y is the radius of a circle of the bounding surface in a plane per- ular to the axis of revolution, and x is the abscissa, or distance of the from the origin of coördinate axes.

efore, to find the volume of any surface of revolution:

the value of y and dy from the equation of the meridian curve in of x and dx, then substitute these values in the differential equation, tegrate between the proper limits of x.

pplication of this rule we may find:

curved surface of a cylinder equals the product of the circumference base into the altitude.

convex surface of a cone equals the product of the circumference of se into half the slant height.

surface of a sphere is equal to the area of four great circles, or equal curved surface of the circumscribing cylinder.

ature of volumes of revolution.—A volume of revolution lume generated by the revolution of a plane figure about a fixed line the axis.

denote the volume by V, $dV = \pi y^2\,dx$.

area of a circle described by any ordinate y is πy^2; hence the differ- of a volume of revolution is equal to the area of a circle perpendicular axis into the differential of the axis.

differential of a volume generated by the revolution of a plane figure the axis of Y is $\pi x^2 dy$.

nd the value of V for any given volume of revolution:

the value of y^2 in terms of x from the equation of the meridian substitute this value in the differential equation, and then integrate n the required limits of x.

cient $\frac{d^2u}{dx^2}$, which is read: second differential of u divided by the square of the differential of x (or dx squared).

The third differential coefficient $\frac{d^3u}{dx^3}$ is read: third differential of u divided by dx cubed.

The differentials of the different orders are obtained by multiplying the differential coefficients by the corresponding powers of dx; thus $\frac{d^3u}{dx^3}dx^3 =$ third differential of u.

Sign of the first differential coefficient.—If we have a curve whose equation is $y = fx$, referred to rectangular coördinates, the curve will recede from the axis of X when $\frac{dy}{dx}$ is positive, and approach the axis when it is negative, when the curve lies within the first angle of the coördinate axes. For all angles and every relation of y and x the curve will recede from the axis of X when the ordinate and first differential coefficient have the same sign, and approach it when they have different signs. If the tangent of the curve becomes parallel to the axis of X at any point $\frac{dy}{dx} = 0$. If the tangent becomes perpendicular to the axis of X at any point $\frac{dy}{dx} = \infty$.

Sign of the second differential coefficient.—The second differential coefficient has the same sign as the ordinate when the curve is convex toward the axis of abscissa and a contrary sign when it is concave.

Maclaurin's Theorem.—For developing into a series any function of a single variable as $u = A + Bx + Cx^2 + Dx^3 + Ex^4$, etc., in which A, B, C, etc., are independent of x:

$$u = (u)_{x=0} + \left(\frac{du}{dx}\right)_{x=0} x + \frac{1}{1 \cdot 2}\left(\frac{d^2u}{dx^2}\right)_{x=0} x^2 + \frac{1}{1 \cdot 2 \cdot 3}\left(\frac{d^3u}{dx^3}\right)_{x=0} x^3 + \text{etc.}$$

In applying the formula, omit the expressions $x = 0$, although the coefficients are always found under this hypothesis.

EXAMPLES:

$$(a+x)^m = a^m + ma^{m-1}x + \frac{m}{1}\frac{(m-1)}{2}a^{m-2}x^2 + \frac{m}{1}\frac{(m-1)}{2}\frac{(m-2)}{3}a^{m-3}x^3 + \text{etc.}$$

$$\frac{1}{a+x} = \frac{1}{a} - \frac{x}{a^2} + \frac{x^2}{a^3} - \frac{x^3}{a^4} + \dots \frac{x^n}{a^{n+1}}, \text{ etc.}$$

Taylor's Theorem.—For developing into a series any function of the sum or difference of two independent variables, as $u' = f(x \pm y)$:

$$u' = u + \frac{du}{dx}y + \frac{d^2u}{dx^2}\frac{y^2}{1 \cdot 2} + \frac{d^3u}{dx^3}\frac{y^3}{1 \cdot 2 \cdot 3} + \text{etc.},$$

in which u is what u' becomes when $y = 0$, $\frac{du}{dx}$ is what $\frac{du'}{dx}$ becomes when $y = 0$, etc.

Maxima and minima.—To find the maximum or minimum value of a function of a single variable:

1. Find the first differential coefficient of the function, place it equal to 0, and determine the roots of the equation.
2. Find the second differential coefficient, and substitute each real root, in succession, for the variable in the second member of the equation. Each root which gives a negative result will correspond to a maximum value of the function, and each which gives a positive result will correspond to a *minimum value.*

EXAMPLE.—To find the value of x which will render the function y a *maximum or minimum* in the equation of the circle, $y^2 + x^2 = R^2$;

$$\frac{dy}{dx} = -\frac{x}{y}; \text{ making } -\frac{x}{y} = 0 \text{ gives } x = 0.$$

›ond differential coefficient is: $\frac{d^2y}{dx} = -\frac{x^2+y^2}{y^3}$.

$= 0$, $y = R$; hence $\frac{d^2y}{dx^2} = -\frac{1}{R}$, which being negative, y is a maxi-
· R positive.

lying the rule to practical examples we first find an expression for
tion which is to be made a maximum or minimum.

such expression a constant quantity is found as a factor, it may
ed in the operation; for the product will be a maximum or a mini-
en the variable factor is a maximum or a minimum.

value of the independent variable which renders a function a max-
r a minimum will render any power or root of that function a
m or minimum; hence we may square both members of an equa-
ree it of radicals before differentiating.

se rules we may find:

aximum rectangle which can be inscribed in a triangle is one whose
is half the altitude of the triangle.

titude of the maximum cylinder which can be inscribed in a cone is
⅓ the altitude of the cone.

irface of a cylindrical vessel of a given volume, open at the top, is a
n when the altitude equals half the diameter.

titude of a cylinder inscribed in a sphere when its convex surface is
num is $r\sqrt{2}$. r = radius.

ltitude of a cylinder inscribed in a sphere when the volume is a
m is $\frac{2r}{\sqrt{3}}$.

›rential of an exponential function.

$$\text{If} \quad u = a^x. \quad \ldots \ldots \ldots \quad (1)$$

$$\text{then } du = da^x = a^x k\,dx, \quad \ldots \ldots \quad (2)$$

ı k is a constant dependent on a.

elation between a and k is $a^{\frac{1}{k}} = e$; whence $a = e^k$, ... (3)

ı e = 2.7182818 . . . the base of the Naperian system of logarithms.

arithms.—The logarithms in the Naperian system are denoted by
log or hyperbolic log, hyp. log, or $\log_e$; and in the common system
by log.

$$k = \text{Nap. log}\, a, \quad \log a = k \log e \; \ldots \ldots \quad (4)$$

ommon logarithm of e, = log 2.7182818 . . . = .4342945 . . . , is called
lulus of the common system, and is denoted by M. Hence, if we have
›erian logarithm of a number we can find the common logarithm of
ıe number by multiplying by the modulus. Reciprocally, Nap.
om. log × 2.3025851.

›quation (4) we make $a = 10$, we have

$$1 = k \log e, \quad \text{or} \quad \frac{1}{k} = \log e = M.$$

, the modulus of the common system is equal to 1, divided by the
ıı logarithm of the common base.

ı equation (2) we have

$$\frac{du}{u} = \frac{da^x}{a^x} = k\,dx.$$

make $a = 10$, the base of the common system, $x = \log u$, and

$$d(\log u) = dx = \frac{du}{u} \times \frac{1}{k} = \frac{du}{u} \times M.$$

the differential of a common logarithm of a quantity is equal to the
ıl of the quantity divided by the quantity, into the modulus.

ıke $a = e$, the base of the Naperian system, x becomes the Nape rian logarithm of u, and k becomes 1 (see equation (3)); hence $M = 1$,

$$d(\text{Nap. log } u) = dx = \frac{du}{a^x}; = \frac{du}{u}.$$

That is, the differential of a Naperian logarithm of a quantity is equal differential of the quantity divided by the quantity; and in the Na system the modulus is 1.

Since k is the Naperian logarithm of a, $du = a^x \; l \; a \; dx$. That differential of a function of the form a^x is equal to the function, in Naperian logarithm of the base a, into the differential of the exponer

If we have a differential in a fractional form, in which the numer the differential of the denominator, the integral is the Naperian loga of the denominator. Integrals of fractional differentials of other forn given below:

Differential forms which have known integrals; ponential functions. (l = Nap. log.)

1. $$\int a^x \, l \, a \, dx = a^x + C;$$

2. $$\int \frac{dx}{x} = \int dx \, x^{-1} = lx + C;$$

3. $$\int (xy^{x-1} dy + y^x \, l y \times dx) = y^x + C;$$

4. $$\int \frac{dx}{\sqrt{x^2 \pm a^2}} = l(x + \sqrt{x^2 \pm a^2}) + C;$$

5. $$\int \frac{dx}{\sqrt{x^2 \pm 2ax}} = l\left(x \pm a + \sqrt{x^2 \pm 2ax}\right) + C;$$

6. $$\int \frac{2adx}{a^2 - x^2} = l\left(\frac{a + x}{a - x}\right) + C;$$

7. $$\int \frac{2adx}{x^2 - a^2} = l\left(\frac{x - a}{x + a}\right) + C;$$

8. $$\int \frac{2adx}{x\sqrt{a^2 + x^2}} = l\left(\frac{\sqrt{a^2 + x^2} - a}{\sqrt{a^2 + x^2} + a}\right) + C;$$

9. $$\int \frac{2adx}{x\sqrt{a^2 - x^2}} = l\left(\frac{a - \sqrt{a^2 - x^2}}{a + \sqrt{a^2 - x^2}}\right) + C;$$

10. $$\int \frac{x^{-2} dx}{\sqrt{x + x^{-2}}} = -\, l\left(\frac{1 + \sqrt{1 + a^2 x^2}}{x}\right) + C.$$

Circular functions.—Let z denote an arc in the first quadran sine, x its cosine, v its versed sine, and t its tangent; and the followin tion be employed to designate an arc by any one of its functions, viz.,

$\sin^{-1} y$ denotes an arc of which y is the sine

$\cos^{-1} x$ " " " " " x is the cosine,

$\tan^{-1} t$ " " " " " t is the tangent and "arc whose sine is y," etc.),—we have the following differential forms which have known integrals (r = radius):

$$\int \cos z\,dz = \sin z + C;$$

$$\int -\sin z\,dz = \cos z + C;$$

$$\int \frac{dy}{\sqrt{1-y^2}} = \sin^{-1} y + C;$$

$$\int \frac{-dx}{\sqrt{1-x^2}} = \cos^{-1} x + C;$$

$$\int \frac{dv}{\sqrt{2v-v^2}} = \text{ver-sin}^{-1} v + C;$$

$$\int \frac{dt}{1+t^2} = \tan^{-1} t + C;$$

$$\int \frac{rdy}{\sqrt{r^2-y^2}} = \sin^{-1} y + C;$$

$$\int \frac{-rdx}{\sqrt{r^2-x^2}} = \cos^{-1} x + C;$$

$$\int \sin z\,dz = \text{ver-sin}\, z + C;$$

$$\int \frac{dz}{\cos^2 z} = \tan z + C;$$

$$\int \frac{rd\,v}{\sqrt{2rv+v^2}} = \text{ver-sin}^{-1} v + C;$$

$$\int \frac{r^2dt}{r^2+t^2} = \tan^{-1} t + C;$$

$$\int \frac{du}{\sqrt{a^2-u^2}} = \sin^{-1}\frac{u}{a} + C;$$

$$\int \frac{-du}{\sqrt{a^2-u^2}} = \cos^{-1}\frac{u}{a} + C;$$

$$\int \frac{du}{\sqrt{2au-u^2}} = \text{ver-sin}^{-1}\frac{u}{a} + C;$$

$$\int \frac{adu}{a^2+u^2} = \tan^{-1}\frac{u}{a} + C.$$

The cycloid.—If a circle be rolled along a straight line, any point of the circumference, as P, will describe a curve which is called a cycloid. The circle is called the generating circle, and P the generating point.

The transcendental equation of the cycloid is

$$x = \text{ver-sin}^{-1} y - \sqrt{2ry - y^2},$$

and the differential equation is $dx = \dfrac{ydx}{\sqrt{2ry - y^2}}$.

The area of the cycloid is equal to three times the area of the generating circle.

The surface described by the arc of a cycloid when revolved about its base is equal to 64 thirds of the generating circle.

The volume of the solid generated by revolving a cycloid about its base is equal to five eighths of the circumscribing cylinder.

Integral calculus.—In the integral calculus we have to return from the differential to the function from which it was derived A number of differential expressions are given above, each of which has a known integral corresponding to it, and which being differentiated, will produce the given differential.

In all classes of functions any differential expression may be integrated when it is reduced to one of the known forms; and the operations of the integral calculus consist mainly in making such transformations of given differential expressions as shall reduce them to equivalent ones whose integrals are known.

For methods *of making these transformations* reference must be made to *the text-books on differential and integral calculus.*

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6	.16666667	9	.01449275	2	.00757576	5	.00512820	8	.00387597
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13	.07692308	6	.01315789	9	.00719424	2	.00495049	5	.00377358
14	.07142857	7	.01298701	140	.00714286	3	.00492611	6	.00375940
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3	.04347826	6	.01162791	9	.00671141	12	.00471698	5	.00363636
4	.04166667	7	.01149425	150	.00666667	13	.00469484	6	.00362319
5	.04000000	8	.01136364	1	.00662252	14	.00467290	7	.00361011
6	.03846154	9	.01123595	2	.00657895	15	.00465116	8	.00359712
7	.03703704	90	.01111111	3	.00653595	16	.00462963	9	.00358423
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	.00257732	3	.00220751	18	.00193050	3	.0017
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6	.00143678	1	.00131406	6	.00121065	1	.00112233	6	.00104602
7	.00143472	2	.00131234	7	.00120919	2	.00112108	7	.00104493
8	.00143266	3	.00131062	8	.00120773	3	.00111982	8	.00104384
9	.00143061	4	.00130890	9	.00120627	4	.00111857	9	.00104275
700	.00142857	5	.00130719	830	.00120482	5	.00111732	960	.00104167
1	.00142653	6	.00130548	1	.00120337	6	.00111607	1	.00104[illegible]
2	.00142450	7	.00130378	2	.00120192	7	.00111483	2	.0010[illegible]
3	.00142247	8	.00130208	3	.00120048	8	.00111359	3	.0010[illegible]
4	.00142045	9	.00130039	4	.00119904	9	.00111235	4	.0010[illegible]
5	.00141844	770	.00129870	5	.00119760	900	.00111111	5	.001[illegible]

	No.	Reciprocal.	No.	Reciprocal.	No.	Reciprocal.	No.	Reciprocal.
)	1031	.000969932	1096	.000912409	1161	.000861326	1226	.000815661
3	2	.000968992	7	.000911577	2	.000860585	7	.000814996
6	3	.000968054	8	.000910747	3	.000859845	8	.000814332
9	4	.000967118	9	.000909918	4	.000859106	9	.000813670
33	5	.000966184	1100	.000909091	5	.000858369	1230	.000813008
87	6	.000965251	1	.000908265	6	.000857633	1	.000812348
81	7	.000964320	2	.000907441	7	.000856898	2	.000811688
75	8	.000963391	3	.000906618	8	.000856164	3	.000811030
669	9	.000962464	4	.000905797	9	.000855432	4	.000810373
564	1040	.000961538	5	.000904977	1170	.000854701	5	.000809717
459	1	.000960615	6	.000904159	1	.000853971	6	.000809061
354	2	.000959693	7	.000903342	2	.000853242	7	.000808407
2250	3	.000958774	8	.000902527	3	.000852515	8	.000807754
2145	4	.000957854	9	.000901713	4	.000851789	9	.000807102
2041	5	.000956938	1110	.000900901	5	.000851064	1240	.000806452
01937	6	.000956023	11	.000900090	6	.000850340	1	.000805802
01833	7	.000955110	12	.000899281	7	.000849618	2	.000805153
01729	8	.000954198	13	.000898473	8	.000848896	3	.000804505
01626	9	.000953289	14	.000897666	9	.000848176	4	.000803858
101523	1050	.000952381	15	.000896861	1180	.000847457	5	.000803213
101420	1	.000951475	16	.000896057	1	.000846740	6	.000802568
101317	2	.000950570	17	.000895255	2	.000846024	7	.000801925
101215	3	.000949668	18	.000894454	3	.000845308	8	.000801282
101112	4	.000948767	19	.000893655	4	.000844595	9	.000800640
101010	5	.000947867	1120	.000892857	5	.000843882	1250	.000800000
0100908	6	.000946970	1	.000892061	6	.000843170	1	.000799360
0100806	7	.000946074	2	.000891266	7	.000842460	2	.000798722
0100705	8	.000945180	3	.000890472	8	.000841751	3	.000798085
0100604	9	.000944287	4	.000889680	9	.000841043	4	.000797448
0100502	1060	.000943396	5	.000888889	1190	.000840336	5	.000796813
0100402	1	.000942507	6	.000888099	1	.000839631	6	.000796178
0100301	2	.000941620	7	.000887311	2	.000838926	7	.000795545
0100200	3	.000940734	8	.000886525	3	.000838222	8	.000794913
0100100	4	.000939850	9	.000885740	4	.000837521	9	.000794281
0100000	5	.000938967	1130	.000884956	5	.000836820	1260	.000793651
00999001	6	.000938086	1	.000884173	6	.000836120	1	.000793021
00998004	7	.000937207	2	.000883392	7	.000835422	2	.000792393
00997009	8	.000936330	3	.000882612	8	.000834724	3	.000791766
00996016	9	.000935454	4	.000881834	9	.000834028	4	.000791139
00995025	1070	.000934579	5	.000881057	1200	.000833333	5	.000790514
00994036	1	.000933707	6	.000880282	1	.000832639	6	.000789889
00993049	2	.000932836	7	.000879508	2	.000831947	7	.000789266
00992063	3	.000931966	8	.000878735	3	.000831255	8	.000788643
00991080	4	.000931099	9	.000877963	4	.000830565	9	.000788022
00990099	5	.000930233	1140	.000877193	5	.000829875	1270	.000787402
00989120	6	.000929368	1	.000876424	6	.000829187	1	.000786782
00988142	7	.000928505	2	.000875657	7	.000828500	2	.000786163
00987167	8	.000927644	3	.000874891	8	.000827815	3	.000785546
00986193	9	.000926784	4	.000874126	9	.000827130	4	.000784929
00985222	1080	.000925926	5	.000873362	1210	.000826446	5	.000784314
00984252	1	.000925069	6	.000872600	11	.000825764	6	.000783699
00983284	2	.000924214	7	.000871840	12	.000825082	7	.000783085
00982318	3	.000923361	8	.000871080	13	.000824402	8	.000782473
00981354	4	.000922509	9	.000870322	14	.000823723	9	.000781861
00980392	5	.000921659	1150	.000869565	15	.000823045	1280	.000781250
00979432	6	.000920810	1	.000868810	16	.000822368	1	.000780640
00978474	7	.000919963	2	.000868056	17	.000821693	2	.000780031
00977517	8	.000919118	3	.000867303	18	.000821018	3	.000779423
00976562	9	.000918274	4	.000866551	19	.000820344	4	.000778816
0975610	1090	.000917431	5	.000865801	1220	.000819672	5	.000778210
974659	1	.000916590	6	.000865052	1	.000819001	6	.000777605
73710	2	.000915751	7	.000864304	2	.000818331	7	.00077700
72763	3	.000914913	8	.000863558	3	.000817661	8	.00077639
1817	4	.000914077	9	.000862813	4	.000816993	9	.000775
874	5	.000913242	1160	.000862069	5	.000816326	1290	.000775

No.	Reciprocal.	No.	Reciprocal.	No.	Reciprocal.	No.	Reciprocal.
1356	.000737463	1421	.000703730	1486	.000672948	1551	.000644745
7	.000736920	2	.000703235	7	.000672495	2	.000644330
8	.000736377	3	.000702741	8	.000672043	3	.000643915
9	.000735835	4	.000702247	9	.000671592	4	.000643501
1360	.000735294	5	.000701754	1490	.000671141	5	.000643087
1	.000734754	6	.000701262	1	.000670691	6	.000642673
2	.000734214	7	.000700771	2	.000670241	7	.000642261
3	.000733676	8	.000700280	3	.000669792	8	.000641848
4	.000733138	9	.000699790	4	.000669344	9	.000641437
5	.000732601	1430	.000699301	5	.000668896	1560	.000641026
6	.000732064	1	.000698812	6	.000668449	1	.000640615
7	.000731529	2	.000698324	7	.000668003	2	.000640205
8	.000730994	3	.000697837	8	.000667557	3	.000639795
9	.000730460	4	.000697350	9	.000667111	4	.000639386
1370	.000729927	5	.000696864	1500	.000666667	5	.000638978
1	.000729395	6	.000696379	1	.000666223	6	.000638570
2	.000728863	7	.000695894	2	.000665779	7	.000638162
3	.000728332	8	.000695410	3	.000665336	8	.000637755
4	.000727802	9	.000694927	4	.000664894	9	.000637349
5	.000727273	1440	.000694444	5	.000664452	1570	.000636943
6	.000726744	1	.000693962	6	.000664011	1	.000636537
7	.000726216	2	.000693481	7	.000663570	2	.000636132
8	.000725689	3	.000693001	8	.000663130	3	.000635728
9	.000725163	4	.000692521	9	.000662691	4	.000635324
1380	.000724638	5	.000692041	1510	.000662252	5	.000634921
1	.000724113	6	.000691563	11	.000661813	6	.000634518
2	.000723589	7	.000691085	12	.000661376	7	.000634115
3	.000723066	8	.000690608	13	.000660939	8	.000633714
4	.000722543	9	.000690131	14	.000660502	9	.000633312
5	.000722022	1450	.000689655	15	.000660066	1580	.000632911
6	.000721501	1	.000689180	16	.000659631	1	.000632511
7	.000720980	2	.000688705	17	.000659196	2	.000632111
8	.000720461	3	.000688231	18	.000658761	3	.000631712
9	.000719942	4	.000687758	19	.000658328	4	.000631313
1390	.000719424	5	.000687285	1520	.000657895	5	.000630915
1	.000718907	6	.000686813	1	.000657462	6	.000630517
2	.000718391	7	.000686341	2	.000657030	7	.000630120
3	.000717875	8	.000685871	3	.000656598	8	.000629723
4	.000717360	9	.000685401	4	.000656168	9	.000629327
5	.000716846	1460	.000684932	5	.000655738	1590	.000628931
6	.000716332	1	.000684463	6	.000655308	1	.000628536
7	.000715820	2	.000683994	7	.000654879	2	.000628141
8	.000715308	3	.000683527	8	.000654450	3	.000627746
9	.000714796	4	.000683060	9	.000654022	4	.000627353
1400	.000714286	5	.000682594	1530	.000653595	5	.000626959
1	.000713776	6	.000682128	1	.000653168	6	.000626566
2	.000713267	7	.000681663	2	.000652742	7	.000626174
3	.000712758	8	.000681199	3	.000652316	8	.000625782
4	.000712251	9	.000680735	4	.000651890	9	.000625391
5	.000711744	1470	.000680272	5	.000651466	1600	.000625000
6	.000711238	1	.000679810	6	.000651042	2	.000624219
7	.000710732	2	.000679348	7	.000650618	4	.000623441
8	.000710227	3	.000678887	8	.000650195	6	.000622665
9	.000709723	4	.000678426	9	.000649773	8	.000621890
1410	.000709220	5	.000677966	1540	.000649351	1610	.000621118
11	.000708717	6	.000677507	1	.000648929	2	.000620347
12	.000708215	7	.000677048	2	.000648508	4	.000619578
13	.000707714	8	.000676590	3	.000648088	6	.000618812
14	.000707214	9	.000676132	4	.000647668	8	.000618047
15	.000706714	1480	.000675676	5	.000647249	1620	.000617284
16	.000706215	1	.000675219	6	.000646830	2	.00061652[illegible]
17	.000705716	2	.000674764	7	.000646412	4	.0006157[illegible]
18	.000705219	3	.000674309	8	.000645995	6	.00061[illegible]
19	.000704722	4	.000673854	9	.000645578	8	.0006[illegible]
1420	.000704225	5	.000673401	1550	.000645161	1630	.000[illegible]

ıose sine is y," etc.),—we have the following differential forms ıown integrals (r = radius):

$= \sin z + C;$	$\int \sin z\, dz = \text{ver-sin}\, z + C;$
$= \cos z + C;$	$\int \frac{dz}{\cos^2 z} = \tan z + C;$
$= \sin^{-1} y + C;$	$\int \frac{rd\,v}{\sqrt{2rv + v^2}} = \text{ver-sin}^{-1} v + C;$
$= \cos^{-1} x + C;$	$\int \frac{r^2 dt}{r^2 + t^2} = \tan^{-1} t + C;$
$= \text{ver-sin}^{-1} v + C;$	$\int \frac{du}{\sqrt{a^2 - u^2}} = \sin^{-1} \frac{u}{a} + C;$
$= \tan^{-1} t + C;$	$\int \frac{-du}{\sqrt{a^2 - u^2}} = \cos^{-1} \frac{u}{a} + C;$
$= \sin^{-1} y + C;$	$\int \frac{du}{\sqrt{2au - u^2}} = \text{ver-sin}^{-1} \frac{u}{a} + C;$
$= \cos^{-1} x + C;$	$\int \frac{adu}{a^2 + u^2} = \tan^{-1} \frac{u}{a} + C.$

oid.—If a circle be rolled along a straight line, any point of ence, as P, will describe a curve which is called a cycloid. The l the generating circle, and P the generating point.
ndental equation of the cycloid is

$$x = \text{ver-sin}^{-1} y - \sqrt{2ry - y^2},$$

ential equation is $dx = \frac{ydx}{\sqrt{2ry - y^2}}.$

the cycloid is equal to three times the area of the generating

described by the arc of a cycloid when revolved about its base thirds of the generating circle.
of the solid generated by revolving a cycloid about its base is eighths of the circumscribing cylinder.
calculus.—In the integral calculus we have to return from al to the function from which it was derived. A number of xpressions are given above, each of which has a known in-oonding to it, and which being differentiated, will produce the ıtial.
s of functions any differential expression may be integrated luced to one of the known forms; and the operations of the llus consist mainly in making such transformations of given xpressions as shall reduce them to equivalent ones whose in-ıown.
s of making these transformations reference must be made to s on differential and integral calculus.

RECIPROCALS OF NUMBERS.

No.	Reciprocal.	No.	Reciprocal.	No.	Reciprocal.	No.	Reciprocal.	No.	Re
1	1.00000000	64	.01562500	127	.00787402	190	.00526316	253	[illegible]
2	.50000000	5	.01538461	8	.00781250	1	.00523560	4	[illegible]
3	.33333333	6	.01515151	9	.00775194	2	.00520833	5	[illegible]
4	.25000000	7	.01492537	130	.00769231	3	.00518135	6	[illegible]
5	.20000000	8	.01470588	1	.00763359	4	.00515464	7	[illegible]
6	.16666667	9	.01449275	2	.00757576	5	.00512820	8	[illegible]
7	.14285714	70	.01428571	3	.00751880	6	.00510204	9	[illegible]
8	.12500000	1	.01408451	4	.00746269	7	.00507614	260	[illegible]
9	.11111111	2	.01388889	5	.00740741	8	.00505051	1	[illegible]
10	.10000000	3	.01369863	6	.00735294	9	.00502513	2	[illegible]
11	.09090909	4	.01351351	7	.00729927	200	.00500000	3	[illegible]
12	.08333333	5	.01333333	8	.00724638	1	.00497512	4	[illegible]
13	.07692308	6	.01315789	9	.00719424	2	.00495049	5	[illegible]
14	.07142857	7	.01298701	140	.00714286	3	.00492611	6	[illegible]
15	.06666667	8	.01282051	1	.00709220	4	.00490196	7	[illegible]
16	.06250000	9	.01265823	2	.00704225	5	.00487805	8	[illegible]
17	.05882353	80	.01250000	3	.00699301	6	.00485437	9	[illegible]
18	.05555556	1	.01234568	4	.00694444	7	.00483092	270	[illegible]
19	.05263158	2	.01219512	5	.00689655	8	.00480769	1	[illegible]
20	.05000000	3	.01204819	6	.00684931	9	.00478469	2	[illegible]
1	.04761905	4	.01190476	7	.00680272	210	.00476190	3	[illegible]
2	.04545455	5	.01176471	8	.00675676	11	.00473934	4	[illegible]
3	.04347826	6	.01162791	9	.00671141	12	.00471698	5	[illegible]
4	.04166667	7	.01149425	150	.00666667	13	.00469484	6	[illegible]
5	.04000000	8	.01136364	1	.00662252	14	.00467290	7	[illegible]
6	.03846154	9	.01123595	2	.00657895	15	.00465116	8	[illegible]
7	.03703704	90	.01111111	3	.00653595	16	.00462963	9	[illegible]
8	.03571429	1	.01098901	4	.00649351	17	.00460829	280	[illegible]
9	.03448276	2	.01086956	5	.00645161	18	.00458716	1	[illegible]
30	.03333333	3	.01075269	6	.00641026	19	.00456621	2	[illegible]
1	.03225806	4	.01063830	7	.00636943	220	.00454545	3	[illegible]
2	.03125000	5	.01052632	8	.00632911	1	.00452489	4	[illegible]
3	.03030303	6	.01041667	9	.00628931	2	.00450450	5	[illegible]
4	.02941176	7	.01030928	160	.00625000	3	.00448430	6	[illegible]
5	.02857143	8	.01020408	1	.00621118	4	.00446429	7	[illegible]
6	.02777778	9	.01010101	2	.00617284	5	.00444444	8	[illegible]
7	.02702703	100	.01000000	3	.00613497	6	.00442478	9	[illegible]
8	.02631579	1	.00990099	4	.00609756	7	.00440529	290	[illegible]
9	.02564103	2	.00980392	5	.00606061	8	.00438596	1	[illegible]
40	.02500000	3	.00970874	6	.00602410	9	.00436681	2	[illegible]
1	.02439024	4	.00961538	7	.00598802	230	.00434783	3	[illegible]
2	.02380952	5	.00952381	8	.00595238	1	.00432900	4	[illegible]
3	.02325581	6	.00943396	9	.00591716	2	.00431034	5	[illegible]
4	.02272727	7	.00934579	170	.00588235	3	.00429184	6	[illegible]
5	.02222222	8	.00925926	1	.00584795	4	.00427350	7	[illegible]
6	.02173913	9	.00917431	2	.00581395	5	.00425532	8	[illegible]
7	.02127660	110	.00909091	3	.00578035	6	.00423729	9	[illegible]
8	.02083333	11	.00900901	4	.00574713	7	.00421941	300	[illegible]
9	.02040816	12	.00892857	5	.00571429	8	.00420168	1	[illegible]
50	.02000000	13	.00884956	6	.00568182	9	.00418410	2	[illegible]
1	.01960784	14	.00877193	7	.00564972	240	.00416667	3	[illegible]
2	.01923077	15	.00869565	8	.00561798	1	.00414938	4	[illegible]
3	.01886792	16	.00862069	9	.00558659	2	.00413223	5	[illegible]
4	.01851852	17	.00854701	180	.00555556	3	.00411523	6	[illegible]
5	.01818182	18	.00847458	1	.00552486	4	.00409836	7	[illegible]
6	.01785714	19	.00840336	2	.00549451	5	.00408163	8	[illegible]
7	.01754386	120	.00833333	3	.00546448	6	.00406504	9	[illegible]
8	.01724138	1	.00826446	4	.00543478	7	.00404858	310	[illegible]
9	.01694915	2	.00819672	5	.00540540	8	.00403226	11	[illegible]
60	.01666667	3	.00813008	6	.00537634	9	.00401606	12	[illegible]
[illegible]	.016[illegible]	[illegible]	[illegible]06452	7	.00534759	250	.00400000	13	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]0000	8	.00531914	1	.00398406	14	[illegible]

pro-	No.	Reciprocal.	No.	Reciprocal.	No.	Reciprocal.	No.	Reciprocal.
16456	381	.00262467	446	.00224215	511	.00195695	576	.00173611
15457	2	.00261780	7	.00223714	12	.00195312	7	.00173310
14465	3	.00261097	8	.00223214	13	.00194932	8	.00173010
13480	4	.00260417	9	.00222717	14	.00194552	9	.00172712
12500	5	.00259740	450	.00222222	15	.00194175	580	.00172414
11526	6	.00259067	1	.00221729	16	.00193798	1	.00172117
10559	7	.00258398	2	.00221239	17	.00193424	2	.00171821
09597	8	.00257732	3	.00220751	18	.00193050	3	.00171527
08642	9	.00257069	4	.00220264	19	.00192678	4	.00171233
07692	390	.00256410	5	.00219780	520	.00192308	5	.00170940
06748	1	.00255754	6	.00219298	1	.00191939	6	.00170648
05810	2	.00255102	7	.00218818	2	.00191571	7	.00170358
04878	3	.00254453	8	.00218341	3	.00191205	8	.00170068
03951	4	.00253807	9	.00217865	4	.00190840	9	.00169779
03030	5	.00253165	460	.00217391	5	.00190476	590	.00169491
02115	6	.00252525	1	.00216920	6	.00190114	1	.00169205
01205	7	.00251889	2	.00216450	7	.00189753	2	.00168919
00300	8	.00251256	3	.00215983	8	.00189394	3	.00168634
99401	9	.00250627	4	.00215517	9	.00189036	4	.00168350
98507	400	.00250000	5	.00215054	530	.00188679	5	.00168067
97619	1	.00249377	6	.00214592	1	.00188324	6	.00167785
96736	2	.00248756	7	.00214133	2	.00187970	7	.00167504
95858	3	.00248139	8	.00213675	3	.00187617	8	.00167224
94985	4	.00247525	9	.00213220	4	.00187266	9	.00166945
94118	5	.00246914	470	.00212766	5	.00186916	600	.00166667
93255	6	.00246305	1	.00212314	6	.00186567	1	.00166389
92398	7	.00245700	2	.00211864	7	.00186220	2	.00166113
91545	8	.00245098	3	.00211416	8	.00185874	3	.00165837
90698	9	.00244499	4	.00210970	9	.00185528	3	.00165563
89855	410	.00243902	5	.00210526	540	.00185185	5	.00165289
89017	11	.00243309	6	.00210084	1	.00184843	6	.00165016
88184	12	.00242718	7	.00209644	2	.00184502	7	.00164745
87356	13	.00242131	8	.00209205	3	.00184162	8	.00164474
86533	14	.00241546	9	.00208768	4	.00183823	9	.00164204
85714	15	.00240964	480	.00208333	5	.00183486	610	.00163934
84900	16	.00240385	1	.00207900	6	.00183150	11	.00163666
84091	17	.00239808	2	.00207469	7	.00182815	12	.00163399
83286	18	.00239234	3	.00207039	8	.00182482	13	.00163132
82486	19	.00238663	4	.00206612	9	.00182149	14	.00162866
81690	420	.00238095	5	.00206186	550	.00181818	15	.00162602
80899	1	.00237530	6	.00205761	1	.00181488	16	.00162338
80112	2	.00236967	7	.00205339	2	.00181159	17	.00162075
79330	3	.00236407	8	.00204918	3	.00180832	18	.00161812
78551	4	.00235849	9	.00204499	4	.00180505	19	.00161551
77778	5	.00235294	490	.00204082	5	.00180180	620	.00161290
77008	6	.00234742	1	.00203666	6	.00179856	1	.00161031
76243	7	.00234192	2	.00203252	7	.00179533	2	.00160772
75482	8	.00233645	3	.00202840	8	.00179211	3	.00160514
74725	9	.00233100	4	.00202429	9	.00178891	3	.00160256
73973	430	.00232558	5	.00202020	560	.00178571	5	.00160000
73224	1	.00232019	6	.00201613	1	.00178253	6	.00159744
72480	2	.00231481	7	.00201207	2	.00177936	7	.00159490
71739	3	.00230947	8	.00200803	3	.00177620	8	.00159236
71008	4	.00230115	9	.00200401	4	.00177305	9	.00158982
70270	5	.00229885	500	.00200000	5	.00176991	630	.00158730
69542	6	.00229358	1	.00199601	6	.00176678	1	.00158479
68817	7	.00228833	2	.00199203	7	.00176367	2	.00158228
68096	8	.00228310	3	.00198807	8	.00176056	3	.00157978
67380	9	.00227790	4	.00198413	9	.00175747	4	.00157729
66667	440	.00227273	5	.00198020	570	.00175439	5	.00157480
65957	1	.00226757	6	.00197628	1	.00175131	6	.00157233
65252	2	.00226244	7	.00197239	2	.00174825	7	[illegible]
64550	3	.00225734	8	.00196850	3	.00174520		
[illegible]	4	[illegible]	9	[illegible]	4	[illegible]		

cipro-cal.	No.	Reciprocal.	No.	Reciprocal.	No.	Reciprocal.	No.	Reciprocal.
03520	1031	.000969932	1096	.000912409	1161	.000861326	1226	.000815661
03413	2	.000968992	7	.000911577	2	.000860585	7	.000814996
03306	3	.000968054	8	.000910747	3	.000859845	8	.000814332
03199	4	.000967118	9	.000909918	4	.000859106	9	.000813670
03093	5	.000966184	1100	.000909091	5	.000858369	1230	.000813008
02987	6	.000965251	1	.000908265	6	.000857633	1	.000812348
02881	7	.000964320	2	.000907441	7	.000856898	2	.000811688
02775	8	.000963391	3	.000906618	8	.000856164	3	.000811030
02669	9	.000962464	4	.000905797	9	.000855432	4	.000810373
02564	1040	.000961538	5	.000904977	1170	.000854701	5	.000809717
02459	1	.000960615	6	.000904159	1	.000853971	6	.000809061
02354	2	.000959693	7	.000903342	2	.000853242	7	.000808407
02250	3	.000958774	8	.000902527	3	.000852515	8	.000807754
02145	4	.000957854	9	.000901713	4	.000851789	9	.000807102
02041	5	.000956938	1110	.000900901	5	.000851064	1240	.000806452
01937	6	.000956023	11	.000900090	6	.000850340	1	.000805802
01833	7	.000955110	12	.000899281	7	.000849618	2	.000805153
01729	8	.000954198	13	.000898473	8	.000848896	3	.000804505
01626	9	.000953289	14	.000897666	9	.000848176	4	.000803858
01523	1050	.000952381	15	.000896861	1180	.000847457	5	.000803213
01420	1	.000951475	16	.000896057	1	.000846740	6	.000802568
01317	2	.000950570	17	.000895255	2	.000846024	7	.000801925
01215	3	.000949668	18	.000894454	3	.000845308	8	.000801282
01112	4	.000948767	19	.000893655	4	.000844595	9	.000800640
01010	5	.000947867	1120	.000892857	5	.000843882	1250	.000800000
00908	6	.000946970	1	.000892061	6	.000843170	1	.000799360
00806	7	.000946074	2	.000891266	7	.000842460	2	.000798722
00705	8	.000945180	3	.000890472	8	.000841751	3	.000798085
00604	9	.000944287	4	.000889680	9	.000841043	4	.000797448
00502	1060	.000943396	5	.000888889	1190	.000840336	5	.000796813
00402	1	.000942507	6	.000888099	1	.000839631	6	.000796178
00301	2	.000941620	7	.000887311	2	.000838926	7	.000795545
00200	3	.000940734	8	.000886525	3	.000838222	8	.000794913
00100	4	.000939850	9	.000885740	4	.000837521	9	.000794281
00000	5	.000938967	1130	.000884956	5	.000836820	1260	.000793651
999001	6	.000938086	1	.000884173	6	.000836120	1	.000793021
998004	7	.000937207	2	.000883392	7	.000835422	2	.000792393
997009	8	.000936330	3	.000882612	8	.000834724	3	.000791766
996016	9	.000935454	4	.000881834	9	.000834028	4	.000791139
995025	1070	.000934579	5	.000881057	1200	.000833333	5	.000790514
994036	1	.000933707	6	.000880282	1	.000832639	6	.000789889
993049	2	.000932836	7	.000879508	2	.000831947	7	.000789266
992063	3	.000931966	8	.000878735	3	.000831255	8	.000788643
991080	4	.000931099	9	.000877963	4	.000830565	9	.000788022
990099	5	.000930233	1140	.000877193	5	.000829875	1270	.000787402
989120	6	.000929368	1	.000876424	6	.000829187	1	.000786782
988142	7	.000928505	2	.000875657	7	.000828500	2	.000786163
987167	8	.000927644	3	.000874891	8	.000827815	3	.000785546
986193	9	.000926784	4	.000874126	9	.000827130	4	.000784929
985222	1080	.000925926	5	.000873362	1210	.000826446	5	.000784314
984252	1	.000925069	6	.000872600	11	.000825764	6	.000783699
983284	2	.000924214	7	.000871840	12	.000825082	7	.000783085
982318	3	.000923361	8	.000871080	13	.000824402	8	.000782473
981354	4	.000922509	9	.000870322	14	.000823723	9	.000781861
980392	5	.000921659	1150	.000869565	15	.000823045	1280	.000781250
979432	6	.000920810	1	.000868810	16	.000822368	1	.000780640
978474	7	.000919963	2	.000868056	17	.000821693	2	.000780031
977517	8	.000919118	3	.000867303	18	.000821018	3	.000779423
976562	9	.000918274	4	.000866551	19	.000820344	4	.000778816
975610	1090	.000917431	5	.000865801	1220	.000819672	5	.000778210
974659	1	.000916590	6	.000865052	1	.000819001	6	.000777605
973710	2	.000915751	7	.000864304	2	.000818331	7	.000777001
72763	3	.000914913	8	.000863558	3	.000817661	8	[illegible]
71817	4	.000914077	9	.000862813	4	.000816993	9	[illegible]
0874	5	.000913242	1160	.000862069	5	.000816326	12[illegible]	[illegible]

No.	Reciprocal.	No.	Reciprocal.	No.	Reciprocal.	No.	Reciprocal.	No.	Reci- c
1291	.000774593	1356	.000737463	1421	.000703730	1486	.000672948	1551	.0006
2	.000773994	7	.000736920	2	.000703235	7	.000672495	2	.0006
3	.000773395	8	.000736377	3	.000702741	8	.000672043	3	.0006
4	.000772797	9	.000735835	4	.000702247	9	.000671592	4	.0006
5	.000772201	1360	.000735294	5	.000701754	1490	.000671141	5	.0006
6	.000771605	1	.000734754	6	.000701262	1	.000670691	6	.0006
7	.000771010	2	.000734214	7	.000700771	2	.000670241	7	.0006
8	.000770416	3	.000733676	8	.000700280	3	.000669792	8	.0006
9	.000769823	4	.000733138	9	.000699790	4	.000669344	9	.0006
1300	.000769231	5	.000732601	1430	.000699301	5	.000668896	1560	.0006
1	.000768639	6	.000732064	1	.000698812	6	.000668449	1	.0006
2	.000768049	7	.000731529	2	.000698324	7	.000668003	2	.0006
3	.000767459	8	.000730994	3	.000697837	8	.000667557	3	.0006
4	.000766871	9	.000730460	4	.000697350	9	.000667111	4	.0006
5	.000766283	1370	.000729927	5	.000696864	1500	.000666667	5	.0006
6	.000765697	1	.000729395	6	.000696379	1	.000666223	6	.0006
7	.000765111	2	.000728863	7	.000695894	2	.000665779	7	.0006
8	.000764526	3	.000728332	8	.000695410	3	.000665336	8	.0006
9	.000763942	4	.000727802	9	.000694927	4	.000664894	9	.0006
1310	.000763359	5	.000727273	1440	.000694444	5	.000664452	1570	.0006
11	.000762776	6	.000726744	1	.000693962	6	.000664011	1	.0006
12	.000762195	7	.000726216	2	.000693481	7	.000663570	2	.0006
13	.000761615	8	.000725689	3	.000693001	8	.000663130	3	.0006
14	.000761035	9	.000725163	4	.000692521	9	.000662691	4	.0006
15	.000760456	1380	.000724638	5	.000692041	1510	.000662252	5	.0006
16	.000759878	1	.000724113	6	.000691563	11	.000661813	6	.0006
17	.000759301	2	.000723589	7	.000691085	12	.000661376	7	.0006
18	.000758725	3	.000723066	8	.000690608	13	.000660939	8	.0006
19	.000758150	4	.000722543	9	.000690131	14	.000660502	9	.0006
1320	.000757576	5	.000722022	1450	.000689655	15	.000660066	1580	.0006
1	.000757002	6	.000721501	1	.000689180	16	.000659631	1	.0006
2	.000756430	7	.000720980	2	.000688705	17	.000659196	2	.0006
3	.000755858	8	.000720461	3	.000688231	18	.000658761	3	.0006
4	.000755287	9	.000719942	4	.000687758	19	.000658328	4	.0006
5	.000754717	1390	.000719424	5	.000687285	1520	.000657895	5	.0006
6	.000754148	1	.000718907	6	.000686813	1	.000657462	6	.0006
7	.000753579	2	.000718391	7	.000686341	2	.000657030	7	.0006
8	.000753012	3	.000717875	8	.000685871	3	.000656598	8	.0006
9	.000752445	4	.000717360	9	.000685401	4	.000656168	9	.0006
1330	.000751880	5	.000716846	1460	.000684932	5	.000655738	1590	.0006
1	.000751315	6	.000716332	1	.000684463	6	.000655308	1	.0006
2	.000750750	7	.000715820	2	.000683994	7	.000654879	2	.0006
3	.000750187	8	.000715308	3	.000683527	8	.000654450	3	.0006
4	.000749625	9	.000714796	4	.000683060	9	.000654022	4	.0006
5	.000749064	1400	.000714286	5	.000682594	1530	.000653595	5	.0006
6	.000748503	1	.000713776	6	.000682128	1	.000653168	6	.0006
7	.000747943	2	.000713267	7	.000681663	2	.000652742	7	.0006
8	.000747384	3	.000712758	8	.000681199	3	.000652316	8	.0006
9	.000746826	4	.000712251	9	.000680735	4	.000651890	9	.0006
1340	.000746269	5	.000711744	1470	.000680272	5	.000651466	1600	.0006
1	.000745712	6	.000711238	1	.000679810	6	.000651042	2	.0006
2	.000745156	7	.000710732	2	.000679348	7	.000650618	4	.0006
3	.000744602	8	.000710227	3	.000678887	8	.000650195	6	.0006
4	.000744048	9	.000709723	4	.000678426	9	.000649773	8	.0006
5	.000743494	1410	.000709220	5	.000677966	1540	.000649351	1610	.0006
6	.000742942	11	.000708717	6	.000677507	1	.000648929	2	.0006
7	.000742390	12	.000708215	7	.000677048	2	.000648508	4	.0006
8	.000741840	13	.000707714	8	.000676590	3	.000648088	6	.0006
9	.000741290	14	.000707214	9	.000676132	4	.000647668	8	.0006
1350	.000740741	15	.000706714	1480	.000675676	5	.000647249	1620	.0006
1	.000740192	16	.000706215	1	.000675219	6	.000646830	2	.0006
2	.000739645	17	.000705716	2	.000674764	7	.000646412	4	.0006
3	.000739098	18	.000705219	3	.000674309	8	.000645995	6	.000
4	.0007[illegible]	[illegible]	[illegible]722	4	.000673854	9	.000645578	8	.00
5	.0007[illegible]	[illegible]	[illegible]25	5	.000673401	1550	.000645161	1630	.0

Square.	Cube.	Sq. Root.	Cube Root.	No.	Square.	Cube.	Sq. Root.	Cube Root.
44100	9261000	14.4914	5.9439	265	70225	18609625	16.2788	6.4232
44521	9393931	14.5258	5.9533	266	707.6	18821096	16.3095	6.4312
44944	9528128	14.5602	5.9627	267	71289	19034163	16.3401	6.4393
45369	9663597	14.5945	5.9721	268	71824	19248832	16.3707	6.4473
45796	9800344	14.6287	5.9814	269	72361	19465109	16.4012	6.4553
46225	9938375	14.6629	5.9907	270	72900	19683000	16.4317	6.4633
46656	10077696	14.6969	6.0000	271	73441	19902511	16.4621	6.4713
47089	10218313	14.7309	6.0092	272	73984	20123648	16.4924	6.4792
47524	10360232	14.7648	6.0185	273	74529	20346417	16.5227	6.4872
47961	10503459	14.7986	6.0277	274	75076	20570824	16.5529	6.4951
48400	10648000	14.8324	6.0368	275	75625	20796875	16.5831	6.5030
48841	10793861	14.8661	6.0459	276	76176	21024576	16.6132	6.5108
49284	10941048	14.8997	6.0550	277	76729	21253933	16.6433	6.5187
49729	11089567	14.9332	6.0641	278	77284	21484952	16.6733	6.5265
50176	11239424	14.9666	6.0732	279	77841	21717639	16.7033	6.5343
50625	11390625	15.0000	6.0822	280	78400	21952000	16.7332	6.5421
51076	11543176	15.0333	6.0912	281	78961	22188041	16.7631	6.5499
51529	11697083	15.0665	6.1002	282	79524	22425768	16.7929	6.5577
51984	11852352	15.0997	6.1091	283	80089	22665187	16.8226	6.5654
52441	12008989	15.1327	6.1180	284	80656	22906304	16.8523	6.5731
52900	12167000	15.1658	6.1269	285	81225	23149125	16.8819	6.5808
53361	12326391	15.1987	6.1358	286	81796	23393656	16.9115	6.5885
53824	12487168	15.2315	6.1446	287	82369	23639903	16.9411	6.5962
54289	12649337	15.2643	6.1534	288	82944	23887872	16.9706	6.6039
54756	12812904	15.2971	6.1622	289	83521	24137569	17.0000	6.6115
55225	12977875	15.3297	6.1710	290	84100	24389000	17.0294	6.6191
55696	13144256	15.3623	6.1797	291	84681	24642171	17.0587	6.6267
56169	13312053	15.3948	6.1885	292	85264	24897088	17.0880	6.6343
56644	13481272	15.4272	6.1972	293	85849	25153757	17.1172	6.6419
57121	13651919	15.4596	6.2058	294	86436	25412184	17.1464	6.6494
57600	13824000	15.4919	6.2145	295	87025	25672375	17.1756	6.6569
58081	13997521	15.5242	6.2231	296	87616	25934336	17.2047	6.6644
58564	14172488	15.5563	6.2317	297	88209	26198073	17.2337	6.6719
59049	14348907	15.5885	6.2403	298	88804	26463592	17.2627	6.6794
59536	14526784	15.6205	6.2488	299	89401	26730899	17.2916	6.6869
60025	14706125	15.6525	6.2573	300	90000	27000000	17.3205	6.6943
60516	14886936	15.6844	6.2658	301	90601	27270901	17.3494	6.7018
61009	15069223	15.7162	6.2743	302	91204	27543608	17.3781	6.7092
61504	15252992	15.7480	6.2828	303	91809	27818127	17.4069	6.7166
62001	15438249	15.7797	6.2912	304	92416	28094464	17.4356	6.7240
62500	15625000	15.8114	6.2996	305	93025	28372625	17.4642	6.7313
63001	15813251	15.8430	6.3080	306	93636	28652616	17.4929	6.7387
63504	16003008	15.8745	6.3164	307	94249	28934443	17.5214	6.7460
64009	16194277	15.9060	6.3247	308	94864	29218112	17.5499	6.7533
64516	16387064	15.9374	6.3330	309	95481	29503629	17.5784	6.7606
65025	16581375	15.9687	6.3413	310	96100	29791000	17.6068	6.7679
65536	16777216	16.0000	6.3496	311	96721	30080231	17.6352	6.7752
66049	16974593	16.0312	6.3579	312	97344	30371328	17.6635	6.7824
66564	17173512	16.0624	6.3661	313	97969	30664297	17.6918	6.7897
67081	17373979	16.0935	6.3743	314	98596	30959144	17.7200	6.7969
67600	17576000	16.1245	6.3825	315	99225	31255875	17.7482	6.8041
8121	17779581	16.1555	6.3907	316	99856	31554496	17.7764	6.8113
644	17984728	16.1864	6.3988	317	100489	31855013	17.8045	6.818
69	18191447	16.2173	6.4070	318	101124	32157432	17.8325	6.825
6	18399744	16.2481	6.4151	319	101761	32461759	17.8606	6.83

UARES, CUBES, SQUARE ROOTS AND CUBE ROOTS OF NUMBERS FROM .1 TO 1600.

uare.	Cube.	Sq. Root.	Cube Root.	No.	Square.	Cube.	Sq. Root.	Cube Root.
01	.001	.3162	.4642	3.1	9.61	29.791	1.761	[illegible]
0225	.0034	.3873	.5313	.2	10 24	32.768	1.789	[illegible]
04	.008	.4472	.5848	.3	10.89	35 937	1.817	[illegible]
0625	.0156	.500	.6300	.4	11.56	39.304	1.844	[illegible]
09	027	.5477	.6694	.5	12.25	42.875	1.871	[illegible]
1225	.0429	.5916	.7047	.6	12.96	46.656	1.897	[illegible]
16	.064	.6325	.7368	.7	13.69	50.653	1.924	[illegible]
2025	.0911	.6708	.7663	.8	14.44	54.872	1.949	[illegible]
25	.125	.7071	.7937	.9	15.21	59.319	1.975	[illegible]
3025	.1664	.7416	.8193	4.	16.	64.	2.	[illegible]
36	.216	.7746	.8434	.1	16.81	68.921	2.025	[illegible]
4225	.2746	.8062	.8662	.2	17.64	74.088	2.049	[illegible]
49	.343	.8367	.8879	.3	18.49	79.507	2.074	[illegible]
5625	.4219	.8660	.9086	.4	19.36	85.184	2.098	[illegible]
64	.512	.8944	.9283	.5	20.25	91.125	2.121	[illegible]
7225	.6141	.9219	.9473	.6	21.16	97.336	2.145	[illegible]
81	.729	.9487	.9655	.7	22.09	103.823	2.168	[illegible]
9025	.8574	.9747	.9830	.8	23.04	110.592	2.191	[illegible]
	1.	1.	1.	.9	24.01	117.649	2.214	[illegible]
1025	1.158	1.025	1.016	5.	25.	125.	2.2361	[illegible]
21	1.331	1.049	1.032	.1	26.01	132.651	2.258	[illegible]
3225	1.521	1.072	1.048	.2	27.04	140.608	2.280	[illegible]
44	1.728	1.095	1.063	.3	28.09	148.877	2.302	[illegible]
5625	1.953	1.118	1.077	.4	29.16	157.464	2.324	[illegible]
69	2.197	1.140	1.091	.5	30.25	166.375	2.345	[illegible]
8225	2.460	1.162	1.105	.6	31.36	175.616	2.366	[illegible]
96	2.744	1.183	1.119	.7	32.49	185.193	2.387	[illegible]
1025	3.019	1.204	1.132	.8	33.64	195.112	2.408	[illegible]
25	3.375	1.2247	1.1447	.9	34.81	205.379	2.429	[illegible]
4025	3.724	1.245	1.157	6.	36.	216.	2.4495	[illegible]
56	4.096	1.265	1.170	.1	37.21	226.981	2.470	[illegible]
7225	4.492	1.285	1.182	.2	38.44	238.328	2.490	[illegible]
89	4.913	1.304	1.193	.3	39.69	250.047	2.510	[illegible]
0625	5.359	1.323	1.205	.4	40.96	262.144	2.530	[illegible]
24	5.832	1.342	1.216	.5	42.25	274.625	2.550	[illegible]
4225	6.332	1.360	1.228	.6	43.56	287.496	2.569	[illegible]
61	6.859	1.378	1.239	.7	44.89	300.763	2.588	[illegible]
8025	7.415	1.396	1.249	.8	46.24	314.432	2.608	[illegible]
	8.	1.4142	1.2599	.9	47.61	328.509	2.627	[illegible]
41	9.261	1.449	1.281	7.	49.	343.	2.6458	[illegible]
84	10.648	1.483	1.301	.1	50.41	357.911	2.665	[illegible]
29	12.167	1.517	1.320	.2	51.84	373.248	2.683	[illegible]
76	13.824	1.549	1.339	.3	53.29	389.017	2.702	[illegible]
25	15.625	1.581	1.357	.4	54.76	405.224	2.720	[illegible]
76	17.576	1.612	1.375	.5	56.25	421.875	2.739	[illegible]
29	19.683	1.643	1.392	.6	57.76	438.976	2.757	[illegible]
84	21.952	1.673	1.409	.7	59.29	456.533	2.775	[illegible]
41	24.389	1.703	1.426	.8	60.84	474.552	2.793	[illegible]
	27	1.7321	1.4422	.9	62.41	493.039	2.811	[illegible]

No.	Square.	Cube.	Sq. Root.	Cube Root.	No.	Square.	Cube.	Sq. Root.	Cube Root.
430	184900	79507000	20.7364	7.5478	485	235225	114084125	22.0227	7.8568
431	185761	80062991	20.7605	7.5537	486	236196	114791256	22.0454	7.8622
432	186624	80621568	20.7846	7.5595	487	237169	115501303	22.0681	7.8676
433	187489	81182737	20.8087	7.5654	488	238144	116214272	22.0907	7.8730
434	188356	81746504	20.8327	7.5712	489	239121	116930169	22.1133	7.8784
435	189225	82312875	20.8567	7.5770	490	240100	117649000	22.1359	7.8837
436	190096	82881856	20.8806	7.5828	491	241081	118370771	22.1585	7.8891
437	190969	83453453	20.9045	7.5886	492	242064	119095488	22.1811	7.8944
438	191844	84027672	20.9284	7.5944	493	243049	119823157	22.2036	7.8998
439	192721	84604519	20.9523	7.6001	494	244036	120553784	22.2261	7.9051
440	193600	85184000	20.9762	7.6059	495	245025	121287375	22.2486	7.9105
441	194481	85766121	21.0000	7.6117	496	246016	122023936	22.2711	7.9158
442	195364	86350888	21.0238	7.6174	497	247009	122763473	22.2935	7.9211
443	196249	86938307	21.0476	7.6232	498	248004	123505992	22.3159	7.9264
444	197136	87528384	21.0713	7.6289	499	249001	124251499	22.3383	7.9317
445	198025	88121125	21.0950	7.6346	500	250000	125000000	22.3607	7.9370
446	198916	88716536	21.1187	7.6403	501	251001	125751501	22.3830	7.9423
447	199809	89314623	21.1424	7.6460	502	252004	126506008	22.4054	7.9476
448	200704	89915392	21.1660	7.6517	503	253009	127263527	22.4277	7.9528
449	201601	90518849	21.1896	7.6574	504	254016	128024064	22.4499	7.9581
450	202500	91125000	21.2132	7.6631	505	255025	128787625	22.4722	7.9634
451	203401	91733851	21.2368	7.6688	506	256036	129554216	22.4944	7.9686
452	204304	92345408	21.2603	7.6744	507	257049	130323843	22.5167	7.9739
453	205209	92959677	21.2838	7.6800	508	258064	131096512	22.5389	7.9791
454	206116	93576664	21.3073	7.6857	509	259081	131872229	22.5610	7.9843
455	207025	94196375	21.3307	7.6914	510	260100	132651000	22.5832	7.9896
456	207936	94818816	21.3542	7.6970	511	261121	133432831	22.6053	7.9948
457	208849	95443993	21.3776	7.7026	512	262144	134217728	22.6274	8.0000
458	209764	96071912	21.4009	7.7082	513	263169	135005697	22.6495	8.0052
459	210681	96702579	21.4243	7.7138	514	264196	135796744	22.6716	8.0104
460	211600	97336000	21.4476	7.7194	515	265225	136590875	22.6936	8.0156
461	212521	97972181	21.4709	7.7250	516	266256	137388096	22.7156	8.0208
462	213444	98611128	21.4942	7.7306	517	267289	138188413	22.7376	8.0260
463	214369	99252847	21.5174	7.7362	518	268324	138991832	22.7596	8.0311
464	215296	99897344	21.5407	7.7418	519	269361	139798359	22.7816	8.0363
465	216225	100544625	21.5639	7.7473	520	270400	140608000	22.8035	8.0415
466	217156	101194696	21.5870	7.7529	521	271441	141420761	22.8254	8.0466
467	218089	101847563	21.6102	7.7584	522	272484	142236648	22.8473	8.0517
468	219024	102503232	21.6333	7.7639	523	273529	143055667	22.8692	8.0569
469	219961	103161709	21.6564	7.7695	524	274576	143877824	22.8910	8.0620
470	220900	103823000	21.6795	7.7750	525	275625	144703125	22.9129	8.0671
471	221841	104487111	21.7025	7.7805	526	276676	145531576	22.9347	8.0723
472	222784	105154048	21.7256	8.7860	527	277729	146363183	22.9565	8.0774
473	223729	105823817	21.7486	7.7915	528	278784	147197952	22.9783	8.0825
474	224676	106496424	21.7715	7.7970	529	279841	148035889	23.0000	8.0876
475	225625	107171875	21.7945	7.8025	530	280900	148877000	23.0217	8.0927
476	226576	107850176	21.8174	7.8079	531	281961	149721291	23.0434	8.0978
477	227529	108531333	21.8403	7.8134	532	283024	150568768	23.0651	8.1028
478	228484	109215352	21.8632	7.8188	533	284089	151419437	23.0868	8.1079
479	229441	109902239	21.8861	7.8243	534	285156	152273304	23.1084	8.1130
480	230400	110592000	21.9089	7.8297	535	286225	153130375	23.130	8.1180
481	231361	111284641	21.9317	7.8352	536	287296	153990656	23.1517	8.123
482	232324	111980168	21.9545	7.8406	537	288369	154854153	23.1733	8.128
483	233289	112678587	21.9773	7.8460	538	289444	155720872	23.1948	8.13
484	234256	113379904	22.0000	7.8514	539	290521	156590819	23.2164	8.1

are.	Cube.	Sq. Root.	Cube Root.	No.	Square.	Cube.	Sq. Root.	Cube Root.
.	512.	2.8284	2.	45	2025	91125	6.7082	3.5569
.61	531.441	2.846	2.008	46	2116	97336	6.7823	3.5830
.24	551.368	2.864	2.017	47	2209	103823	6.8557	3.6088
.89	571.787	2.881	2.025	48	2304	110592	6.9282	3.6342
.56	592.704	2.898	2.033	49	2401	117649	7.	3.6593
.25	614.125	2.915	2.041	50	2500	125000	7.0711	3.6840
.96	636.056	2.933	2.049	51	2601	132651	7.1414	3.7084
.69	658.503	2.950	2.057	52	2704	140608	7.2111	3.7325
.44	681.472	2.966	2.065	53	2809	148877	7.2801	3.7563
.21	704.969	2.983	2.072	54	2916	157464	7.3485	3.7798
.	729.	3.	2.0801	55	3025	166375	7.4162	3.8030
.81	753.571	3.017	2.088	56	3136	175616	7.4833	3.8259
.64	778.688	3.033	2.095	57	3249	185193	7.5498	3.8485
.49	804.357	3.050	2.103	58	3364	195112	7.6158	3.8709
.36	830.584	3.066	2.110	59	3481	205379	7.6811	3.8930
0.25	857.375	3.082	2.118	60	3600	216000	7.7460	3.9149
.16	884.736	3.098	2.125	61	3721	226981	7.8102	3.9365
.09	912.673	3.114	2.133	62	3844	238328	7.8740	3.9579
.04	941.192	3.130	2.140	63	3969	250047	7.9373	3.9791
.01	970.299	3.146	2.147	64	4096	262144	8.	4.
	1000	3.1623	2.1544	65	4225	274625	8.0623	4.0207
	1331	3.3166	2.2240	66	4356	287496	8.1240	4.0412
	1728	3.4641	2.2894	67	4489	300763	8.1854	4.0615
	2197	3.6056	2.3513	68	4624	314432	8.2462	4.0817
	2744	3.7417	2.4101	69	4761	328509	8.3066	4.1016
	3375	3.8730	2.4662	70	4900	343000	8.3666	4.1213
	4096	4.	2.5198	71	5041	357911	8.4261	4.1408
	4913	4.1231	2.5713	72	5184	373248	8.4853	4.1602
	5832	4.2426	2.6207	73	5329	389017	8.5440	4.1793
	6859	4.3589	2.6684	74	5476	405224	8.6023	4.1983
	8000	4.4721	2.7144	75	5625	421875	8.6603	4.2172
	9261	4.5826	2.7589	76	5776	438976	8.7178	4.2358
	10648	4.6904	2.8020	77	5929	456533	8.7750	4.2543
	12167	4.7958	2.8439	78	6084	474552	8.8318	4.2727
	13824	4.8990	2.8845	79	6241	493039	8.8882	4.2908
	15625	5.	2.9240	80	6400	512000	8.9443	4.3089
	17576	5.0990	2.9625	81	6561	531441	9.	4.3267
	19683	5.1962	3.	82	6724	551368	9.0554	4.3445
	21952	5.2915	3.0366	83	6889	571787	9.1104	4.3621
	24389	5.3852	3.0723	84	7056	592704	9.1652	4.3795
	27000	5.4772	3.1072	85	7225	614125	9.2195	4.3968
	29791	5.5678	3.1414	86	7396	636056	9.2736	4.4140
	32768	5.6569	3.1748	87	7569	658503	9.3276	4.4310
	35937	5.7446	3.2075	88	7744	681472	9.3808	4.4480
	39304	5.8310	3.2396	89	7921	704969	9.4340	4.4647
	42875	5.9161	3.2711	90	8100	729000	9.4868	4.4814
	46656	6.	3.3019	91	8281	753571	9.5394	4.4979
	50653	6.0828	3.3322	92	8464	778688	9.5917	4.5144
	54872	6.1644	3.3620	93	8649	804357	9.6437	4.5307
	59319	6.2450	3.3912	94	8836	830584	9.6954	4.5468
	64000	6.3246	3.4200	95	9025	857375	9.7468	4.5629
	68921	6.4031	3.4482	96	9216	884736	9.7980	[illegible]
	74088	6.4807	3.4760	97	9409	912673	[illegible]	[illegible]
	79507	6.5574	3.5034	98	9604	941192	[illegible]	[illegible]
	85184	6.6332	3.5303	99	9801	970299	[illegible]	[illegible]

No.	Square.	Cube.	Sq. Root.	Cube Root.	No.	Square.	Cube.
100	10000	1000000	10.	4.6416	155	24025	3723875
101	10201	1030301	10.0499	4.6570	156	24336	3796416
102	10404	1061208	10.0995	4.6723	157	24649	3869893
103	10609	1092727	10.1489	4.6875	158	24964	3944312
104	10816	1124864	10.1980	4.7027	159	25281	4019679
105	11025	1157625	10.2470	4.7177	160	25600	4096000
106	112 6	1191016	10.2956	4.7326	161	25921	4173281
107	11449	1225043	10.3441	4.7475	162	26244	4251528
108	11664	1259712	10.3923	4.7622	163	26569	4330747
109	11881	1295029	10.4403	4.7769	164	26896	4410944
110	12100	1331000	10.4881	4.7914	165	27225	4492125
111	12321	1367631	10.5357	4.8059	166	27556	4574296
112	12544	1404928	10.5830	4.8203	167	27889	4657463
113	12769	1442897	10.6301	4.8346	168	28224	4741632
114	12996	1481544	10.6771	4.8488	169	28561	4826809
115	13225	1520875	10.7238	4.8629	170	28900	4913000
116	13456	1560896	10.7703	4.8770	171	29241	5000211
117	13689	1601613	10.8167	4.8910	172	29584	5088448
118	13924	1643032	10.8628	4.9049	173	29929	5177717
119	14161	1685159	10.9087	4.9187	174	30276	5268024
120	14400	1728000	10.9545	4.9324	175	30625	5359375
121	14641	1771561	11.0000	4.9461	176	30976	5451776
122	14884	1815848	11.0454	4.9597	177	31329	5545233
123	15129	1860867	11.0905	4.9732	178	31684	5639752
124	15376	1906624	11.1355	4.9866	179	32041	5735339
125	15625	1953125	11.1803	5.0000	180	32400	5832000
126	15876	2000376	11.2250	5.0133	181	32761	5929741
127	16129	2048383	11.2694	5 0265	182	33124	6028568
128	16384	2097152	11.3137	5.0397	183	33489	6128487
129	16641	2146689	11.3578	5.0528	184	33856	6229504
130	16900	2197000	11.4018	5.0658	185	34225	6331625
131	17161	2248091	11.4455	5.0788	186	34596	6434856
132	17424	2299968	11.4891	5.0916	187	34969	6539203
133	17689	2352637	11.5326	5.1045	188	35344	6644672
134	17956	2406104	11.5758	5.1172	189	35721	6751269
135	18225	2460375	11.6190	5.1299	190	36100	6859000
136	18496	2515456	11.6619	5.1426	191	36481	6967871
137	18769	2571353	11.7047	5.1551	192	36864	7077888
138	19044	2628072	11.7473	5.1676	193	37249	7189057
139	19321	2685619	11.7898	5.1801	194	37636	7301384
140	19600	2744000	11 8322	5.1925	195	38025	7414875
141	19881	2803221	11.8743	5.2048	196	38416	7529536
142	20164	2863288	11.9164	5.2171	197	38809	7645373
143	20449	2924207	11.9583	5.2293	198	39204	7762392
144	20736	2985984	12.0000	5.2415	199	39601	7880599
145	21025	3048625	12.0416	5.2536	200	40000	8000000
146	21316	3112136	12.0830	5.2656	201	40401	8120601
147	21609	3176523	12.1244	5.2776	202	40804	8242408
148	21904	3241792	12.1655	5.2896	203	41209	8365427
[illegible]	22201	3307949	12.2066	5.3015	204	41616	8489664
[illegible]	[illegible]	[illegible]	12.2474	5.3133	205	42025	8615125
[illegible]	[illegible]	[illegible]	12.2882	5.1251	206	42436	8741816
[illegible]	[illegible]	[illegible]	12.3288	5.3368	207	42849	8869743
[illegible]	[illegible]	[illegible]	12.3693	5.3485	208	43264	8998912

No.	Square.	Cube.	Sq. Root.	Cube Root.	No.	Square.	Cube.	Sq. Root.	Cube Root.
870	756900	658503000	29.4958	9.5464	925	855625	791453125	30.4138	9.7435
871	758641	660776311	29.5127	9.5501	926	857476	794022776	30.4302	9.7470
872	760384	663054848	29.5296	9.5537	927	859329	796597983	30.4467	9.7505
873	762129	665338617	29.5466	9.5574	928	861184	799178752	30.4631	9.7540
874	763876	667627624	29.5635	9.5610	929	863041	801765089	30.4795	9.7575
875	765625	669921875	29.5804	9.5647	930	864900	804357000	30.4959	9.7610
876	767376	672221376	29.5973	9.5683	931	866761	806954491	30.5123	9.7645
877	769129	674526133	29.6142	9.5719	932	868624	809557568	30.5287	9.7680
878	770884	676836152	29.6311	9.5756	933	870489	812166237	30.5450	9.7715
879	772641	679151439	29.6479	9.5792	934	872356	814780504	30.5614	9.7750
880	774400	681472000	29.6648	9.5828	935	874225	817400375	30.5778	9.7785
881	776161	683797841	29.6816	9.5865	936	876096	820025856	30.5941	9.7819
882	777924	686128968	29.6985	9.5901	937	877969	822656953	30.6105	9.7854
883	779689	688465387	29.7153	9.5937	938	879844	825293672	30.6268	9.7889
884	781456	690807104	29.7321	9.5973	939	881721	827936019	30.6431	9.7924
885	783225	693154125	29.7489	9.6010	940	883600	830584000	30.6594	9.7959
886	784996	695506456	29.7658	9.6046	941	885481	833237621	30.6757	9.7993
887	786769	697864103	29.7825	9.6082	942	887364	835896888	30.6920	9.8028
888	788544	700227072	29.7993	9.6118	943	889249	838561807	30.7083	9.8063
889	790321	702595369	29.8161	9.6154	944	891136	841232384	30.7246	9.8097
890	792100	704969000	29.8329	9.6190	945	893025	843908625	30.7409	9.8132
891	793881	707347971	29.8496	9.6226	946	894916	846590536	30.7571	9.8167
892	795664	709732288	29.8664	9.6262	947	896809	849278123	30.7734	9.8201
893	797449	712121957	29.8831	9.6298	948	898704	851971392	30.7896	9.8236
894	799236	714516984	29.8998	9.6334	949	900601	854670349	30.8058	9.8270
895	801025	716917375	29.9166	9.6370	950	902500	857375000	30.8221	9.8305
896	802816	719323136	29.9333	9.6406	951	904401	860085351	30.8383	9.8339
897	804609	721734273	29.9500	9.6442	952	906304	862801408	30.8545	9.8374
898	806404	724150792	29.9666	9.6477	953	908209	865523177	30.8707	9.8408
899	808201	726572699	29.9833	9.6513	954	910116	868250664	30.8869	9.8443
900	810000	729000000	30.0000	9.6549	955	912025	870983875	30.9031	9.8477
901	811801	731432701	30.0167	9.6585	956	913936	873722816	30.9192	9.8511
902	813604	733870808	30.0333	9.6620	957	915849	876467493	30.9354	9.8546
903	815409	736314327	30.0500	9.6656	958	917764	879217912	30.9516	9.8580
904	817216	738763264	30.0666	9.6692	959	919681	881974079	30.9677	9.8614
905	819025	741217625	30.0832	9.6727	960	921600	884736000	30.9839	9.8648
906	820836	743677416	30.0998	9.6763	961	923521	887503681	31.0000	9.8683
907	822649	746142643	30.1164	9.6799	962	925444	890277128	31.0161	9.8717
908	824464	748613312	30.1330	9.6834	963	927369	893056347	31.0322	9.8751
909	826281	751089429	30.1496	9.6870	964	929296	895841344	31.0483	9.8785
910	828100	753571000	30.1662	9.6905	965	931225	898632125	31.0644	9.8819
911	829921	756058031	30.1828	9.6941	966	933156	901428696	31.0805	9.8854
912	831744	758550528	30.1993	9.6976	967	935089	904231063	31.0966	9.8888
913	833569	761048497	30.2159	9.7012	968	937024	907039232	31.1127	9.8922
914	835396	763551944	30.2324	9.7047	969	938961	909853209	31.1288	9.8956
915	837225	766060875	30.2490	9.7082	970	940900	912673000	31.1448	9.8990
916	839056	768575296	30.2655	9.7118	971	942841	915498611	31.1609	9.9024
917	840889	771095213	30.2820	9.7153	972	944784	918330048	31.1769	9.9058
918	842724	773620632	30.2985	9.7188	973	946729	921167317	31.1929	9.9092
919	844561	776151559	30.3150	9.7224	974	948676	924010424	31.2090	9.9126
920	846400	778688000	30.3315	9.7259	975	950625	926859375	31.2250	9.9160
921	848241	781229961	30.3480	9.7294	976	952576	929714176	31.2410	9.9194
922	850084	783777448	30.3645	9.7329	977	954529	932574833	31.2570	9.92
923	851929	786330467	30.3809	9.7364	978	956484	935441352	31.2730	9.926
924	853776	788889024	30.3974	9.7400	979	958441	938313739	31.2890	[illegible]

No.	Square.	Cube.	Sq. Root.	Cube Root.	No.	Square.	Cube.	S… Ro…
320	102400	32768000	17.8885	6.8399	375	140625	52734375	19.3
321	103041	33076161	17.9165	6.8470	376	141376	53157376	19.3
322	103684	33386248	17.9444	6.8541	377	142129	53582633	19.4
323	104329	33698267	17.9722	6.8612	378	142884	54010152	19.4
324	104976	34012224	18.0000	6.8683	379	143641	54439939	19.4
325	105625	34328125	18.0278	6.8753	380	144400	54872000	19.4
326	106276	34645976	18.0555	6.8824	381	145161	55306341	19.5
327	106929	34965783	18.0831	6.8894	382	145924	55742968	19.5
328	107584	35287552	18.1108	6.8964	383	146689	56181887	19.5
329	108241	35611289	18.1384	6.9034	384	147456	56623104	19.5
330	108900	35937000	18.1659	6.9104	385	148225	57066625	19.6
331	109561	36264691	18.1934	6.9174	386	148996	57512456	19.6
332	110224	36594368	18.2209	6.9244	387	149769	57960603	19.6
333	110889	36926037	18.2483	6.9313	388	150544	58411072	19.6
334	111556	37259704	18.2757	6.9382	389	151321	58863869	19.7
335	112225	37595375	18.3030	6.9451	390	152100	59319000	19.7
336	112896	37933056	18.3303	6.9521	391	152881	59776471	19.7
337	113569	38272753	18.3576	6.9589	392	153664	60236288	19.7
338	114244	38614472	18.3848	6.9658	393	154449	60698457	19.8
339	114921	38958219	18.4120	6.9727	394	155236	61162984	19.8
340	115600	39304000	18.4391	6.9795	395	156025	61629875	19.8
341	116281	39651821	18.4662	6.9864	396	156816	62099136	19.8
342	116964	40001688	18.4932	6.9932	397	157609	62570773	19.9
343	117649	40353607	18.5203	7.0000	398	158404	63044792	19.9
344	118336	40707584	18.5472	7.0068	399	159201	63521199	19.9
345	119025	41063625	18.5742	7.0136	400	160000	64000000	20.0
346	119716	41421736	18.6011	7.0203	401	160801	64481201	20.0
347	120409	41781923	18.6279	7.0271	402	161604	64964808	20.0
348	121104	42144192	18.6548	7.0338	403	162409	65450827	20.0
349	121801	42508549	18.6815	7.0406	404	163216	65939264	20.0
350	122500	42875000	18.7083	7.0473	405	164025	66430125	20.1
351	123201	43243551	18.7350	7.0540	406	164836	66923416	20.1
352	123904	43614208	18.7617	7.0607	407	165649	67419143	20.1
353	124609	43986977	18.7883	7.0674	408	166464	67917312	20.1
354	125316	44361864	18.8149	7.0740	409	167281	68417929	20.2
355	126025	44738875	18.8414	7.0807	410	168100	68921000	20.2
356	126736	45118016	18.8680	7.0873	411	168921	69426531	20.2
357	127449	45499293	18.8944	7.0940	412	169744	69934528	20.2
358	128164	45882712	18.9209	7.1006	413	170569	70444997	20.3
359	128881	46268279	18.9473	7.1072	414	171396	70957944	20.3
360	129600	46656000	18.9737	7.1138	415	172225	71473375	20.3
361	130321	47045881	19.0000	7.1204	416	173056	71991296	20.3
362	131044	47437928	19.0263	7.1269	417	173889	72511713	20.4
363	131769	47832147	17.0526	7.1335	418	174724	73034632	20.4
364	132496	48228544	19.0788	7.1400	419	175561	73560059	20.4
365	133225	48627125	19.1050	7.1466	420	176400	74088000	20.4
366	133956	49027896	19.1311	7.1531	421	177241	74618461	20.5
367	134689	49430863	19.1572	7.1596	422	178084	75151448	20.5
368	135424	49836032	19.1833	7.1661	423	178929	75686967	20.5
369	136161	50243409	19.2094	7.1726	424	179776	76225024	20.5
[illegible]	136900	50653000	19.2354	7.1791	425	180625	76765625	20.6
[illegible]	[illegible]	51064811	19.2614	7.1855	426	181476	77308776	20.6
[illegible]	[illegible]	[illegible]78848	19.2873	7.1920	427	182329	77854483	20
[illegible]	[illegible]	[illegible]17	19.3132	7.1984	428	183184	78402752	20
[illegible]	[illegible]	[illegible]	19.3391	7.2048	429	184041	78953589	2

2932375	33.0908	10.3071	1150	1322500	1520875000	33.9116	10.4769
6532736	33.1059	10.3103	1151	1324801	1524845951	33.9264	10.4799
20139673	33.1210	10.3134	1152	1327104	1528823808	33.9411	10.4830
23753192	33.1361	10.3165	1153	1329409	1532808577	33.9559	10.4860
27373299	33.1512	10.3197	1154	1331716	1536800264	33.9706	10.4890
31000000	33.1662	10.3228	1155	1334025	1540798875	33.9853	10.4921
34633301	33.1813	10.3259	1156	1336336	1544804416	34.0000	10.4951
38273208	33.1964	10.3290	1157	1338649	1548816893	34.0147	10.4981
41919727	33.2114	10.3322	1158	1340964	1552836312	34.0294	10.5011
45572864	33.2264	10.3353	1159	1343281	1556862679	34.0441	10.5042
49232625	33.2415	10.3384	1160	1345600	1560896000	34.0588	10.5072
52899016	33.2566	10.3415	1161	1347921	1564936281	34.0735	10.5102
56572043	33.2716	10.3447	1162	1350244	1568983528	34.0881	10 5132
60251712	33.2866	10.3478	1163	1352569	1573037747	34.1028	10.5162
63938029	33.3017	10.3509	1164	1354896	1577098944	34.1174	10.5192
67631000	33.3167	10.3540	1165	1357225	1581167125	34.1321	10 5223
71330631	33.3317	10.3571	1166	1359556	1585242296	34.1467	10.5253
75036928	33.3467	10.3602	1167	1361889	1589324463	34.1614	10.5283
78749897	33.3617	10.3633	1168	1364224	1593413632	34.1760	10.5313
82469544	33.3766	10.3664	1169	1366561	1597509809	34 1906	10.5343
86195875	33.3916	10 3695	1170	1368900	1601613000	34.2053	10.5373
89928896	33.4066	10.3726	1171	1371241	1605723211	34.2199	10.5403
93668613	33.4215	10.3757	1172	1373584	1609840448	34.2345	10.5433
97415032	33.4365	10.3788	1173	1375929	1613964717	34.2491	10.5463
01168159	33.4515	10.3819	1174	1378276	1618096024	34.2637	10.5493
04928000	33.4664	10.3850	1175	1380625	1622234375	34.2783	10.5523
08694561	33.4813	10.3881	1176	1382976	1626379776	34.2929	10.5553
12467848	33.4963	10.3912	1177	1385329	1630532233	34.3074	10.5583
16247867	33.5112	10.3943	1178	1387684	1634691752	34.3220	10.5612
20084624	33.5261	10.3973	1179	1390041	1638858339	34.3366	10.5642
23828125	33.5410	10.4004	1180	1392400	1643032000	34.3511	10.5672

No.	Square.	Cube.	Sq. Root.	Cube Root.	No.	Square.	Cube.	R
540	291600	157464000	23.2379	8.1433	595	354025	210644875	24
541	292681	158340421	23.2594	8.1483	596	355216	211708736	24
542	293764	159220088	23.2809	8.1533	597	356409	212776173	24
543	294849	160103007	23.3024	8.1583	598	357604	213847192	24
544	295936	160989184	23.3238	8.1633	599	358801	214921799	24
545	297025	161878625	23.3452	8.1683	600	360000	216000000	24
546	298116	162771336	23.3666	8.1733	601	361201	217081801	24
547	299209	163667323	23.3880	8.1783	602	362404	218167208	24
548	300304	164566592	23.4094	8.1833	603	363609	219256227	24
549	301401	165469149	23.4307	8.1882	604	364816	220348864	24
550	302500	166375000	23.4521	8.1932	605	366025	221445125	24
551	303601	167284151	23.4734	8.1982	606	367236	222545016	24
552	304704	168196608	23.4947	8.2031	607	368449	223648543	24
553	305809	169112377	23.5160	8.2081	608	369664	224755712	24
554	306916	170031464	23.5372	8.2130	609	370881	225866529	24
555	308025	170953875	23.5584	8.2180	610	372100	226981000	24
556	309136	171879616	23.5797	8.2229	611	373321	228099131	24
557	310249	172808693	23.6008	8.2278	612	374544	229220928	24
558	311364	173741112	23.6220	8.2327	613	375769	230346397	24
559	312481	174676879	23.6432	8.2377	614	376996	231475544	24
560	313600	175616000	23.6643	8.2426	615	378225	232608375	24
561	314721	176558481	23.6854	8.2475	616	379456	233744896	24
562	315844	177504328	23.7065	8.2524	617	380689	234885113	24
563	316969	178453547	23.7276	8.2573	618	381924	236029032	24
564	318096	179406144	23.7487	8.2621	619	383161	237176659	24
565	319225	180362125	23.7697	8.2670	620	384400	238328000	24
566	320356	181321496	23.7908	8.2719	621	385641	239483061	24
567	321489	182284263	23.8118	8.2768	622	386884	240641848	24
568	322624	183250432	23.8328	8.2816	623	388129	241804367	24
*569	323761	184220009	23.8537	8.2865	624	389376	242970624	24
570	324900	185193000	23.8747	8.2913	625	390625	244140625	25
571	326041	186169411	23.8956	8.2962	626	391876	245314376	25
572	327184	187149248	23.9165	8.3010	627	393129	246491883	25
573	328329	188132517	23.9374	8.3059	628	394384	247673152	25
574	329476	189119224	23.9583	8.3107	629	395641	248858189	25
575	330625	190109375	23.9792	8.3155	630	396900	250047000	25
576	331776	191102976	24.0000	8.3203	631	398161	251239591	25
577	332929	192100033	24.0208	8.3251	632	399424	252435968	25
578	334084	193100552	24.0416	8.3300	633	400689	253636137	25
579	335241	194104539	24.0624	8.3348	634	401956	254840104	25
580	336400	195112000	24.0832	8.3396	635	403225	256047875	25
581	337561	196122941	24.1039	8.3443	636	404496	257259456	25
582	338724	197137368	24.1247	8.3491	637	405769	258474853	25
583	339889	198155287	24.1454	8.3539	638	407044	259694072	25
584	341056	199176704	24.1661	8.3587	639	408321	260917119	25
585	342225	200201625	24.1868	8.3634	640	409600	262144000	25
586	343396	201230056	24.2074	8.3682	641	410881	263374721	25
587	344569	202262003	24.2281	8.3730	642	412164	264609288	25
588	345744	203297472	24.2487	8.3777	643	413449	265847707	25
589	346921	204336469	24.2693	8.3825	644	414736	267089984	25
[illegible]	[illegible]	[illegible]9000	24.2899	8.3872	645	416025	268336125	25
[illegible]	[illegible]	[illegible]071	24.3105	8.3919	646	417316	269586136	25
[illegible]	[illegible]	[illegible]88	24.3311	8.3967	647	418609	270840023	[illegible]
[illegible]	[illegible]	[illegible]57	24.3516	8.4014	648	419904	272097792	[illegible]
[illegible]	[illegible]	[illegible]54	24.3721	8.4061	649	421201	273359449	[illegible]

025	281011375	25.5930	8.6845	710	504100	357911000	26.6458	8.9211
336	282300416	25.6125	8.6890	711	505521	359425431	26.6646	8.9253
649	283593393	25.6320	8.6934	712	506944	360944128	26.6833	8.9295
964	284890312	25.6515	8.6978	713	508369	362467097	26.7021	8.9337
281	286191179	25.6710	8.7022	714	509796	363994344	26.7208	8.9378
600	287496000	25.6905	8.7066	715	511225	365525875	26.7395	8.9420
921	288804781	25.7099	8.7110	716	512656	367061696	26.7582	8.9462
244	290117528	25.7294	8.7154	717	514089	368601813	26.7769	8.9503
569	291434247	25.7488	8.7198	718	515524	370146232	26.7955	8.9545
896	292754944	25.7682	8.7241	719	516961	371694959	26.8142	8.9587
225	294079625	25.7876	8.7285	720	518400	373248000	26.8328	8.9628
556	295408296	25.8070	8.7329	721	519841	374805361	26.8514	8.9670
889	296740963	25.8263	8.7373	722	521284	376367048	26.8701	8.9711
224	298077632	25.8457	8.7416	723	522729	377933067	26.8887	8.9752
561	299418309	25.8650	8.7460	724	524176	379503424	26.9072	8.9794
900	300763000	25.8844	8.7503	725	525625	381078125	26.9258	8.9835
241	302111711	25.9037	8.7547	726	527076	382657176	26.9444	8.9876
584	303464448	25.9230	8.7590	727	528529	384240 83	26.9629	8.9918
929	304821217	25.9422	8.7634	728	529984	385828352	26.9815	8.9959
276	306182024	25.9615	8.7677	729	531441	387420489	27.0000	9.0000
625	307546875	25.9808	8.7721	730	532900	389017000	27.0185	9.0041
976	308915776	26.0000	8.7764	731	534361	390617891	27.0370	9.0082
329	310288733	26.0192	8.7807	732	535824	392223168	27.0555	9.0123
684	311665752	26.0384	8.7850	733	537289	393832837	27.0740	9.0164
041	413046839	26.0576	8.7893	734	538756	395446904	27.0924	9.0205
400	314432000	26.0768	8.7937	735	540225	397065375	27.1109	9.0246
761	315821241	26.0960	8.7980	736	541696	398688256	27.1293	9.0287
124	317214568	26.1151	8.8023	737	543169	400315553	27.1477	9.0328
489	318611987	26.1343	8.8066	738	544644	401947272	27.1662	9.0369
856	320013504	26.1534	8.8109	739	546121	403583419	27.1846	9.0410
225	321419125	26.1725	8.8152	740	547600	405224000	27.2029	9.0450
596	322828856	26.1916	8.8194	741	549801	406869021	27.2213	9.0491
969	324242703	26.2107	8.8237	742	550564	408518488	27.2397	9.0532
344	325660672	26.2298	8.8280	743	552049	410172407	27.2580	9.0572

No.	Square.	Cube.	Sq. Root.	Cube. Root.
980	960400	941192000	31.3050	9.9329
981	962361	944076141	31.3209	9.9363
982	964324	946966168	31.3369	9.9396
983	966289	949862087	31.3528	9.9430
984	968256	952763904	31.3688	9.9464
985	970225	955671625	31.3847	9.9497
986	972196	958585256	31.4006	9.9531
987	974169	961504803	31.4166	9.9565
988	976144	964430272	31.4325	9.9598
989	978121	967361669	31.4484	9.9632
990	980100	970299000	31.4643	9.9666
991	982081	973242271	31.4802	9.9699
992	984064	976191488	31.4960	9.9733
993	986049	979146657	31.5119	9.9766
994	988036	982107784	31.5278	9.9800
995	990025	985074875	31.5436	9.9833
996	992016	988047936	31.5595	9.9866
997	994009	991026973	31.5753	9.9900
998	996004	994011992	31.5911	9.9933
999	998001	997002999	31.6070	9.9967
1000	1000000	1000000000	31.6228	10.0000
1001	1002001	1003003001	31.6386	10.0033
1002	1004004	1006012008	31.6544	10.0067
1003	1006009	1009027027	31.6702	10.0100
1004	1008016	1012048064	31.6860	10.0133
1005	1010025	1015075125	31.7017	10.0166
1006	1012036	1018108216	31.7175	10.0200
1007	1014049	1021147343	31.7333	10.0233
1008	1016064	1024192512	31.7490	10.0266
1009	1018081	1027243729	31.7648	10.0299
1010	1020100	1030301000	31.7805	10.0332
1011	1022121	1033364331	31.7962	10.0365
1012	1024144	1036433728	31.8119	10.0398
1013	1026169	1039509197	31.8277	10.0431
1014	1028196	1042590744	31.8434	10.0465
1015	1030225	1045678375	31.8591	10.0498
1016	1032256	1048772096	31.8748	10.0531
1017	1034289	1051871913	31.8904	10.0563
1018	1036324	1054977832	31.9061	10.0596
1019	1038361	1058089859	31.9218	10.0629
1020	1040400	1061208000	31.9374	10.0662
1021	1042441	1064332261	31.9531	10.0695
1022	1044484	1067462648	31.9687	10.0728
1023	1046529	1070599167	31.9844	10.0761
1024	1048576	1073741824	32.0000	10.0794
1025	1050625	1076890625	32.0156	10.0826
1026	1052676	1080045576	32.0312	10.0859
1027	1054729	1083206683	32.0468	10.0892
1028	1056784	1086373952	32.0624	10.0925
1029	1058841	1089547389	32.0780	10.0957
[illegible]	1060900	1092727000	32.0936	10.0990
[illegible]	1062961	1095912791	32.1092	10.1023
[illegible]	[illegible]	[illegible]	32.1248	10.1055
[illegible]	[illegible]	[illegible]	32.1403	10.1088
[illegible]	[illegible]	[illegible]	32.1559	10.1121

No.	Square.	Cube.	Sq. Root.
1035	1071225	1108717875	32.1714
1036	1073296	1111934656	32.1870
1037	1075369	1115157653	32.2025
1038	1077444	1118386872	32.2180
1039	1079521	1121622319	32.2335
1040	1081600	1124864000	32.2490
1041	1083681	1128111921	32.2645
1042	1085764	1131366088	32.2800
1043	1087849	1134626507	32.2955
1044	1089936	1137893184	32.3110
1045	1092025	1141166125	32.3265
1046	1094116	1144445336	32.3419
1047	1096209	1147730823	32.3574
1048	1098304	1151022592	32.3728
1049	1100401	1154320649	32.3883
1050	1102500	1157625000	32.4037
1051	1104601	1160935651	32.4191
1052	1106704	1164252608	32.4345
1053	1108809	1167575877	32.4500
1054	1110916	1170905464	32.4654
1055	1113025	1174241375	32.4808
1056	1115136	1177583616	32.4962
1057	1117249	1180932193	32.5115
1058	1119364	1184287112	32.5269
1059	1121481	1187648379	32.5423
1060	1123600	1191016000	32.5576
1061	1125721	1194389981	32.5730
1062	1127844	1197770328	32.5883
1063	1129969	1201157047	32.6036
1064	1132096	1204550144	32.6190
1065	1134225	1207949625	32.6343
1066	1136356	1211355496	32.6497
1067	1138489	1214767763	32.6650
1068	1140624	1218186432	32.6803
1069	1142761	1221611509	32.6956
1070	1144900	1225043000	32.7109
1071	1147041	1228480911	32.7261
1072	1149184	1231925248	32.7414
1073	1151329	1235376017	32.7567
1074	1153476	1238833224	32.7719
1075	1155625	1242296875	32.7872
1076	1157776	1245766976	32.8024
1077	1159929	1249243533	32.8177
1078	1162084	1252726552	32.8329
1079	1164241	1256216039	32.8481
1080	1166400	1259712000	32.8634
1081	1168561	1263214441	32.8786
1082	1170724	1266723368	32.8938
1083	1172889	1270238787	32.9090
1084	1175056	1273760704	32.9242
1085	1177225	1277289125	32.9393
1086	1179396	1280824056	32.9545
1087	1181569	1284365503	32.9697
1088	1183744	1287913472	32.9848
1089	1185921	1291467969	33.000

No.	Square.	Cube.	Sq. Root.	Cube Root.	No.	Square.	Cube.	Sq. Root.	Cube Root.
1530	2340900	3581577000	39.1152	11.5230	1565	2449225	3833037125	39.5601	11.6102
1531	2343961	3588604291	39.1280	11.5255	1566	2452356	3840389496	39.5727	11.6126
1532	2347024	3595640768	39.1408	11.5280	1567	2455489	3847751263	39.5854	11.6151
1533	2350089	3602686437	39.1535	11.5305	1568	2458624	3855123432	39.5980	11.6176
1534	2353156	3609741304	39.1663	11.5330	1569	2461761	3862503009	39.6106	11.6200
1535	2356225	3616805375	39.1791	11.5355	1570	2464900	3869893000	39.6232	11.6225
1536	2359296	3623878656	39.1918	11.5380	1571	2468041	3877292411	39.6358	11.6250
1537	2362369	3630961153	39.2046	11.5405	1572	2471184	3884701248	39.6485	11.6274
1538	2365444	3638052872	39.2173	11.5430	1573	2474329	3892119517	39.6611	11.6299
1539	2368521	3645153819	39.2301	11.5455	1574	2477476	3899547224	39.6737	11.6324
1540	2371600	3652264000	39.2428	11.5480	1575	2480625	3906984375	39.6863	11.6348
1541	2374681	3659383421	39.2556	11.5505	1576	2483776	3914430976	39.6989	11.6373
1542	2377764	3666512088	39.2683	11.5530	1577	2486929	3921887033	39.7115	11.6398
1543	2380849	3673650007	39.2810	11.5555	1578	2490084	3929352552	39.7240	11.6422
1544	2383936	3680797184	39.2938	11.5580	1579	2493241	3936827539	39.7366	11.6447
1545	2387025	3687953625	39.3065	11.5605	1580	2496400	3944312000	39.7492	11.6471
1546	2390116	3695119336	39.3192	11.5630	1581	2499561	3651805941	39.7618	11.6496
1547	2393209	3702294323	39.3319	11.5655	1582	2502724	3959309368	39.7744	11.6520
1548	2396304	3709478592	39.3446	11.5680	1583	2505889	3966822287	39.7869	11.6545
1549	2399401	3716672149	39.3573	11.5705	1584	2509056	3974344704	39.7995	11.6570
1550	2402500	3723875000	39.3700	11.5729	1585	2512225	3981876625	39.8121	11.6594
1551	2405601	3731087151	39.3827	11.5754	1586	2515396	3989418056	39.8246	11.6619
1552	2408704	3738308608	39.3954	11.5779	1587	2518569	3996969003	39.8372	11.6643
1553	2411809	3745539377	39.4081	11.5804	1588	2521744	4004529472	39.8497	11.6668
1554	2414916	3752779464	39.4208	11.5829	1589	2524921	4012099469	39.8623	11.6692
1555	2418025	3760028875	39.4335	11.5854	1590	2528100	4019679000	39.8748	11.6717
1556	2421136	3767287616	39.4462	11.5879	1591	2531281	4027268071	39.8873	11.6741
1557	2424249	3774555693	39.4588	11.5903	1592	2534464	4034866688	39.8999	11.6765
1558	2427364	3781833112	39.4715	11.5928	1593	2537649	4042474857	39.9124	11.6790
1559	2430481	3789119879	39.4842	11.5953	1594	2540836	4050092584	39.9249	11.6814
1560	2433600	3796416000	39.4968	11.5978	1595	2544025	4057719875	39.9375	11.7839
1561	2436721	3803721481	39.5095	11.6003	1596	2547216	4065356736	39.9500	11.6863
1562	2439844	3811036328	39.5221	11.6027	1597	2550409	4073003173	39.9625	11.6888
1563	2442969	3818360547	39.5348	11.6052	1598	2553604	4080659192	39.9750	11.6912
1564	2446096	3825694144	39.5474	11.6077	1599	2556801	4088324799	39.9875	11.6936
					1600	2560000	4096000000	40.0000	11.6961

SQUARES AND CUBES OF DECIMALS.

No.	Square.	Cube.	No.	Square.	Cube.	No.	Square.	Cube.
.1	.01	.001	.01	.0001	.000 001	.001	.00 00 01	.000 000 001
.2	.04	.008	.02	.0001	.000 008	.002	.00 00 04	.000 000 008
.3	.09	.027	.03	.0009	.000 027	.003	.00 00 09	.000 000 027
.4	.16	.064	.04	.0016	.000 064	.004	.00 00 16	.000 000 064
.5	.25	.125	.05	.0025	.000 125	.005	.00 00 25	.000 000 125
.6	.36	.216	.06	.0036	.000 216	.006	.00 00 36	.000 000 216
.7	.49	.343	.07	.0049	.000 343	.007	.00 00 49	.000 000 343
.8	.64	.512	.08	.0064	.000 512	.008	.00 00 64	.000 000 512
.9	.81	.729	.09	.0081	.000 729	.009	.00 00 81	.000 000 729
1.0	1.00	1.000	.10	.0100	.001 000	.010	.00 01 00	.000 001 000
1.2	1.44	1.728	.12	.0144	.001 728	.012	.00 01 44	.000 001 728

te that the square has twice as many decimal places, and the cube three as many decimal places, as the root.

No.	Square.	Cube.	Sq. Root.	Cube Root.	No.	Square.	Cube.	Sq. Root.	Cube Root.
1200	1440000	1728000000	34.6410	10.6266	1255	1575025	1976656375	35.4260	10.
1201	1442401	1732323601	34.6554	10.6295	1256	1577536	1981385216	35.4401	10.
1202	1444804	1736654408	34.6699	10.6325	1257	1580049	1986121593	35.4542	10.
1203	1447209	1740992427	34.6843	10.6354	1258	1582564	1990865512	35.4683	10.
1204	1449616	1745337664	34.6987	10.6384	1259	1585081	1995616979	35.4824	10.
1205	1452025	1749690125	34.7131	10.6413	1260	1587600	2000376000	35.4965	10.
1206	1454436	1754049816	34.7275	10.6443	1261	1590121	2005142581	35.5106	10.
1207	1456849	1758416743	34.7419	10.6472	1262	1592644	2009916728	35.5246	10.
1208	1459264	1762790912	34.7563	10.6501	1263	1595169	2014698447	35.5387	10.
1209	1461681	1767172329	34.7707	10.6530	1264	1597696	2019487744	35.5528	10.
1210	1464100	1771561000	34.7851	10.6560	1265	1600225	2024284625	35.5668	10.
1211	1466521	1775956931	34.7994	10.6590	1266	1602756	2029089096	35.5809	10.
1212	1468944	1780360128	34.8138	10.6619	1267	1605289	2033901163	35.5949	10.
1213	1471369	1784770597	34.8281	10.6648	1268	1607824	2038720832	35.6090	10.
1214	1473796	1789188344	34.8425	10.6678	1269	1610361	2043548109	35.6230	10.
1215	1476225	1793613375	34.8569	10.6707	1270	1612900	2048383000	35.6371	10.
1216	1478656	1798045696	34.8712	10.6736	1271	1615441	2053225511	35.6511	10.
1217	1481089	1802485313	34.8855	10.6765	1272	1617984	2058075648	35.6651	10.
1218	1483524	1806932232	34.8999	10.6795	1273	1620529	2062933417	35.6791	10.
1219	1485961	1811386459	34.9142	10.6824	1274	1623076	2067798824	35.6931	10.
1220	1488400	1815848000	34.9285	10.6853	1275	1625625	2072671875	35.7071	10.
1221	1490841	1820316861	34.9428	10.6882	1276	1628176	2077552576	35.7211	10.
1222	1493284	1824793048	34.9571	10.6911	1277	1630729	2082440933	35.7351	10.
1223	1495729	1829276567	34.9714	10.6940	1278	1633284	2087336952	35.7491	10.
1224	1498176	1833767424	34.9857	10.6970	1279	1635841	2092240639	35.7631	10.
1225	1500625	1838265625	35.0000	10.6999	1280	1638400	2097152000	35.7771	10.
1226	1503076	1842771176	35.0143	10.7028	1281	1640961	2102071041	35.7911	10.
1227	1505529	1847284083	35.0286	10.7057	1282	1643524	2106997768	35.8050	10.
1228	1507984	1851804352	35.0428	10.7086	1283	1646089	2111932187	35.8190	10.
1229	1510441	1856331989	35.0571	10.7115	1284	1648656	2116874304	35.8329	10.
1230	1512900	1860867000	35.0714	10.7144	1285	1651225	2121824125	35.8469	10.
1231	1515361	1865409391	35.0856	10.7173	1286	1653796	2126781656	35.8608	10.
1232	1517824	1869959168	35.0999	10.7202	1287	1656369	2131746903	35.8748	10.
1233	1520289	1874516337	35.1141	10.7231	1288	1658944	2136719872	35.8887	10.
1234	1522756	1879080904	35.1283	10.7260	1289	1661521	2141700569	35.9026	10.
1235	1525225	1883652875	35.1426	10.7289	1290	1664100	2146689000	35.9166	10.
1236	1527696	1888232256	35.1568	10.7318	1291	1666681	2151685171	35.9305	10.
1237	1530169	1892819053	35.1710	10.7347	1292	1669264	2156689088	35.9444	10.
1238	1532644	1897413272	35.1852	10.7376	1293	1671849	2161700757	35.9583	10.
1239	1535121	1902014919	35.1994	10.7405	1294	1674436	2166720184	35.9722	10.
1240	1537600	1906624000	35.2136	10.7434	1295	1677025	2171747375	35.9861	10.
1241	1540081	1911240521	35.2278	10.7463	1296	1679616	2176782336	36.0000	10.
1242	1542564	1915864488	35.2420	10.7491	1297	1682209	2181825073	36.0139	10.
1243	1545049	1920495907	35.2562	10.7520	1298	1684804	2186875592	36.0278	10.
1244	1547536	1925134784	35.2704	10.7549	1299	1687401	2191933899	36.0416	10.
1245	1550025	1929781125	35.2846	10.7578	1300	1690000	2197000000	36.0555	10.
1246	1552516	1934434936	35.2987	10.7607	1301	1692601	2202073901	36.0694	10.
1247	1555009	1939096223	35.3129	10.7635	1302	1695204	2207155608	36.0832	10.
1248	1557504	1943764992	35.3270	10.7664	1303	1697809	2212245127	36.0971	10.
1249	1560001	1948441249	35.3412	10.7693	1304	1700416	2217342464	36.1109	10.
1250	1562500	1953125000	35.3553	10.7722	1305	1703025	2222447625	36.1248	10.
1251	[illegible]	[illegible]16251	35.3695	10.7750	1306	1705636	2227560616	36.1386	10.
[illegible]	[illegible]	[illegible]008	35.3836	10.7779	1307	1708249	2232681443	36.152[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	35.3977	10.7808	1308	1710864	2237810112	36.166[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	35.4119	[illegible]	1309	1713481	2242946629	36.180[illegible]	[illegible]

RENCES AND AREAS OF CIRCLES.

s.	Diam.	Circum.	Area.	Diam.	Circum.	Area.
7854	65	204.20	3318.31	129	405.27	13069.81
1416	66	207.34	3421.19	**130**	408.41	13273.23
0686	67	210.49	3525.65	131	411.55	13478.22
5664	68	213.63	3631.68	132	414.69	13684 78
335	69	216.77	3739.28	133	417 83	13892.91
274	**70**	219.91	3848.45	134	420.97	14102.61
485	71	223.05	3959.19	135	424.12	14313.88
266	72	226.19	4071.50	136	427.26	14526.72
517	73	229.34	4185.39	137	430.40	14741.14
540	74	232.48	4300.84	138	433.54	14957.12
033	75	235.62	4417.86	139	436.68	15174.68
10	76	238.76	4536.46	**140**	439.82	15393.80
73	77	241.90	4656.63	141	442.96	15614.50
94	78	245.04	4778.36	142	446.11	15836.77
71	79	248.19	4901.67	143	449.25	16060.61
06	**80**	251.33	5026.55	144	452.39	16286.02
98	81	254.47	5153.00	145	455.53	16513.00
47	82	257.61	5281.02	146	458.67	16741.55
53	83	260.75	5410.61	147	461.81	16971 67
16	84	263.89	5541.77	148	464.96	17203.36
36	85	267.04	5674 50	149	468.10	17436 62
13	86	270.18	5808.80	**150**	471.24	17671.46
48	87	273.32	5944.68	151	474.38	17907 86
39	88	276.46	6082.12	152	477.52	18145.84
87	89	279.60	6221.14	153	480.66	18385.39
03	**90**	282.74	6361.73	154	483.81	18626.50
56	91	285.88	6503.88	155	486.95	18869.19
75	92	289.03	6647.61	156	490.09	19113.45
52	93	292.17	6792.91	157	493.23	19359.28
86	94	295.31	6939.78	158	496.37	19606.68
77	95	298.45	7088.22	159	499.51	19855.65
25	96	301.59	7238.23	**160**	502.65	20106.19
30	97	304.73	7389.81	161	505.80	20358.31
92	98	307.88	7542.96	162	508.94	20611.99
11	99	311.02	7697.69	163	512.08	20867.24
88	**100**	314.16	7853.98	164	515.22	21124.07
21	101	317.30	8011.85	165	518.36	21382.46
11	102	320 44	8171.28	166	521.50	21642.43
59	103	323.58	8332.29	167	524.65	21903 97
64	104	326.73	8494.87	168	527.79	22167 08
25	105	329 87	8659.01	169	530.93	22431.76
44	106	333.01	8824.73	**170**	534.07	22698.01
20	107	336.15	8992.02	171	537.21	22965.83
53	108	339.29	9160.88	172	540.35	23235.22
43	109	342.43	9331.32	173	543.50	23506.18
96	**110**	345.58	9503.32	174	546.64	23778.71
94	111	348.72	9676.89	175	549 78	24052.82
56	112	351.86	9852.03	176	552.92	24328.49
74	113	355.00	10028.75	177	556.06	24605.74
50	114	358.14	10207.03	178	559.20	24884.56
82	115	361.28	10386 89	179	562.35	25164.94
72	116	364.42	10568.32	**180**	565.49	25446 90
18	117	367.57	10751.32	181	568.63	25730.43
22	118	370.71	10935.88	182	571.77	26015.53
88	119	373.85	11122.02	183	574.91	26302.20
01	**120**	376.99	11309.73	184	578.05	26590.44
76	121	380.13	11499.01	185	581.19	26880.25
08	122	383.27	11689.87	186	584.34	27171.63
97	123	386.42	11882.29	187	587.48	27464.59
3	124	389 56	12076.28	188	590.62	27759.11
7	125	392.70	12271.85	189	593.76	28055.21
	126	395.84	12468.98	**190**	596.90	28352.87
	127	398.98	12667.69	191	600.04	28652.11
	128	402.12	12867.96	192	603.19	28952.92

No.	Square.	Cube.	Sq. Root.	Cube Root.	No.	Square.	Cube.	Sq. Root.	Cube Root.
1420	2016400	2863288000	37.6829	11.2399	1475	2175625	3209046875	38.4057	11.
1421	2019241	2869341461	37.6962	11.2425	1476	2178576	3215578176	38.4187	11.
1422	2022084	2875403448	37.7094	11.2452	1477	2181529	3222118333	38.4318	11.
1423	2024929	2881473967	37.7227	11.2478	1478	2184484	3228667352	38.4448	11.
1424	2027776	2887553024	37.7359	11.2505	1479	2187441	3235225239	38.4578	11.
1425	2030625	2893640625	37.7492	11.2531	1480	2190400	3241792000	38.4708	11.
1426	2033476	2899736776	37.7624	11.2557	1481	2193361	3248367641	38.4838	11.
1427	2036329	2905841483	37.7757	11.2583	1482	2196324	3254952168	38.4968	11.
1428	2039184	2911954752	37.7889	11.2610	1483	2199289	3261545587	38.5097	11.
1429	2042041	2918076589	37.8021	11.2636	1484	2202256	3268147904	38.5227	11.
1430	2044900	2924207000	37.8153	11.2662	1485	2205225	3274759125	38.5357	11.
1431	2047761	2930345991	37.8286	11.2689	1486	2208196	3281379256	38.5487	11.
1432	2050624	2936493568	37.8418	11.2715	1487	2211169	3288008303	38.5616	11.
1433	2053489	2942649737	37.8550	11.2741	1488	2214144	3294646272	38.5746	11.
1434	2056356	2948814504	37.8682	11.2767	1489	2217121	3301293169	38.5876	11.
1435	2059225	2954987875	37.8814	11.2793	1490	2220100	3307949000	38.6005	11.
1436	2062096	2961169856	37.8946	11.2820	1491	2223081	3314613771	38.6135	11.
1437	2064969	2967360453	37.9078	11.2846	1492	2226064	3321287488	38.6264	11.
1438	2067844	2973559672	37.9210	11.2872	1493	2229049	3327970157	38.6394	11.
1439	2070721	2979767519	37.9342	11.2898	1494	2232036	3634661784	38.6523	11.
1440	2073600	2985984000	37.9473	11.2924	1495	2235025	3341362375	38.6652	11.
1441	2076481	2992209121	37.9605	11.2950	1496	2238016	3348071936	38.6782	11.
1442	2079364	2998442888	37.9737	11.2977	1497	2241009	3354790473	38.6911	11.
1443	2082249	3004685307	37.9868	11.3003	1498	2244004	3361517992	38.7040	11.
1444	2085136	3010936384	38.0000	11.3029	1499	2247001	3368254499	38.7169	11.
1445	2088025	3017196125	38.0132	11.3055	1500	2250000	3375000000	38.7298	11.
1446	2090916	3023464536	38.0263	11.3081	1501	2253001	3381754501	38.7427	11.
1447	2093809	3029741623	38.0395	11.3107	1502	2256004	3388518008	38.7556	11.
1448	2096704	3036027392	38.0526	11.3133	1503	2259009	3395290527	38.7685	11.
1449	2099601	3042321849	38.0657	11.3159	1504	2262016	3402072064	38.7814	11.
1450	2102500	3048625000	38.0789	11.3185	1505	2265025	3408862625	38.7943	11.
1451	2105401	3054936851	38.0920	11.3211	1506	2268036	3415662216	38.8072	11.
1452	2108304	3061257408	38.1051	11.3237	1507	2271049	3422470843	38.8201	11.
1453	2111209	3067586677	38.1182	11.3263	1508	2274064	3429288512	38.8330	11.
1454	2114116	3073924664	38.1314	11.3289	1509	2277081	3436115229	38.8458	11.
1455	2117025	3080271375	38.1445	11.3315	1510	2280100	3442951000	38.8587	11.
1456	2119936	3086626816	38.1576	11.3341	1511	2283121	3449795831	38.8716	11.
1457	2122849	3092990993	38.1707	11.3367	1512	2286144	3456649728	38.8844	11.
1458	2125764	3099363912	38.1838	11.3393	1513	2289169	3463512697	38.8973	11.
1459	2128681	3105745579	38.1969	11.3419	1514	2292196	3470384744	38.9102	11.
1460	2131600	3112136000	38.2099	11.3445	1515	2295225	3477265875	38.9230	11.
1461	2134521	3118535181	38.2230	11.3471	1516	2298256	3484156096	38.9358	11.
1462	2137444	3124943128	38.2361	11.3496	1517	2301289	3491055413	38.9487	11.
1463	2140369	3131359847	38.2492	11.3522	1518	2304324	3597963832	38.9615	11.
1464	2143296	3137785344	38.2623	11.3548	1519	2307361	3504881359	38.9744	11.
1465	2146225	3144219625	38.2753	11.3574	1520	2310400	3511808000	38.9872	11.
1466	2149156	3150662696	38.2884	11.3600	1521	2313441	3518743761	39.0000	11.
1467	2152089	3157114563	38.3014	11.3626	1522	2316484	3525688648	39.0128	11.
1468	2155024	3163575232	38.3145	11.3652	1523	2319529	3532642667	39.0256	11.
1469	2157961	3170044709	38.3275	11.3677	1524	2322576	3539605824	39.0384	11.
[illegible]	2160900	3176523000	38.3406	11.3703	1525	2325625	3546578125	39.0512	11.
[illegible]	[illegible]	[illegible]	38.3536	11.3729	1526	2328676	3553559576	39.0640	11.
[illegible]	[illegible]	[illegible]	38.3667	11.3755	1527	2331729	3560550183	39.0768	11.
[illegible]	[illegible]	[illegible]	38.3797	11.3780	1528	2334784	3567549952	39.0896	11.
[illegible]	[illegible]	[illegible]	38.3927	11.3806	1529	2337841	3574558889	39.1024	11.

Diam.	Circum.	Area.	Diam.	Circum.	Area.	Diam.	Circum.	Area.
394	1237.79	121922.07	461	1448.27	166913.60	528	1658.76	218956.44
395	1240.93	122541.75	462	1451.42	167638.53	529	1661.90	219786.61
396	1244.07	123163.00	463	1454.56	168365.02	530	1665.04	220618.34
397	1247.21	123785.82	464	1457.70	169093.08	531	1668.19	221451.65
398	1250.35	124410.21	465	1460.84	169822.72	532	1671.33	222286.53
399	1253.50	125036.17	466	1463.98	170553.92	533	1674.47	223122.98
400	1256.64	125663.71	467	1467.12	171286.70	534	1677.61	223961.00
401	1259.78	126292.81	468	1470.27	172021.05	535	1680.75	224800.59
402	1262.92	126923.48	469	1473.41	172756.97	536	1683.89	225641.75
403	1266.06	127555.73	470	1476.55	173494.45	537	1687.04	226484.48
404	1269.20	128189.55	471	1479.69	174233.51	538	1690.18	227328.79
405	1272.35	128824.93	472	1482.83	174974.14	539	1693.32	228174.66
406	1275.49	129461.89	473	1485.97	175716.35	540	1696.46	229022.10
407	1278.63	130100.42	474	1489.11	176460.12	541	1699.60	229871.12
408	1281.77	130740.52	475	1492.26	177205.46	542	1702.74	230721.71
409	1284.91	131382.19	476	1495.40	177952.37	543	1705.88	231573.86
410	1288.05	132025.43	477	1498.54	178700.86	544	1709.03	232427.59
411	1291.19	132670.24	478	1501.68	179450.91	545	1712.17	233282.89
412	1294.34	133316.63	479	1504.82	180202.54	546	1715.31	234139.76
413	1297.48	133964.58	480	1507.96	180955.74	547	1718.45	234998.20
414	1300.62	134614.10	481	1511.11	181710.50	548	1721.59	235858.21
415	1303.76	135265.20	482	1514.25	182466.84	549	1724.73	236719.79
416	1306.90	135917.86	483	1517.39	183224.75	550	1727.88	237582.94
417	1310.04	136572.10	484	1520.53	183984.23	551	1731.02	238447.67
418	1313.19	137227.91	485	1523.67	184745.28	552	1734.16	239313.96
419	1316.33	137885.29	486	1526.81	185507.90	553	1737.30	240181.83
420	1319.47	138544.24	487	1529.96	186272.10	554	1740.44	241051.26
421	1322.61	139204.76	488	1533.10	187037.86	555	1743.58	241922.27
422	1325.75	139866.85	489	1536.24	187805.19	556	1746.73	242794.85
423	1328.89	140530.51	490	1539.38	188574.10	557	1749.87	243068.99
424	1332.04	141195.74	491	1542.52	189344.57	558	1753.01	244544.71
425	1335.18	141862.54	492	1545.66	190116.62	559	1756.15	245422.00
426	1338.32	142530.92	493	1548.81	190890.24	560	1759.29	246300.86
427	1341.46	143200.86	494	1551.95	191665.43	561	1762.43	247181.30
428	1344.60	143872.38	495	1555.09	192442.18	562	1765.58	248063.30
429	1347.74	144545.46	496	1558.23	193220.51	563	1768.72	248946.87
[illegible]	1350.88	145220.12	497	1561.37	194000.41	564	1771.86	249832.01
[illegible]	1354.03	145896.35	498	1564.51	194781.89	565	1775.00	250718.73
[illegible]	1357.17	146574.15	499	1567.65	195564.93	566	1778.14	251607.01
[illegible]	1360.31	147253.52	500	1570.80	196349.54	567	1781.28	252496.87
[illegible]	1363.45	147934.46	501	1573.94	197135.72	568	1784.42	253388.30
[illegible]	1366.59	148616.97	502	1577.08	197923.48	569	1787.57	254281.29
[illegible]	1369.73	149301.05	503	1580.22	198712.80	570	1790.71	255175.86
[illegible]	1372.88	149986.70	504	1583.36	199503.70	571	1793.85	256072.00
[illegible]	1376.02	150673.93	505	1586.50	200296.17	572	1796.99	256969.71
[illegible]	1379.16	151362.72	506	1589.65	201090.20	573	1800.13	257868.99
[illegible]	1382.30	152053.08	507	1592.79	201885.81	574	1803.27	258769.85
[illegible]	1385.44	152745.02	508	1595.93	202682.99	575	1806.42	259672.27
[illegible]	1388.58	153438.53	509	1599.07	203481.74	576	1809.56	260576.26
[illegible]	1391.73	154133.60	510	1602.21	204282.06	577	1812.70	261481.83
[illegible]	1394.87	154830.25	511	1605.35	205083.95	578	1815.84	262388.96
[illegible]	1398.01	155528.47	512	1608.50	205887.42	579	1818.98	263297.67
[illegible]	1401.15	156228.26	513	1611.64	206692.45	580	1822.12	264207.94
[illegible]	1404.29	156929.62	514	1614.78	207499.05	581	1825.27	265119.79
[illegible]	1407.43	157632.55	515	1617.92	208307.23	582	1828.41	266033.21
[illegible]	1410.58	158337.06	516	1621.06	209116.97	583	1831.55	266948.20
[illegible]	1413.72	159043.13	517	1624.20	209928.29	584	1834.69	267864.76
[illegible]	1416.86	159750.77	518	1627.34	210741.18	585	1837.83	268782.89
[illegible]	1420.00	160459.99	519	1630.49	211555.63	586	1840.97	269702.59
[illegible]	1423.14	161170.77	520	1633.63	212371.66	587	1844.11	270623.86
[illegible]	1426.28	161883.13	521	1636.77	213189.26	588	1847.26	271546.70
[illegible]	[illegible]29.42	162597.05	522	1639.91	214008.43	589	1850.40	272471.12
[illegible]	[illegible]2.57	163312.55	523	1643.05	214829.17	590	1853.54	273397.10
[illegible]	[illegible].71	164029.62	524	1646.19	215651.49	591	1856.68	274324.66
[illegible]	[illegible]85	164748.26	525	1649.34	216475.37	592	1859.82	275253.78
[illegible]	[illegible]99	165468.47	526	1652.48	217300.82	593	1862.96	276184.4[illegible]
[illegible]	[illegible]3	166190.25	527	1655.62	218127.85	594	1866.11	277116.7[illegible]

FIFTH ROOTS AND FIFTH POWE[RS]

(Abridged from TRAUTWINE.)

No. or Root.	Power.	No. or Root.	Power.	No. or Root.	Power.	No. or Root.	Power.
.10	.000010	3.7	693.440	9.8	90392	21.8	4923597
.15	.000075	3.8	792.352	9.9	95099	22.0	5153632
.20	.000320	3.9	902.242	10.0	100000	22.2	5392186
.25	.000977	4.0	1024.00	10.2	110408	22.4	5639493
.30	.002430	4.1	1158.56	10.4	121665	22.6	5895793
.35	.005252	4.2	1306.91	10.6	133823	22.8	6161327
.40	.010240	4.3	1470.08	10.8	146933	23.0	6436343
.45	.018453	4.4	1649.16	11.0	161051	23.2	6721093
.50	.031250	4.5	1845.28	11.2	176234	23.4	7015834
.55	.050328	4.6	2059.63	11.4	192541	23.6	7320825
.60	.077760	4.7	2293.45	11.6	210034	23.8	7636332
.65	.116029	4.8	2548.04	11.8	228776	24.0	7962624
.70	168070	4.9	2824.75	12.0	248832	24.2	8299976
.75	.237305	5.0	3125.00	12.2	270271	24.4	8648666
.80	.327680	5.1	3450.25	12.4	293163	24.6	9008978
.85	.443705	5.2	3802.04	12.6	317580	24.8	9381200
.90	.590490	5.3	4181.95	12.8	343597	25.0	9765625
.95	.773781	5.4	4591.65	13.0	371293	25.2	10162550
1.00	1.00000	5.5	5032.84	13.2	400746	25.4	10572278
1.05	1.27628	5.6	5507.32	13.4	432040	25.6	10995116
1.10	1.61051	5.7	6016.92	13.6	465259	25.8	11431377
1.15	2.01135	5.8	6563.57	13.8	500490	26.0	11881376
1.20	2.48832	5.9	7149.24	14.0	537824	26.2	12345437
1.25	3.05176	6.0	7776.00	14.2	577353	26.4	12823886
1.30	3.71293	6.1	8445.96	14.4	619174	26.6	13317055
1.35	4.48403	6.2	9161.33	14.6	663383	26.8	13825281
1.40	5.37824	6.3	9924.37	14.8	710082	27.0	14348907
1.45	6.40973	6.4	10737	15.0	759375	27.2	14888280
1.50	7.59375	6.5	11603	15.2	811368	27.4	15443752
1.55	8.94661	6.6	12523	15.4	866171	27.6	16015681
1.60	10.4858	6.7	13501	15.6	923896	27.8	16604430
1.65	12.2298	6.8	14539	15.8	984658	28.0	17210368
1.70	14.1986	6.9	15640	16.0	1048576	28.2	17833868
1.75	16.4131	7.0	16807	16.2	1115771	28.4	18475309
1.80	18.8957	7.1	18042	16.4	1186367	28.6	19135075
1.85	21.6700	7.2	19349	16.6	1260493	28.8	19813557
1.90	24.7610	7.3	20731	16.8	1338278	29.0	20511149
1.95	28.1951	7.4	22190	17.0	1419857	29.2	21228253
2.00	32.0000	7.5	23730	17.2	1505366	29.4	21965275
2.05	36.2051	7.6	25355	17.4	1594947	29.6	22722628
2.10	40.8410	7.7	27068	17.6	1688742	29.8	23500728
2.15	45.9401	7.8	28872	17.8	1786899	30.0	24300000
2.20	51.5363	7.9	30771	18.0	1889568	30.5	26393634
2.25	57.6650	8.0	32768	18.2	1996903	31.0	28629151
2.30	64.3634	8.1	34868	18.4	2109061	31.5	31013642
2.35	71.6703	8.2	37074	18.6	2226203	32.0	33554432
2.40	79.6262	8.3	39390	18.8	2348493	32.5	36259082
2.45	88.2735	8.4	41821	19.0	2476099	33.0	39135393
2.50	97.6562	8.5	44371	19.2	2609193	33.5	42191410
2.55	107.820	8.6	47043	19.4	2747949	34.0	45435424
2.60	118.814	8.7	49842	19.6	2892547	34.5	48875980
2.70	143.489	8.8	52773	19.8	3043168	35.0	52521875
2.80	172.104	8.9	55841	20.0	3200000	35.5	56382167
2.90	205.111	9.0	59049	20.2	3363232	36.0	60466176
3.00	243.000	9.1	62403	20.4	3533059	36.5	64783487
3.10	286.292	9.2	65908	20.6	3709677	37.0	69343957
3.20	335.544	9.3	69569	20.8	3893289	37.5	74157715
[illegible]	391.354	9.4	73390	21.0	4084101	38.0	79235168
[illegible]	[illegible]54.254	9.5	77378	21.2	4282322	38.5	84587005
[illegible]	[illegible]5.219			21.4	4488166	39.0	90224199
[illegible]	[illegible]			21.6	4701850	39.5	96158012

MFERENCES AND AREAS OF CIRCLES.

Area.	Diam.	Circum.	Area.	Diam.	Circum.	Area.
0.7854	65	204.20	3318.31	129	405.27	13069.81
3.1416	66	207.34	3421.19	**130**	408.41	13273.23
7.0686	67	210.49	3525.65	131	411.55	13478.22
12.5664	68	213.63	3631.68	132	414.69	13684 78
19.635	69	216.77	3739.28	133	417.83	13892.91
28 274	**70**	219.91	3848.45	134	420.97	14102.61
38.485	71	223.05	3959.19	135	424.12	14313.88
50.266	72	226.19	4071.50	136	427.26	14526.72
63.617	73	229.34	4185.39	137	430.40	14741.14
78.540	74	232.48	4300.84	138	433.54	14957.12
95.033	75	235.62	4417.86	139	436.68	15174.68
113.10	76	238.76	4536.46	**140**	439.82	15393.80
132.73	77	241.90	4656.63	141	442.96	15614.50
153.94	78	245.04	4778.36	142	446.11	15836.77
176.71	79	248.19	4901.67	143	449.25	16060.61
201.06	**80**	251.33	5026.55	144	452.39	16286.02
226.98	81	254.47	5153.00	145	455.53	16513.00
254.47	82	257.61	5281.02	146	458.67	16741.55
283.53	83	260.75	5410.61	147	461.81	16971.67
314.16	84	263.89	5541.77	148	464.96	17203.36
346.36	85	267.04	5674 50	149	468.10	17436 62
380.13	86	270.18	5808.80	**150**	471.24	17671.46
415.48	87	273.32	5944.68	151	474.38	17907 86
452.39	88	276.46	6082.12	152	477.52	18145.84
490.87	89	279.60	6221.14	153	480.66	18385.39
530.93	**90**	282.74	6361.73	154	483.81	18626.50
572.56	91	285.88	6503.88	155	486.95	18869.19
615.75	92	289.03	6647.61	156	490.09	19113.45
660.52	93	292.17	6792.91	157	493.23	19359.28
706.86	94	295.31	6939.78	158	496.37	19606.68
754.77	95	298.45	7088.22	159	499.51	19855.65
804.25	96	301.59	7238.23	**160**	502.65	20106.19
855.30	97	304.73	7389.81	161	505.80	20358.31
907.92	98	307.88	7542.96	162	508.94	20611.99
962.11	99	311.02	7697.69	163	512.08	20867.24
1017.88	**100**	314.16	7853.98	164	515.22	21124 07
1075.21	101	317.30	8011.85	165	518.36	21382.46
1134.11	102	320.44	8171.28	166	521.50	21642.43
1194.59	103	323.58	8332.29	167	524.65	21903 97
1256.64	104	326.73	8494.87	168	527.79	22167 08
1320 25	105	329 87	8659.01	169	530.93	22431.76
1385.44	106	333.01	8824.73	**170**	534.07	22698.01
1452.20	107	336.15	8992.02	171	537.21	22965.83
1520.53	108	339.29	9160.88	172	540.35	23235.22
1590.43	109	342.43	9331.32	173	543.50	23506.18
1661.96	**110**	345.58	9503.32	174	546.64	23778.71
1734.94	111	348.72	9676.89	175	549 78	24052.82
1809.56	112	351.86	9852.03	176	552.92	24328.49
1885.74	113	355.00	10028.75	177	556.06	24605.74
1963.50	114	358.14	10207.03	178	559.20	24884.56
2042.82	115	361.28	10386 89	179	562.35	25164.94
2123.72	116	364.42	10568.32	**180**	565.49	25446 90
2206.18	117	367.57	10751.32	181	568.63	25730.43
2290 22	118	370.71	10935.88	182	571.77	26015.53
2375.83	119	373.85	11122.02	183	574.91	26302.20
2463.01	**120**	376.99	11309.73	184	578.05	26590.44
2551.76	121	380.13	11499.01	185	581.19	26880.25
2642.08	122	383.27	11689.87	186	584.34	27171.63
2733.97	123	386.42	11882.29	187	587.48	27464.59
2827.43	124	389.56	12076.28	188	590.62	27759.11
2922.47	125	392.70	*12271.85*	189	593.76	28055.21
3019.07	*126*	*395.84*	*12468.98*	**190**	596.90	28352.87
3117.25	*127*	*398.98*	*12667.69*	191	600.04	28652.[illegible]
3216.99	*128*	*402.12*	*12867.96*	192	603.19	2895[illegible]

Diam.	Circum.	Area.	Diam.	Circum.	Area.	Diam.	Circum
193	606.33	29255.30	**260**	816.81	53092.92	327	1027.
194	609.47	29559.25	261	819.96	53502.11	328	1030.
195	612.61	29864.77	262	823.10	53912.87	329	1033.
196	615.75	30171.86	263	826.24	54325.21	**330**	1036.
197	618.89	30480.52	264	829.38	54739.11	331	1039.
198	622.04	30790.75	265	832.52	55154.59	332	1043.
199	625.18	31102.55	266	835.66	55571.63	333	1046.
200	628.32	31415.93	267	838.81	55990.25	334	1049.
201	631.46	31730.87	268	841.95	56410.44	335	1052.
202	634.60	32047.39	269	845.09	56832.20	336	1055.
203	637.74	32365.47	**270**	848.23	57255.53	337	1058
204	640.88	32685.13	271	851.37	57680.43	338	1061
205	644.03	33006.36	272	854.51	58106.90	339	1065
206	647.17	33329.16	273	857.65	58534.94	**340**	1068
207	650.31	33653.53	274	860.80	58964.55	341	1071
208	653.45	33979.47	275	863.94	59395.74	342	1074
209	656.59	34306.98	276	867.08	59828.49	343	1077
210	659.73	34636.06	277	870.22	60262.82	344	108
211	662.88	34966.71	278	873.36	60698.71	345	108
212	666.02	35298.94	279	876.50	61136.18	346	108
213	669.16	35632.73	**280**	879.65	61575.22	347	109
214	672.30	35968.09	281	882.79	62015.82	348	109
215	675.44	36305.03	282	885.93	62458.00	349	109
216	678.58	36643.54	283	889.07	62901.75	**350**	109
217	681.73	36983.61	284	892.21	63347.07	351	110
218	684.87	37325.26	285	895.35	63793.97	352	110
219	688.01	37668.48	286	898.50	64242.43	353	110
220	691.15	38013.27	287	901.64	64692.46	354	111
221	694.29	38359.63	288	904.78	65144.07	355	111
222	697.43	38707.56	289	907.92	65597.24	356	111
223	700.58	39057.07	**290**	911.06	66051.99	357	112
224	703.72	39408.14	291	914.20	66508.30	358	112
225	706.86	39760.78	292	917.35	66966.19	359	112
226	710.00	40115.00	293	920.49	67425.65	**360**	113
227	713.14	40470.78	294	923.63	67886.68	361	113
228	716.28	40828.14	295	926.77	68349.28	362	113
229	719.42	41187.07	296	929.91	68813.45	363	114
230	722.57	41547.56	297	933.05	69279.19	364	114
231	725.71	41909.63	298	936.19	69746.50	365	114
232	728.85	42273.27	299	939.34	70215.38	366	114
233	731.99	42638.48	**300**	942.48	70685.83	367	115
234	735.13	43005.26	301	945.62	71157.86	368	115
235	738.27	43373.61	302	948.76	71631.45	369	115
236	741.42	43743.54	303	951.90	72106.62	**370**	116
237	744.56	44115.03	304	955.04	72583.36	371	116
238	747.70	44488.09	305	958.19	73061.66	372	116
239	750.84	44862.73	306	961.33	73541.54	373	117
240	753.98	45238.93	307	964.47	74022.99	374	117
241	757.12	45616.71	308	967.61	74506.01	375	117
242	760.27	45996.06	309	970.75	74990.60	376	118
243	763.41	46376.98	**310**	973.89	75476.76	377	118
244	766.55	46759.47	311	977.04	75964.50	378	118
245	769.69	47143.52	312	980.18	76453.80	379	119
246	772.83	47529.16	313	983.32	76944.67	**380**	119
247	775.97	47916.36	314	986.46	77437.12	381	119
248	779.11	48305.13	315	989.60	77931.13	382	120
249	782.26	48695.47	316	992.74	78426.72	383	120
250	785.40	49087.39	317	995.88	78923.88	384	120
251	788.54	49480.87	318	999.03	79422.60	385	12
252	791.68	49875.92	319	1002.17	79922.90	386	121
253	794.82	50272.55	**320**	1005.31	80424.77	387	121
254	797.96	50670.75	321	1008.45	80928.21	388	121
255	801.11	51070.52	322	1011.59	81433.22	389	12
256	804.25	51471.85	323	1014.73	81939.80	**390**	12
[illegible]	[illegible]	51874.76	324	1017.88	82447.96	391	12
[illegible]	[illegible]	52279.24	325	1021.02	82957.68	392	12
[illegible]	[illegible]	52685.29	326	1024.16	83468.98	393	12

am.	Circum.	Area.	Diam.	Circum.	Area.	Diam.	Circum.	Area.
	42.804	145.80	21 ⅞	68.722	375.83	30 ⅛	94.640	712.76
	43.197	148.49	22.	69.115	380.13	¼	95.033	718.69
	43.590	151.20	⅛	69.508	384.46	⅜	95.426	724.64
	43.982	153.94	¼	69.900	388.82	½	95.819	730.62
	44.375	156.70	⅜	70.293	393.20	⅝	96.211	736.62
	44.768	159.48	½	70.686	397.61	¾	96.604	742.64
	45.160	162.30	⅝	71.079	402.04	⅞	96.997	748.69
	45.553	165.13	¾	71.471	406.49	31.	97.389	754.77
	45.946	167.99	⅞	71.864	410.97	⅛	97.782	760.87
	46.338	170.87	23.	72.257	415.48	¼	98.175	766.99
	46.731	173.78	⅛	72.649	420.00	⅜	98.567	773.14
	47.124	176.71	¼	73.042	424.56	½	98.960	779.31
	47.517	179.67	⅜	73.435	429.13	⅝	99.353	785.51
	47.909	182.65	½	73.827	433.74	¾	99.746	791.73
	48.302	185.66	⅝	74.220	438.36	⅞	100.138	797.98
	48.695	188.69	¾	74.613	443.01	32.	100.531	804.25
	49.087	191.75	⅞	75.006	447.69	⅛	100.924	810.54
	49.480	194.83	24.	75.398	452.39	¼	101.316	816.86
	49.873	197.93	⅛	75.791	457.11	⅜	101.709	823.21
	50.265	201.06	¼	76.184	461.86	½	102.102	829.58
	50.658	204.22	⅜	76.576	466.64	⅝	102.494	835.97
	51.051	207.39	½	76.969	471.44	¾	102.887	842.39
	51.444	210.60	⅝	77.362	476.26	⅞	103.280	848.83
	51.836	213.82	¾	77.754	481.11	33.	103.673	855.30
	52.229	217.08	⅞	78.147	485.98	⅛	104.065	861.79
	52.622	220.35	25.	78.540	490.87	¼	104.458	868.31
	53.014	223.65	⅛	78.933	495.79	⅜	104.851	874.85
	53.407	226.98	¼	79.325	500.74	½	105.243	881.41
	53.800	230.33	⅜	79.718	505.71	⅝	105.636	888.00
	54.192	233.71	½	80.111	510.71	¾	106.029	894.62
	54.585	237.10	⅝	80.503	515.72	⅞	106.421	901.26
	54.978	240.53	¾	80.896	520.77	34.	106.814	907.92
	55.371	243.98	⅞	81.289	525.84	⅛	107.207	914.61
	55.763	247.45	26.	81.681	530.93	¼	107.600	921.32
	56.156	250.95	⅛	82.074	536.05	⅜	107.992	928.06
	56.549	254.47	¼	82.467	541.19	½	108.385	934.82
	56.941	258.02	⅜	82.860	546.35	⅝	108.778	941.61
	57.334	261.59	½	83.252	551.55	¾	109.170	948.42
	57.727	265.18	⅝	83.645	556.76	⅞	109.563	955.25
	58.119	268.80	¾	84.038	562.00	35.	109.956	962.11
	58.512	272.45	⅞	84.430	567.27	⅛	110.348	969.00
	58.905	276.12	27.	84.823	572.56	¼	110.741	975.91
	59.298	279.81	⅛	85.216	577.87	⅜	111.134	982.84
	59.690	283.53	¼	85.608	583.21	½	111.527	989.80
	60.083	287.27	⅜	86.001	588.57	⅝	111.919	996.78
	60.476	291.04	½	86.394	593.96	¾	112.312	1003.8
	60.868	294.83	⅝	86.786	599.37	⅞	112.705	1010.8
	61.261	298.65	¾	87.179	604.81	36.	113.097	1017.9
	61.654	302.49	⅞	87.572	610.27	⅛	113.490	1025.0
	62.046	306.35	28.	87.965	615.75	¼	113.883	1032.1
	62.439	310.24	⅛	88.357	621.26	⅜	114.275	1039.2
	62.832	314.16	¼	88.750	626.80	½	114.668	1046.3
	63.225	318.10	⅜	89.143	632.36	⅝	115.061	1053.5
	63.617	322.06	½	89.535	637.94	¾	115.454	1060.7
	64.010	326.05	⅝	89.928	643.55	⅞	115.846	1068.0
	64.403	330.06	¾	90.321	649.18	37.	116.239	1075.2
	64.795	334.10	⅞	90.713	654.81	⅛	116.632	1082.5
	65.188	338.16	29.	91.106	660.52	¼	117.024	1089.8
	65.581	342.25	⅛	91.499	666.23	⅜	117.417	1097.1
	65.973	346.36	¼	91.892	671.96	½	117.810	1104.5
	66.366	350.50	⅜	92.284	677.71	⅝	118.202	1111.8
	66.759	354.66	½	92.677	683.49	¾	118.596	1119.2
	7.152	358.84	⅝	93.070	689.30	⅞	118.988	1126.7
	7.544	363.05	¾	93.462	695.13	38.	119.381	1134.1
	.937	367.28	⅞	93.855	700.98	⅛	119.773	1141.
	330	371.54	30.	94.248	706.86	¼	120.166	1149.

Diam.	Circum.	Area.	Diam.	Circum.	Area.	Diam.	Circum.
595	1869.25	278050.58	663	2082.88	345236.69	731	2296.50
596	1872.39	278985.99	664	2086.02	346278.91	732	2299.65
597	1875.53	279922.97	665	2089.16	347322.70	733	2302.79
598	1878.67	280861.52	666	2092.30	348368.07	734	2305.93
599	1881.81	281801.65	667	2095.44	349415.00	735	2309.07
600	1884.96	282743.34	668	2098.58	350463.51	736	2312.21
601	1888.10	283686.60	669	2101.73	351513.59	737	2315.35
602	1891.24	284631.44	**670**	2104.87	352565.24	738	2318.50
603	1894.38	285577.84	671	2108.01	353618.45	739	2321.64
604	1897.52	286525.82	672	2111.15	354673.24	**740**	2324.78
605	1900.66	287475.36	673	2114.29	355729.60	741	2327.92
606	1903.81	288426.48	674	2117.43	356787.54	742	2331.06
607	1906.95	289379.17	675	2120.58	357847.04	743	2334.20
608	1910.09	290333.43	676	2123.72	358908.11	744	2337.34
609	1913.23	291289.26	677	2126.86	359970.75	745	2340.49
610	1916.37	292246.66	678	2130.00	361034.97	746	2343.63
611	1919.51	293205.63	679	2133.14	362100.75	747	2346.77
612	1922.65	294166.17	**680**	2136.28	363168.11	748	2349.91
613	1925.80	295128.28	681	2139.42	364237.04	749	2353.05
614	1928.94	296091.97	682	2142.57	365307.54	**750**	2356.19
615	1932.08	297057.22	683	2145.71	366379.60	751	2359.34
616	1935.22	298024.05	684	2148.85	367453.24	752	2362.48
617	1938.36	298992.44	685	2151.99	368528.45	753	2365.62
618	1941.50	299962.41	686	2155.13	369605.23	754	2368.76
619	1944.65	300933.95	687	2158.27	370683.59	755	2371.90
620	1947.79	301907.05	688	2161.42	371763.51	756	2375.04
621	1950.93	302881.73	689	2164.56	372845.00	757	2378.19
622	1954.07	303857.98	**690**	2167.70	373928.07	758	2381.33
623	1957.21	304835.80	691	2170.84	375012.70	759	2384.47
624	1960.35	305815.20	692	2173.98	376098.91	**760**	2387.61
625	1963.50	306796.16	693	2177.12	377186.68	761	2390.75
626	1966.64	307778.69	694	2180.27	378276.03	762	2393.89
627	1969.78	308762.79	695	2183.41	379366.95	763	2397.04
628	1972.92	309748.47	696	2186.55	380459.44	764	2400.18
629	1976.06	310735.71	697	2189.69	381553.50	765	2403.32
630	1979.20	311724.53	698	2192.83	382649.13	766	2406.46
631	1982.35	312714.92	699	2195.97	383746.33	767	2409.60
632	1985.49	313706.88	**700**	2199.11	384845.10	768	2412.74
633	1988.63	314700.40	701	2202.26	385945.44	769	2415.88
634	1991.77	315695.50	702	2205.40	387047.36	**770**	2419.03
635	1994.91	316692.17	703	2208.54	388150.84	771	2422.17
636	1998.05	317690.42	704	2211.68	389255.90	772	2425.31
637	2001.19	318690.23	705	2214.82	390362.52	773	2428.45
638	2004.34	319691.61	706	2217.96	391470.72	774	2431.59
639	2007.48	320694.56	707	2221.11	392580.49	775	2434.73
640	2010.62	321699.09	708	2224.25	393691.82	776	2437.88
641	2013.76	322705.18	709	2227.39	394804.73	777	2441.02
642	2016.90	323712.85	**710**	2230.53	395919.21	778	2444.16
643	2020.04	324722.09	711	2233.67	397035.26	779	2447.30
644	2023.19	325732.89	712	2236.81	398152.89	**780**	2450.44
645	2026.33	326745.27	713	2239.96	399272.08	781	2453.58
646	2029.47	327759.22	714	2243.10	400392.84	782	2456.73
647	2032.61	328774.74	715	2246.24	401515.18	783	2459.87
648	2035.75	329791.83	716	2249.38	402639.08	784	2463.01
649	2038.89	330810.49	717	2252.52	403764.56	785	2466.15
650	2042.04	331830.72	718	2255.66	404891.60	786	2469.29
651	2045.18	332852.58	719	2258.81	406020.22	787	2472.43
652	2048.32	333875.90	**720**	2261.95	407150.41	788	2475.58
653	2051.46	334900.85	721	2265.09	408282.17	789	2478.72
654	2054.60	335927.36	722	2268.23	409415.50	**790**	2481.86
655	2057.74	336955.45	723	2271.37	410550.40	791	2485.00
656	2060.88	337985.10	724	2274.51	411686.87	792	2488.14
657	[illegible].03	339016.33	725	2277.65	412824.91	793	2491.28
[illegible]	[illegible]	[illegible]0049.13	726	2280.80	413964.52	794	2494.42
[illegible]	[illegible]	[illegible]083.50	727	2283.94	415105.71	795	2497.57
[illegible]	[illegible]	[illegible]19.44	728	2287.08	416248.46	796	2500.71
[illegible]	[illegible]	[illegible]6.95	729	2290.22	417392.79	797	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

Diam.	Circum.	Area.	Diam.	Circum.	Area.
71 3/8	224.231	4001.1	79 5/8	250.149	4979.5
1/2	224.624	4015.2	3/4	250.542	4995.2
5/8	225.017	4029.2	7/8	250.935	5010.9
3/4	225.409	4043.3	80.	251.327	5026.5
7/8	225.802	4057.4	1/8	251.720	5042.3
72.	226.195	4071.5	1/4	252.113	5058.0
1/8	226.587	4085.7	3/8	252.506	5073.8
1/4	226.980	4099.8	1/2	252.898	5089.6
3/8	227.373	4114.0	5/8	253.291	5105.4
1/2	227.765	4128.2	3/4	253.684	5121.2
5/8	228.158	4142.5	7/8	254.076	5137.1
3/4	228.551	4156.8	81.	254.469	5153.0
7/8	228.944	4171.1	1/8	254.862	5168.9
73.	229.336	4185.4	1/4	255.254	5184.9
1/8	229.729	4199.7	3/8	255.647	5200.8
1/4	230.122	4214.1	1/2	256.040	5216.8
3/8	230.514	4228.5	5/8	256.433	5232.8
1/2	230.907	4242.9	3/4	256.825	5248.9
5/8	231.300	4257.4	7/8	257.218	5264.9
3/4	231.692	4271.8	82.	257.611	5281.0
7/8	232.085	4286.3	1/8	258.003	5297.1
74.	232.478	4300.8	1/4	258.396	5313.3
1/8	232.871	4315.4	3/8	258.789	5329.4
1/4	233.263	4329.9	1/2	259.181	5345.6
3/8	233.656	4344.5	5/8	259.574	5361.8
1/2	234.049	4359.2	3/4	259.967	5378.1
5/8	234.441	4373.8	7/8	260.359	5394.3
3/4	234.834	4388.5	83.	260.752	5410.6
7/8	235.227	4403.1	1/8	261.145	5426.9
75.	235.619	4417.9	1/4	261.538	5443.3
1/8	236.012	4432.6	3/8	261.930	5459.6
1/4	236.405	4447.4	1/2	262.323	5476.0
3/8	236.798	4462.2	5/8	262.716	5492.4
1/2	237.190	4477.0	3/4	263.108	5508.8
5/8	237.583	4491.8	7/8	263.501	5525.3
3/4	237.976	4506.7	84.	263.894	5541.8
7/8	238.368	4521.5	1/8	264.286	5558.3
76.	238.761	4536.5	1/4	264.679	5574.8
1/8	239.154	4551.4	3/8	265.072	5591.4
1/4	239.546	4566.4	1/2	265.465	5607.9
3/8	239.939	4581.3	5/8	265.857	5624.5
1/2	240.332	4596.3	3/4	266.250	5641.2
5/8	240.725	4611.4	7/8	266.643	5657.8
3/4	241.117	4626.4	85.	267.035	5674.5
7/8	241.510	4641.5	1/8	267.428	5691.2
77.	241.903	4656.6	1/4	267.821	5707.9
1/8	242.295	4671.8	3/8	268.213	5724.7
1/4	242.688	4686.9	1/2	268.606	5741.5
3/8	243.081	4702.1	5/8	268.999	5758.3
1/2	243.473	4717.3	3/4	269.392	5775.1
5/8	243.866	4732.5	7/8	269.784	5791.9
3/4	244.259	4747.8	86.	270.177	5808.8
7/8	244.652	4763.1	1/8	270.570	5825.7
78.	245.044	4778.4	1/4	270.962	5842.6
1/8	245.437	4793.7	3/8	271.355	5859.6
1/4	245.830	4809.0	1/2	271.748	5876.5
3/8	246.222	4824.4	5/8	272.140	5893.5
1/2	246.615	4839.8	3/4	272.533	5910.6
5/8	247.008	4855.2	7/8	272.926	5927.6
3/4	247.400	4870.7	87.	273.319	5944.7
7/8	247.793	4886.2	1/8	273.711	5961.8
79.	248.186	4901.7	1/4	274.104	5978.9
1/8	248.579	4917.2	3/8	274.497	5996.0
1/4	248.971	4932.7	1/2	274.889	6013.2
3/8	249.364	4948.3	5/8	275.282	6030.4
1/2	249.757	4963.9	3/4	275.675	6047.6

CIRCUMFERENCES AND AREAS OF CIR

Advancing by Eighths.

Diam.	Circum.	Area.	Diam.	Circum.	Area.
1/64	.04909	.00019	2 3/8	7.4613	4.4301
1/32	.09818	.00077	7/16	7.6576	4.6664
3/64	.14726	.00173	1/2	7.8540	4.9087
1/16	.19635	.00307	9/16	8.0503	5.1572
3/32	.29452	.00690	5/8	8.2467	5.4119
1/8	.39270	.01227	11/16	8.4430	5.6727
5/32	.49087	.01917	3/4	8.6394	5.9396
3/16	.58905	.02761	13/16	8.8357	6.2126
7/32	.68722	.03758	7/8	9.0321	6.4918
			15/16	9.2284	6.7771
1/4	.78540	.04909			
9/32	.88357	.06213	3.	9.4248	7.0686
5/16	.98175	.07670	1/16	9.6211	7.3662
11/32	1.0799	.09281	1/8	9.8175	7.6699
3/8	1.1781	.11045	3/16	10.014	7.9798
13/32	1.2763	.12962	1/4	10.210	8.2958
7/16	1.3744	.15033	5/16	10.407	8.6179
15/32	1.4726	.17257	3/8	10.603	8.9462
			7/16	10.799	9.2806
1/2	1.5708	.19635	1/2	10.996	9.6211
17/32	1.6690	.22166	9/16	11.192	9.9678
9/16	1.7671	.24850	5/8	11.388	10.321
19/32	1.8653	.27688	11/16	11.585	10.680
5/8	1.9635	.30680	3/4	11.781	11.045
21/32	2.0617	.33824	13/16	11.977	11.416
11/16	2.1598	.37122	7/8	12.174	11.793
23/32	2.2580	.40574	15/16	12.370	12.177
			4.	12.566	12.566
3/4	2.3562	.44179	1/16	12.763	12.962
25/32	2.4544	.47937	1/8	12.959	13.364
13/16	2.5525	.51849	3/16	13.155	13.772
27/32	2.6507	.55914	1/4	13.352	14.186
7/8	2.7489	.60132	5/16	13.548	14.607
29/32	2.8471	.64504	3/8	13.744	15.033
15/16	2.9452	.69029	7/16	13.941	15.466
31/32	3.0434	.73708	1/2	14.137	15.904
			9/16	14.334	16.349
1.	3.1416	.7854	5/8	14.530	16.800
1/16	3.3379	.8866	11/16	14.726	17.257
1/8	3.5343	.9940	3/4	14.923	17.728
3/16	3.7306	1.1075	13/16	15.119	18.190
1/4	3.9270	1.2272	7/8	15.315	18.665
5/16	4.1233	1.3530	15/16	15.512	19.147
3/8	4.3197	1.4849	5.	15.708	19.635
7/16	4.5160	1.6230	1/16	15.904	20.129
1/2	4.7124	1.7671	1/8	16.101	20.629
9/16	4.9087	1.9175	3/16	16.297	21.135
5/8	5.1051	2.0739	1/4	16.493	21.648
11/16	5.3014	2.2365	5/16	16.690	22.166
3/4	5.4978	2.4053	3/8	16.886	22.691
13/16	5.6941	2.5802	7/16	17.082	23.221
7/8	5.8905	2.7612	1/2	17.279	23.758
15/16	6.0868	2.9483	9/16	17.475	24.301
			5/8	17.671	24.850
[illegible]	6.2832	3.1416	11/16	17.868	25.406
[illegible]	[illegible]	3.3410	3/4	18.064	25.967
		[illegible]	13-16	18.261	26.535
			7/8	18.457	27.109
			15-16	18.653	27.688
			6	18.850	28.274

Diam.	Circ
6 1/8	19.
1/4	19.
3/8	20.
1/2	20.
5/8	20.
3/4	21
7/8	21.
7.	21.
1/8	22.
1/4	22.
3/8	23.
1/2	23.
5/8	23.
3/4	24.
7/8	24.
8.	25.
1/8	25.
1/4	25.
3/8	26.
1/2	26.
5/8	27.
3/4	27.
7/8	27.
9.	28.
1/8	28.
1/4	29.
3/8	29.
1/2	29.
5/8	30.
3/4	30.
7/8	31.
10.	31
1/8	31.
1/4	32.
3/8	32.
1/2	32
5/8	33.
3/4	33.
7/8	34.
11.	34.
1/8	34.
1/4	35.
3/8	35.
1/2	36.
5/8	36.
3/4	36.
7/8	37.
12.	37.
1/8	38.
1/4	38.
3/8	38.
1/2	39.
5/8	39.
3/4	40.
7/8	40.
13.	40.
1/8	41.
1/4	41
3/8	4
1/2	4

Circum.	Area.	Diam.	Circum.	Area.	Diam.	Circum.	Area.
42.804	145.80	21 ⅞	68.722	375.83	30 ⅛	94.640	712.76
43.197	148.49	22.	69.115	380.13	¼	95.033	718.69
43.590	151.20	⅛	69.508	384.46	⅜	95.426	724.64
43.982	153.94	¼	69.900	388.82	½	95.819	730.62
44.375	156.70	⅜	70.293	393.20	⅝	96.211	736.62
44.768	159.48	½	70.686	397.61	¾	96.604	742.64
45.160	162.30	⅝	71.079	402.04	⅞	96.997	748.69
45.553	165.13	¾	71.471	406.49	31.	97.389	754.77
45.946	167.99	⅞	71.864	410.97	⅛	97.782	760.87
46.338	170.87	23.	72.257	415.48	¼	98.175	766.99
46.731	173.78	⅛	72.649	420.00	⅜	98.567	773.14
47.124	176.71	¼	73.042	424.56	½	98.960	779.31
47.517	179.67	⅜	73.435	429.13	⅝	99.353	785.51
47.909	182.65	½	73.827	433.74	¾	99.746	791.73
48.302	185.66	⅝	74.220	438.36	⅞	100.138	797.98
48.695	188.69	¾	74.613	443.01	32.	100.531	804.25
49.087	191.75	⅞	75.006	447.69	⅛	100.924	810.54
49.480	194.83	24.	75.398	452.39	¼	101.316	816.86
49.873	197.93	⅛	75.791	457.11	⅜	101.709	823.21
50.265	201.06	¼	76.184	461.86	½	102.102	829.58
50.658	204.22	⅜	76.576	466.64	⅝	102.494	835.97
51.051	207.39	½	76.969	471.44	¾	102.887	842.39
51.444	210.60	⅝	77.362	476.26	⅞	103.280	848.83
51.836	213.82	¾	77.754	481.11	33	103.673	855.30
52.229	217.08	⅞	78.147	485.98	⅛	104.065	861.79
52.622	220.35	25.	78.540	490.87	¼	104.458	868.31
53.014	223.65	⅛	78.933	495.79	⅜	104.851	874.85
53.407	226.98	¼	79.325	500.74	½	105.243	881.41
53.800	230.33	⅜	79.718	505.71	⅝	105.636	888.00
54.192	233.71	½	80.111	510.71	¾	106.029	894.62
54.585	237.10	⅝	80.503	515.72	⅞	106.421	901.26
54.978	240.53	¾	80.896	520.77	34.	106.814	907.92
55.371	243.98	⅞	81.289	525.84	⅛	107.207	914.61
55.763	247.45	26.	81.681	530.93	¼	107.600	921.32
56.156	250.95	⅛	82.074	536.05	⅜	107.992	928.06
56.549	254.47	¼	82.467	541.19	½	108.385	934.82
56.941	258.02	⅜	82.860	546.35	⅝	108.778	941.61
57.334	261.59	½	83.252	551.55	¾	109.170	948.42
57.727	265.18	⅝	83.645	556.76	⅞	109.563	955.25
58.119	268.80	¾	84.038	562.00	35.	109.956	962.11
58.512	272.45	⅞	84.430	567.27	⅛	110.348	969.00
58.905	276.12	27.	84.823	572.56	¼	110.741	975.91
59.298	279.81	⅛	85.216	577.87	⅜	111.134	982.84
59.690	283.53	¼	85.608	583.21	½	111.527	989.80
60.083	287.27	⅜	86.001	588.57	⅝	111.919	996.78
60.476	291.04	½	86.394	593.96	¾	112.312	1003.8
60.868	294.83	⅝	86.786	599.37	⅞	112.705	1010.8
61.261	298.65	¾	87.179	604.81	36.	113.097	1017.9
61.654	302.49	⅞	87.572	610.27	⅛	113.490	1025.0
62.046	306.35	28.	87.965	615.75	¼	113.883	1032.1
62.439	310.24	⅛	88.357	621.26	⅜	114.275	1039.2
62.832	314.16	¼	88.750	626.80	½	114.668	1046.3
63.225	318.10	⅜	89.143	632.36	⅝	115.061	1053.5
63.617	322.06	½	89.535	637.94	¾	115.454	1060.7
64.010	326.05	⅝	89.928	643.55	⅞	115.846	1068.0
64.403	330.06	¾	90.321	649.18	37.	116.239	1075.2
64.795	334.10	⅞	90.713	654.84	⅛	116.632	1082.5
65.188	338.16	29.	91.106	660.52	¼	117.024	1089.8
65.581	342.25	⅛	91.499	666.23	⅜	117.417	1097.1
65.973	346.36	¼	91.892	671.96	½	117.810	1104.5
66.366	350.50	⅜	92.284	677.71	⅝	118.202	1111.8
66.759	354.66	½	92.677	683.49	¾	118.596	1119.2
67.152	358.84	⅝	93.070	689.30	⅞	118.988	1126.7
67.544	363.05	¾	93.462	695.13	38.	119.381	[illegible]
67.937	367.28	⅞	93.855	700.98	⅛	119.7[illegible]	[illegible]
68.330	371.54	30.	94.248	706.86	¼	120.[illegible]	[illegible]

Diam.	Circum.	Area.	Diam	Circum.	Area.	Diam.	Circ
38 3/8	120.559	1156.6	46 5/8	146.477	1707.4	54 7/8	172.
1/2	120.951	1164.2	3/4	146.869	1716.5	55.	172.
5/8	121.344	1171.7	7/8	147.262	1725.7	1/8	173
3/4	121.737	1179.3	47.	147.655	1734.9	1/4	173
7/8	122.129	1186.9	1/8	148.048	1744.2	3/8	173
39.	122.522	1194.6	1/4	148.440	1753.5	1/2	174
1/8	122.915	1202.3	3/8	148.833	1762.7	5/8	174
1/4	123.308	1210.0	1/2	149.226	1772.1	3/4	175
3/8	123.700	1217.7	5/8	149.618	1781.4	7/8	175
1/2	124.093	1225.4	3/4	150.011	1790.8	56.	175
5/8	124.486	1233.2	7/8	150.404	1800.1	1/8	176
3/4	124.878	1241.0	48.	150.796	1809.6	1/4	176
7/8	125.271	1248.8	1/8	151.189	1819.0	3/8	17
40.	125.664	1256.6	1/4	151.582	1828.5	1/2	17
1/8	126.056	1264.5	3/8	151.975	1837.9	5/8	177
1/4	126.449	1272.4	1/2	152.367	1847.5	3/4	178
3/8	126.842	1280.3	5/8	152.760	1857.0	7/8	178.
1/2	127.235	1288.2	3/4	153.153	1866.5	57.	179.
5/8	127.627	1296.2	7/8	153.545	1876.1	1/8	179
3/4	128.020	1304.2	49.	153.938	1885.7	1/4	179
7/8	128.413	1312.2	1/8	154.331	1895.4	3/8	180
41.	128.805	1320.3	1/4	154.723	1905.0	1/2	180.
1/8	129.198	1328.3	3/8	155.116	1914.7	5/8	181
1/4	129.591	1336.4	1/2	155.509	1924.4	3/4	181
3/8	129.983	1344.5	5/8	155.902	1934.2	7/8	181.
1/2	130.376	1352.7	3/4	156.294	1943.9	58.	182.
5/8	130.769	1360.8	7/8	156.687	1953.7	1/8	182.
3/4	131.161	1369.0	50.	157.080	1963.5	1/4	182.
7/8	131.554	1377.2	1/8	157.472	1973.3	3/8	183.
42.	131.947	1385.4	1/4	157.865	1983.2	1/2	183.
1/8	132.340	1393.7	3/8	158.258	1993.1	5/8	184.1
1/4	132.732	1402.0	1/2	158.650	2003.0	3/4	184.5
3/8	133.125	1410.3	5/8	159.043	2012.9	7/8	184.9
1/2	133.518	1418.6	3/4	159.436	2022.8	59.	185.3
5/8	133.910	1427.0	7/8	159.829	2032.8	1/8	185.7
3/4	134.303	1435.4	51.	160.221	2042.8	1/4	186.1
7/8	134.696	1443.8	1/8	160.614	2052.8	3/8	186.5
43.	135.088	1452.2	1/4	161.007	2062.9	1/2	186.9
1/8	135.481	1460.7	3/8	161.399	2073.0	5/8	187.3
1/4	135.874	1469.1	1/2	161.792	2083.1	3/4	187.7
3/8	136.267	1477.6	5/8	162.185	2093.2	7/8	188.1
1/2	136.659	1486.2	3/4	162.577	2103.3	60.	188.4
5/8	137.052	1494.7	7/8	162.970	2113.5	1/8	188.8
3/4	137.445	1503.3	52.	163.363	2123.7	1/4	189.2
7/8	137.837	1511.9	1/8	163.756	2133.9	3/8	189.6
44.	138.230	1520.5	1/4	164.148	2144.2	1/2	190.0
1/8	138.623	1529.2	3/8	164.541	2154.5	5/8	190.4
1/4	139.015	1537.9	1/2	164.934	2164.8	3/4	190.8
3/8	139.408	1546.6	5/8	165.326	2175.1	7/8	191.2
1/2	139.801	1555.3	3/4	165.719	2185.4	61.	191
5/8	140.194	1564.0	7/8	166.112	2195.8	1/8	192.0
3/4	140.586	1572.8	53.	166.504	2206.2	1/4	192.4
7/8	140.979	1581.6	1/8	166.897	2216.6	3/8	192.8
45.	141.372	1590.4	1/4	167.290	2227.0	1/2	193.2
1/8	141.764	1599.3	3/8	167.683	2237.5	5/8	193.6
1/4	142.157	1608.2	1/2	168.075	2248.0	3/4	193.9
3/8	142.550	1617.0	5/8	168.468	2258.5	7/8	194.3
1/2	142.942	1626.0	3/4	168.861	2269.1	62.	194.7
5/8	143.335	1634.9	7/8	169.253	2279.6	1/8	195.1
3/4	143.728	1643.9	54.	169.646	2290.2	1/4	195.5
	144.121	1652.9	1/8	170.039	2300.8	3/8	195.9
	44.513	1661.9	1/4	170.431	2311.5	1/2	196.
	[illegible]	[illegible]	3/8	170.824	2322.1	5/8	196.
			1/2	171.217	2332.8	3/4	197.
			5/8	171.609	2343.5	7/8	197
			3/4	172.002	2354.3	63.	197

rea.	Diam.	Circum.	Area.	Diam.	Circum.	Area.
129.6	71 ⅜	224.231	4001.1	79 ⅝	250.149	4979.5
142.0	½	224.624	4015.2	¾	250 542	4995.2
154.5	⅝	225.017	4029.2	⅞	250.935	5010.9
166.9	¾	225.409	4043 3	80.	251.327	5026.5
179.4	⅞	225.802	4057.4	⅛	251.720	5042.3
191.9	72.	226.195	4071.5	¼	252.113	5058.0
204.4	⅛	226.587	4085 7	⅜	252.506	5073.8
217.0	¼	226.980	4099.8	½	252.898	5089.6
229.6	⅜	227.373	4114.0	⅝	253.291	5105.4
242.2	½	227.765	4128.2	¾	253.684	5121.2
254.8	⅝	228.158	4142.5	⅞	254.076	5137.1
267.5	¾	228.551	4156.8	81.	254 469	5153.0
280.1	⅞	228.944	4171.1	⅛	254.862	5168.9
292.8	73.	229.336	4185.4	¼	255.254	5184.9
305.6	⅛	229.729	4199.7	⅜	255.647	5200.8
318.3	¼	230.122	4214.1	½	256.040	5216.8
331.1	⅜	230.514	4228.5	⅝	256.433	5232.8
343.9	½	230.907	4242.9	¾	256.825	5248.9
356.7	⅝	231.300	4257.4	⅞	257.218	5264.9
369.6	¾	231.692	4271.8	82.	257.611	5281.0
382.4	⅞	232.085	4286.3	⅛	258.003	5297.1
395.3	74.	232.478	4300.8	¼	258.396	5313.3
408.2	⅛	232.871	4315.4	⅜	258 789	5329.4
421.2	¼	233.263	4329.9	½	259.181	5345.6
434.2	⅜	233 656	4344 5	⅝	259.574	5361.8
447.2	½	234.049	4359.2	¾	259.967	5378.1
460.2	⅝	234.441	4373.8	⅞	260.359	5394.3
473.2	¾	234.834	4388.5	83.	260.752	5410.6
486.3	⅞	235.227	4403.1	⅛	261.145	5426.9
499.4	75.	235.619	4417.9	¼	261.538	5443.3
512.5	⅛	236.012	4432.6	⅜	261.930	5459.6
525 7	¼	236.405	4447.4	½	262.323	5476.0
538.8	⅜	236.798	4462.2	⅝	262.7'6	5492.4
552.0	½	237.190	4477.0	¾	263.108	5508.8
565.2	⅝	237.583	4491.8	⅞	263.501	5525.3
578.5	¾	237.976	4506.7	84.	263.894	5541.8
591.7	⅞	238.368	4521.5	⅛	264.286	5558.3
605.0	76.	238.761	4536.5	¼	264.679	5574.8
618.3	⅛	239.154	4551.4	⅜	265.072	5591.4
631.7	¼	239.546	4566.4	½	265.465	5607.9
645.0	⅜	239.939	4581.3	⅝	265.857	5624.5
658.4	½	240.332	4596.3	¾	266.250	5641.2
671.8	⅝	240.725	4611.4	⅞	266.643	5657.8
685.3	¾	241.117	4626.4	85.	267.035	5674.5
698.7	⅞	241.510	4641.5	⅛	267.428	5691.2
712.2	77.	241.903	4656.6	¼	267.821	5707.9
725.7	⅛	242.295	4671.8	⅜	268.213	5724.7
739.3	¼	242.688	4686.9	½	268.606	5741.5
752.8	⅜	243.081	4702.1	⅝	268.999	5758.3
766.4	½	243.473	4717.3	¾	269.392	5775.1
780.0	⅝	243.866	4732.5	⅞	269.784	5791.9
793.7	¾	244.259	4747.8	86.	270.177	5808.8
807.3	⅞	244.652	4763.1	⅛	270.570	5825.7
821.0	78.	245.044	4778.4	¼	270.962	5842.6
834.7	⅛	245.437	4793.7	⅜	271.355	5859.6
848.5	¼	245.830	4809.0	½	271.748	5876.5
862.2	⅜	246.222	4824.4	⅝	272.140	5893.5
876.0	½	246.615	4839 8	¾	272.533	5910.6
889.8	⅝	247.008	4855.2	⅞	272.926	5927.6
903.6	¾	247.400	4870.7	87.	273.319	5944.7
917.5	⅞	247.793	4886.2	⅛	273.711	5961.8
931.4	79.	248.186	4901.7	¼	274.104	[illegible]
945.3	⅛	248.579	4917.2	⅜	274.497	[illegible]
59.2	¼	248.971	4932.7	½	274.8[illegible]	[illegible]
3.1	⅜	249.364	4948.3	⅝	275.[illegible]	[illegible]
7.1	½	249.757	4963.0	¾	275.[illegible]	[illegible]

Diam.	Circum.	Area.	Diam.	Circum.	Area.	Diam.	Cir
87 ⅞	276.067	6064.9	92.	289.027	6647.6	96 ⅛	301.
88.	276.460	6082.1	⅛	289.419	6665.7	¼	302.
⅛	276.853	6099.4	¼	289.812	6683.8	⅜	302.
¼	277.246	6116.7	⅜	290.205	6701.9	½	303.
⅜	277.638	6134.1	½	290.597	6720.1	⅝	303.
½	278.031	6151.4	⅝	290.990	6738.2	¾	303.
⅝	278.424	6168.8	¾	291.383	6756.4	⅞	304.
¾	278.816	6186.2	⅞	291.775	6774.7	97.	304.
⅞	279.209	6203.7	93.	292.168	6792.9	⅛	305.
89.	279.602	6221.1	⅛	292.561	6811.2	¼	305.
⅛	279.994	6238.6	¼	292.954	6829.5	⅜	305.
¼	280.387	6256.1	⅜	293.346	6847.8	½	306.
⅜	280.780	6273.7	½	293.739	6866.1	⅝	306.
½	281.173	6291.2	⅝	294.132	6884.5	¾	307.
⅝	281.565	6 08.8	¾	294.524	6902.9	⅞	307.
¾	281.958	6326.4	⅞	294.917	6921.3	98.	307.
⅞	282.351	6344.1	94.	295.310	6939.8	⅛	308.
90.	282.743	6361.7	⅛	295.702	6958.2	¼	308.
⅛	283.136	6379.4	¼	296.095	6976.7	⅜	309.
¼	283.529	6397.1	⅜	296.488	6995.3	½	309.
⅜	283.921	6414.9	½	296.881	7013.8	⅝	309.
½	284.314	6432.6	⅝	297.273	7032.4	¾	310.
⅝	284.707	6450.4	¾	297.666	7051.0	⅞	310.
¾	285.100	6468.2	⅞	298.059	7069.6	99.	311.
⅞	285.492	6486.0	95.	298.451	7088.2	⅛	311.
91.	285.885	6503.9	⅛	298.844	7106.9	¼	311.
⅛	286.278	6521.8	¼	299.237	7125.6	⅜	312.
¼	286.670	6539.7	⅜	299.629	7144.3	½	312.
⅜	287.063	6557.6	½	300.022	7163.0	⅝	312.
½	287.456	6575.5	⅝	300.415	7181.8	¾	313.
⅝	287.848	6593.5	¾	300.807	7200.6	⅞	313.
¾	288.241	6611.5	⅞	301.200	7219.4	100.	314.
⅞	288.634	6629.6	96.	301.593	7238.2		

DECIMALS OF A FOOT EQUIVALENT TO AND FRACTIONS OF AN INCH.

Inches.	0	⅛	¼	⅜	½	⅝	3
0	0	.01042	.02083	.03125	.04166	.05208	.06
1	.0833	.0937	.1042	.1146	.1250	.1354	.14
2	.1667	.1771	.1875	.1979	.2083	.2188	.22
3	.2500	.2604	.2708	.2813	.2917	.3021	.31
4	.3333	.3437	.3542	.3646	.3750	.3854	.39
5	.4167	.4271	.4375	.4479	.4583	.4688	.47
6	.5000	.5104	.5208	.5313	.5417	.5521	.56
7	.5833	.5937	.6042	.6146	.6250	.6354	.64
8	.6667	.6771	.6875	.6979	.7083	.7188	.72
9	.7500	.7604	.7708	.7813	.7917	.8021	.81
10	.8333	.8437	.8542	.8646	.8750	.8854	.89
11	.9167	.9271	.9375	.9479	.9583	.9688	.97

Versed Sine.	Area.	Versed Sine.	Area.	Versed Sine.	Area.	Versed Sine.	Area.
.313	.21015	.36	.25455	407	.30024	.454	.34676
314	.21108	.361	.25551	.408	.30122	.455	.34776
315	.21201	.362	.25647	.409	.30220	.456	.34876
316	.21294	.363	.25743	.41	.30319	.457	.34975
317	.21387	.364	.25839	.411	.30417	.458	.35075
18	.21480	.365	.25936	412	.30516	.459	.35175
19	.21573	.366	.26032	.413	.30614	.46	.35274
2	.21667	.367	.26128	.414	.30712	.461	.35374
21	.21760	.368	.26225	.415	.30811	.462	.35474
22	.21853	.369	.26321	.416	.30910	.463	.35573
23	.21947	.37	.26418	.417	.31008	.464	.35673
4	.22040	.371	.26514	.418	.31107	.465	.35773
5	.22134	.372	.26611	.419	.31205	.466	.35873
6	.22228	.373	.26708	.42	.31304	.467	.35972
7	.22322	.374	.26805	.421	.31403	.468	.36072
8	.22415	.375	.26901	.422	.31502	.469	.36172
9	.22509	.376	.26998	.423	.31600	.47	.36272
	.22603	.377	.27095	.424	.31699	.471	.36372
	.22697	.378	.27192	.425	.31798	.472	.36471
	.22792	.379	.27289	.426	.31897	.473	.36571
	.22886	.38	.27386	.427	.31996	.474	.36671
	.22980	.381	.27483	.428	.32095	.475	.36771
	.23074	.382	.27580	.429	.32194	.476	.36871
	.23169	.383	.27678	.43	.32293	.477	.36971
	.23263	.384	.27775	.431	.32392	.478	.37071
	.23358	.385	.27872	.432	.32491	.479	.37171
	.23453	.386	.27969	.433	.32590	.48	.37270
	.23547	.387	.28067	.434	.32689	.481	.37370
1	.23642	.388	.28164	.435	.32788	.482	.37470
2	.23737	.389	.28262	.436	.32887	.483	.37570
3	.23832	.39	.28359	.437	.32987	.484	.37670
4	.23927	.391	.28457	.438	.33086	.485	.37770
5	.24022	.392	.28554	.439	.33185	.486	.37870
6	.24117	.393	.28652	.44	.33284	.487	.37970
7	.24212	.394	.28750	.441	.33384	.488	.38070
48	.24307	.395	.28848	.442	.33483	.489	.38170
49	.24403	.396	.28945	.443	.33582	.49	.38270
5	.24498	.397	.29043	.444	.33682	.491	.38370
51	.24593	.398	.29141	.445	.33781	.492	.38470
52	.24689	.399	.29239	.446	.33880	.493	.38570
353	.24784	.4	.29337	.447	.33980	.494	.38670
354	.24880	.401	.29435	448	.34079	.495	.38770
355	.24976	.402	.29533	.449	.34179	.496	.38870
356	.25071	.403	.29631	.45	.34278	.497	.38970
357	.25167	.404	.29729	.451	.34378	.498	.39070
.358	.25263	.405	.29827	.452	.34477	.499	.39170
.359	.25359	.406	.29926	.453	.34577	.5	.39270

nding the area of a segment see Mensuration, page 59.

AREAS OF THE SEGMENTS OF A CIRCLE.

(Diameter = 1; Rise or Versed Sine in parts of Diamet being given.)

RULE FOR USE OF THE TABLE.—Divide the rise or height of the segment the diameter to obtain the versed sine. Multiply the area in the table c responding to this versed sine by the square of the diameter.

If the segment exceeds a semicircle its area is area of circle – area of s ment whose rise is (diam. of circle – rise of given segment).

Given chord and rise, to find diameter. Diam. = (square of half chord rise) + rise. The half chord is a mean proportional between the two pa into which the chord divides the diameter which is perpendicular to it.

Versed Sine.	Area.	Versed Sine.	Area.	Versed Sine.	Area.	Versed Sine.	Area.	Versed Sine.
.001	.00004	.054	.01646	.107	.04514	.16	.08111	.213
.002	.00012	.055	.01691	.108	.04576	.161	.08185	.214
.003	.00022	.056	.01737	.109	.04638	.162	.08258	.215
.004	.00034	.057	.01783	.11	.04701	.163	.08332	.216
.005	.00047	.058	.01830	.111	.04763	.164	.08406	.217
.006	.00062	.059	.01877	.112	.04826	.165	.08480	.218
.007	.00078	.06	.01924	.113	.04889	.166	.08554	.219
.008	.00095	.061	.01972	.114	.04953	.167	.08629	.22
.009	.00113	.062	.02020	.115	.05016	.168	.08704	.221
.01	.00133	.063	.02068	.116	.05080	.169	.08779	.222
.011	.00153	.064	.02117	.117	.05145	.17	.08854	.223
.012	.00175	.065	.02166	.118	.05209	.171	.08929	.224
.013	.00197	.066	.02215	.119	.05274	.172	.09004	.225
.014	.0022	.067	.02265	.12	.05338	.173	.09080	.226
.015	.00244	.068	.02315	.121	.05404	.174	.09155	.227
.016	.00268	.069	.02366	.122	.05469	.175	.09231	.228
.017	.00291	.07	.02417	.123	.05535	.176	.09307	.229
.018	.0032	.071	.02468	.124	.05600	.177	.09384	.23
.019	.00347	.072	.02520	.125	.05666	.178	.09460	.231
.02	.00375	.073	.02571	.126	.05733	.179	.09537	.232
.021	.00403	.074	.02624	.127	.05799	.18	.09613	.233
.022	.00432	.075	.02676	.128	.05866	.181	.09690	.234
.023	.00462	.076	.02729	.129	.05933	.182	.09767	.235
.024	.00492	.077	.02782	.13	.06000	.183	.09845	.236
.025	.00523	.078	.02836	.131	.06067	.184	.09922	.237
.026	.00555	.079	.02889	.132	.06135	.185	.10000	.238
.027	.00587	.08	.02943	.133	.06203	.186	.10077	.239
.028	.00619	.081	.02998	.134	.06271	.187	.10155	.24
.029	.00653	.082	.03053	.135	.06339	.188	.10233	.241
.03	.00687	.083	.03108	.136	.06407	.189	.10312	.242
.031	.00721	.084	.03163	.137	.06476	.19	.10390	.243
.032	.00756	.085	.03219	.138	.06545	.191	.10469	.244
.033	.00791	.086	.03275	.139	.06614	.192	.10547	.245
.034	.00827	.087	.03331	.14	.06683	.193	.10626	.246
.035	.00864	.088	.03387	.141	.06753	.194	.10705	.247
.036	.00901	.089	.03444	.142	.06822	.195	.10784	.248
.037	.00938	.09	.03501	.143	.06892	.196	.10864	.249
.038	.00976	.091	.03559	.144	.06963	.197	.10943	.25
.039	.01015	.092	.03616	.145	.07033	.198	.11023	.251
.04	.01054	.093	.03674	.146	.07103	.199	.11102	.252
.041	.01093	.094	.03732	.147	.07174	.2	.11182	.253
.042	.01133	.095	.03791	.148	.07245	.201	.11262	.254
.043	.01173	.096	.03850	.149	.07316	.202	.11343	.255
.044	.01214	.097	.03909	.15	.07387	.203	.11423	.256
.045	.01255	.098	.03968	.151	.07459	.204	.11504	.257
.046	.01297	.099	.04028	.152	.07531	.205	.11584	.258
.047	.01339	.1	.04087	.153	.07603	.206	.11665	.259
.048	.01382	.101	.04148	.154	.07675	.207	.11746	.26
.049	.01425	.102	.04208	.155	.07747	.208	.11827	.261
.05	.01468	.103	.04269	.156	.07819	.209	.11908	.262
[illegible]	.01512	.104	.04330	.157	.07892	.21	.11990	.263
[illegible]	[illegible]	.105	.04391	.158	.07965	.211	.12071	.264
[illegible]	[illegible]	[illegible]	.04452	.159	.08038	.212	.12153	.265

SPHERES—(*Continued.*)

Diam.	Surface.	Solidity.	Diam.	Surface.	Solidity.	Diam.	Surface.	Solidity.
22 ½	1590.4	5964.1	40 ½	5153.1	34783	70 ½	15615	183471
¾	1626.0	6165.2	41.	5281.1	36087	71.	15837	187402
23.	1661.9	6370.6	½	5410.7	37423	½	16061	191389
¼	1698.2	6580.6	42.	5541.9	38792	72.	16286	195433
½	1735.0	6795.2	½	5674.5	40194	½	16513	199532
¾	1772.1	7014.3	43.	5808.8	41630	73.	16742	203689
24.	1809.6	7238.2	½	5944.7	43099	½	16972	207903
¼	1847.5	7466.7	44.	6082.1	44602	74.	17204	212175
½	1885.8	7700.1	½	6221.2	46141	½	17437	216505
¾	1924.4	7938.3	45.	6361.7	47713	75.	17672	220894
25.	1963.5	8181.3	½	6503.9	49321	½	17908	225341
¼	2002.9	8429.2	46.	6647.6	50965	76.	18146	229848
½	2042.8	8682.0	½	6792.9	52645	½	18386	234414
¾	2083.0	8939.9	47.	6939.9	54362	77.	18626	239041
[illegible]	2123.7	9202.8	½	7088.3	56115	½	18869	243728
[illegible]	2164.7	9470.8	48.	7238.3	57906	78.	19114	248475
[illegible]	2206.2	9744.0	½	7389.9	59734	½	19360	253284
[illegible]	2248.0	10022	49.	7543.1	61601	79.	19607	258155
[illegible]	2290.2	10306	½	7697.7	63506	½	19856	263088
[illegible]	2332.8	10595	50.	7854.0	65450	80.	20106	268083
[illegible]	[illegible]	10889	½	8011.8	67433	½	20358	273141
[illegible]	[illegible]	11189	51.	8171.2	69456	81.	20612	278263
[illegible]	[illegible]	11494	½	8332.3	71519	½	20867	283447
[illegible]	[illegible]	11805	52.	8494.8	73622	82.	21124	288696
[illegible]	[illegible]	12121	½	8658.9	75767	½	21382	294010
[illegible]	[illegible]	12443	53.	8824.8	77952	83.	21642	299388
[illegible]	[illegible]	12770	½	8992.0	80178	½	21904	304831
[illegible]	[illegible]	13103	54.	9160.8	82448	84.	22167	310340
[illegible]	[illegible]	13442	½	9331.2	84760	½	22432	315915
[illegible]	[illegible]	13787	55.	9503.2	87114	85.	22698	321556
[illegible]	[illegible]	14137	½	9676.8	89511	½	22966	327264
[illegible]	[illegible]	14494	56.	9852.0	91953	86.	23235	333039
[illegible]	[illegible]	14856	½	10029	94438	½	23506	338882
[illegible]	[illegible]	15224	57.	10207	96967	87.	23779	344792
[illegible]	[illegible]	15599	½	10387	99541	½	24053	350771
[illegible]	[illegible]	15979	58.	10568	102161	88.	24328	356819
[illegible]	[illegible]	16366	½	10751	104826	½	24606	362935
[illegible]	[illegible]	16758	59.	10936	107536	89.	24885	369122
[illegible]	[illegible]	17157	½	11122	110294	½	25165	375378
[illegible]	[illegible]	17563	60.	11310	113098	90.	25447	381704
[illegible]	[illegible]	17974	½	11499	115949	½	25730	388102
[illegible]	[illegible]	18392	61.	11690	118847	91.	26016	394570
[illegible]	[illegible]	18817	½	11882	121794	½	26302	401109
[illegible]	[illegible]	19248	62.	12076	124789	92.	26590	407721
[illegible]	[illegible]	19685	½	12272	127832	½	26880	414405
[illegible]	[illegible]	20129	63.	12469	130925	93.	27172	421161
[illegible]	[illegible]	20580	½	12668	134067	½	27464	427991
[illegible]	[illegible]	21037	64.	12868	137259	94.	27759	434894
[illegible]	[illegible]	21501	½	13070	140501	½	28055	441871
[illegible]	[illegible]	22449	65.	13273	143794	95.	28353	448920
[illegible]	[illegible]	23425	½	13478	147138	½	28652	456047
[illegible]	[illegible]	24429	66.	13685	150533	96.	28953	463248
[illegible]	[illegible]	25461	½	13893	153980	½	29255	470524
[illegible]	[illegible]	26522	67.	14103	157480	97.	29559	477874
[illegible]	[illegible]	27612	½	14314	161032	½	29865	485302
[illegible]	[illegible]	28731	68.	14527	164637	98.	30172	492808
[illegible]	[illegible]	29880	½	14741	168295	½	30481	500388
[illegible]	[illegible]	31059	69.	14957	172007	99.	30791	50804[illegible]
[illegible]	[illegible]	32270	½	15175	175774	½	31103	5157[illegible]
[illegible]	[illegible]	33510	70.	15394	179595	100.	31416	5235[illegible]

SPHERES.

ome errors of 1 in the last figure only. From TRAUTWINE.)

Surface.	Solidity.	Diam.	Surface.	Solidity.	Diam.	Surface.	Solidity.
.00307	.00002	3 1/4	33.183	17.974	9 7/8	306.36	504.21
.01227	.00013	5-16	34.472	19.031	10.	314.16	523.60
.02761	.00043	3/8	35.784	20.129	1/8	322.06	543.48
.04909	.00102	7-16	37.122	21.268	1/4	330.06	563.86
.07670	.00200	1/2	38.484	22.449	3/8	338.16	584.74
.11045	.00345	9-16	39.872	23.674	1/2	346.36	606.13
.15033	.00548	5/8	41 283	24.942	5/8	354.66	628.04
.19635	.00818	11-16	42.719	26.254	3/4	363.05	650.46
.24851	.01165	3/4	44.179	27.611	7/8	371.54	673.43
.30680	.01598	13-16	45.664	29.016	11.	380.13	696.91
.37123	.02127	7/8	47.173	30.466	1/8	388.83	720.92
.44179	.02761	15-16	48.708	31.965	1/4	397.61	745.51
.51848	.03511	4.	50.265	33.510	3/8	406.49	770.64
.60132	.04385	1/8	53.456	36.751	1/2	415.48	796.33
.69028	.05393	1/4	56.745	40.195	5/8	424.50	822.58
.78540	.06545	3/8	60.133	43 847	3/4	433.73	849.40
.99403	.09319	1/2	63.617	47.713	7/8	443.01	876.79
.2272	.12783	5/8	67.201	51.801	12.	452.39	904.78
.4849	.17014	3/4	70.883	56.116	1/4	471.44	962.52
.7671	.22089	7/8	74.663	60.663	1/2	490.87	1022.7
.0739	.28084	5.	78.540	65.450	3/4	510.71	1085.3
.4053	.35077	1/8	82.516	70.482	13.	530.93	1150.3
.7611	.43143	1/4	86.591	75.767	1/4	551.55	1218.0
.1416	.52360	3/8	90.763	81.308	1/2	572.55	1288.3
.5466	.62804	1/2	95.033	87.113	3/4	593.95	1361.2
.9761	.74551	5/8	99.401	93.189	14.	615.75	1436.8
.4301	.87681	3/4	103.87	99.541	1/4	637.95	1515.1
.9088	1.0227	7/8	108.44	106.18	1/2	660.52	1596.3
.4119	1.1839	6.	113.10	113.10	3/4	683.49	1680.3
.9396	1.3611	1/8	117.87	120.31	15.	706.85	1767.2
.4919	1.5553	1/4	122.72	127.83	1/4	730.63	1857.0
.0686	1.7671	3/8	127.68	135.66	1/2	754.77	1949.8
.6699	1.9974	1/2	132.73	143.79	3/4	779.32	2045.7
.2957	2.2468	5/8	137.89	152.25	16.	804.25	2144.7
.9461	2.5161	3/4	143.14	161.03	1/4	829.57	2246.8
.6211	2.8062	7/8	148.49	170.14	1/2	855.29	2352.1
.321	3.1177	7.	153.94	179.59	3/4	881.42	2460.6
.044	3.4514	1/8	159.49	189.39	17.	907.93	2572.4
.793	3.8083	1/4	165.13	199.53	1/4	934.83	2687.6
.566	4.1888	3/8	170.87	210.03	1/2	962.12	2806.2
.364	4.5939	1/2	176.71	220.89	3/4	989.80	2928.2
.186	5.0243	5/8	182.66	232.13	18.	1017.9	3053.6
.033	5.4809	3/4	188.69	243.73	1/4	1046.4	3182.6
.904	5.9641	7/8	194.83	255.72	1/2	1075.2	3315.3
.800	6.4751	8.	201.06	268.08	3/4	1104.5	3451.5
.721	7.0144	1/8	207.39	280.85	19.	1134.1	3591.4
.666	7.5829	1/4	213.82	294.01	1/4	1164.2	3735.0
.635	8.1813	3/8	220.35	307.58	1/2	1194.6	3882.5
.629	8.8103	1/2	226.98	321.56	3/4	1225.4	4033.7
.648	9.4708	5/8	233.71	335.95	20.	1256.7	4188.8
.691	10.164	3/4	240.53	350.77	1/4	1288.3	4347.8
.758	10.889	7/8	247.45	360.02	1/2	1320.3	4510.9
.850	11.649	9.	254.47	381.70	3/4	1352.7	4677.9
.967	12.443	1/8	261.59	397.83	21.	1385.5	4849.1
.109	13.272	1/4	268.81	414.41	1/4	1418.6	5024.3
.274	14.137	3/8	270.12	431.44	1/2	1452.2	5203.7
.465	15.039	1/2	283.53	448.92	3/4	1486.2	5387.4
		5/8	291.04	466.87	22.	1520.5	5575.3
		3/4	298.65	485.31	1/4	1555.3	5767.6

CYLINDRICAL VESSELS, TANKS, CISTERNS, ETC.

Diameter in Feet and Inches, Area in Square Feet, and U. S. Gallons Capacity for One Foot in Depth.

1 gallon = 231 cubic inches = $\frac{\text{1 cubic foot}}{7.4805}$ = 0.13368 cubic feet.

am.	Area.	Gals.	Diam.		Area.	Gals.	Diam.		Area.	Gals.
In.	Sq. ft.	1 foot depth.	Ft.	In.	Sq. ft.	1 foot depth.	Ft.	In.	Sq. ft.	1 foot depth.
	.785	5 87	5	8	25.22	188 66	**19**		283.53	2120.9
1	.922	6.89	5	9	25.97	194.25	19	3	291.04	2177.1
2	1.069	8.00	5	10	26 73	199.92	19	6	298.65	2234.0
3	1.227	9.18	5	11	27.49	205.67	19	9	306.35	2291.7
4	1.396	10.44	**6**		28.27	211.51	**20**		314.16	2350.1
5	1.576	11.79	6	3	30.68	229.50	20	3	322.06	2409.2
6	1.767	13.22	6	6	33.18	248.23	20	6	330.06	2469.1
7	1.969	14.73	6	9	35.78	267.69	20	9	338 16	2529.6
8	2.182	16.32	**7**		38.48	287.88	**21**		346.36	2591.0
9	2.405	17 99	7	3	41.28	308.81	21	3	354.66	2653.0
10	2.640	19.75	7	6	44.18	330.48	21	6	363.05	2715.8
11	2.885	21.58	7	9	47.17	352.88	21	9	371.54	2779.3
	3.142	23.50	**8**		50.27	376.01	**22**		380.13	2843.6
1	3.409	25.50	8	3	53.46	399.88	22	3	388.82	2908.6
2	3.687	27.58	8	6	56.75	424.48	22	6	397.61	2974.3
3	3.976	29.74	8	9	60.13	449.82	22	9	406.49	3040.8
4	4.276	31.99	**9**		63.62	475.89	**23**		415.48	3108.0
5	4 587	34 31	9	3	67 20	502.70	23	3	424.56	3175.9
6	4.909	36.72	9	6	70.88	530 24	23	6	433.74	3244.6
7	5.241	39.21	9	9	74.66	558.51	23	9	443 01	3314 0
8	5.585	41.78	**10**		78.54	587.52	**24**		452.39	3384.1
9	5.940	44.43	10	3	82.52	617.26	24	3	461.86	3455.0
10	6.305	47.16	10	6	86.59	647.74	24	6	471.44	3526.6
11	6.681	49 98	10	9	90.76	678.95	24	9	481.11	3598.9
	7.069	52 88	**11**		95.03	710.90	**25**		490.87	3672.0
1	7.467	55.86	11	3	99.40	743.58	25	3	500.74	3745.8
2	7.876	58.92	11	6	103.87	776.99	25	6	510.71	3820 3
3	8.296	62.06	11	9	108.43	811.14	25	9	520.77	3895.6
	8.727	65.28	**12**		113.10	846.03	**26**		530.93	3971.6
	9.168	68.58	12	3	117.86	881.65	26	3	541.19	4048.4
	9.621	71.97	12	6	122.72	918.00	26	6	551.55	4125.9
	10.085	75.44	12	9	127.68	955.09	26	9	562.00	4204.1
	10.559	78.99	**13**		132.73	992.91	**27**		572.56	4283.0
	11.045	82 62	13	3	137.89	1031.5	27	3	583.21	4362.7
	11.541	86.33	13	6	143.14	1070.8	27	6	593.96	4443.1
	12.048	90.13	13	9	148.49	1110.8	27	9	604.81	4524.3
	12.566	94.00	**14**		153.94	1151.5	**28**		615.75	4606.2
	13.095	97.96	14	3	159.48	1193.0	28	3	626.80	4688.8
	13.635	102.00	14	6	165.13	1235.3	28	6	637.94	4772 1
	14.186	106.12	14	9	170.87	1278.2	28	9	649.18	4856.2
	14.748	110.32	**15**		176.71	1321.9	**29**		660.52	4941.0
	15.321	114.61	15	3	182 65	1366.4	29	3	671.96	5026 6
	15.90	118.97	15	6	188 69	1411.5	29	6	683.49	5112.9
	16.50	123.42	15	9	194.83	1457.4	29	9	695.13	5199.9
	17.10	127.95	**16**		201.06	1504.1	**30**		706.86	5287.7
	17.72	132.56	16	3	207.39	1551.4	30	3	718.69	5376.2
	18.35	137.25	16	6	213 82	1599.5	30	6	730.62	5465 4
	18.99	142 02	16	9	220.35	1648.4	30	9	742.64	5555.4
	19.63	146.88	**17**		226.98	1697.9	**31**		754.77	5646.1
	20.29	151 82	17	3	231.71	1748.2	31	3	766.99	5737.5
	20.97	156.83	17	6	240.53	1799.3	31	6	779.31	5829.7
	21.65	161.93	17	9	247.45	1851.1	31	9	791.73	5922.6
	22.34	167.12	18		254 47	1903.6	**32**		804 25	6016.2
	23.04	172.38	18	3	261.59	1956.8	32	3	816.86	6110.6
	23.76	177.72	18	6	268.80	2010.8	32	6	829.58	6205.7
	24.48	183.15	18	9	276.12	2065.5	32	9	842.39	6301.5

CONTENTS IN CUBIC FEET AND U. S. GALLO PIPES AND CYLINDERS OF VARIOUS DIAME AND ONE FOOT IN LENGTH.

1 gallon = 231 cubic inches. 1 cubic foot = 7.4805 gallons.

Diameter in Inches.	For 1 Foot in Length.		Diameter in Inches.	For 1 Foot in Length.		Diameter in Inches.	For 1 F Len
	Cubic Ft. also Area in Sq. Ft.	U. S. Gals., 231 Cu. In.		Cubic Ft. also Area in Sq. Ft.	U. S. Gals., 231 Cu. In.		Cubic Ft also Area in Sq. Ft
¼	.0003	.0025	6¾	.2485	1.859	19	1.969
5-16	.0005	.004	7	.2673	1.999	19½	2.074
⅜	.0008	.0057	7¼	.2867	2.145	20	2.182
7-16	.001	.0078	7½	.3068	2.295	20½	2.292
½	.0014	.0102	7¾	.3276	2.45	21	2.405
9-16	.0017	.0129	8	.3491	2.611	21½	2.521
⅝	.0021	.0159	8¼	.3712	2.777	22	2.640
11-16	.0026	.0193	8½	.3941	2.948	22½	2.761
¾	.0031	.0230	8¾	.4176	3.125	23	2.885
13-16	.0036	.0269	9	.4418	3.305	23½	3.012
⅞	.0042	.0312	9¼	.4667	3.491	24	3.142
15-16	.0048	.0359	9½	.4922	3.682	25	3.409
1	.0055	.0408	9¾	.5185	3.879	26	3.687
1¼	.0085	.0638	10	.5454	4.08	27	3.976
1½	.0123	.0918	10¼	.5730	4.286	28	4.276
1¾	.0167	.1249	10½	.6013	4.498	29	4.587
2	.0218	.1632	10¾	.6303	4.715	30	4.909
2¼	.0276	.2066	11	.66	4.937	31	5.241
2½	.0341	.2550	11¼	.6903	5.164	32	5.585
2¾	.0412	.3085	11½	.7213	5.396	33	5.940
3	.0491	.3672	11¾	.7530	5.633	34	6.305
3¼	.0576	.4309	12	.7854	5.875	35	6.681
3½	.0668	.4998	12½	.8522	6.375	36	7.069
3¾	.0767	.5738	13	.9218	6.895	37	7.467
4	.0873	.6528	13½	.994	7.436	38	7.876
4¼	.0985	.7369	14	1.069	7.997	39	8.296
4½	.1134	.8263	14½	1.147	8.578	40	8.727
4¾	.1231	.9206	15	1.227	9.180	41	9.168
5	.1364	1.020	15½	1.310	9.801	42	9.621
5¼	.1503	1.125	16	1.396	10.44	43	10.085
5½	.1650	1.234	16½	1.485	11.11	44	10.559
5¾	.1803	1.349	17	1.576	11.79	45	11.045
6	.1963	1.469	17½	1.670	12.49	46	11.541
6¼	.2131	1.594	18	1.768	13.22	47	12.048
6½	.2304	1.724	18½	1.867	13.96	48	12.566

To find the capacity of pipes greater than the largest given in look in the table for a pipe of one half the given size, and multiply ity by 4; or one of one third its size, and multiply its capacity by 9

To find the *weight* of water in any of the given sizes multiply the in cubic feet by 62¼ or the gallons by 8⅓, or, if a closer approxi required, by the weight of a cubic foot of water at the actual tempe the pipe.

Given the dimensions of a cylinder in inches, to find its capacit gallons: Square the diameter, multiply by the length and by .0034

$$\text{allons} = \frac{d^2 \times .7854 \times l}{231} = .0034d^2l.$$

DRICAL VESSELS, TANKS, CISTERNS, ETC.

r in Feet and Inches, Area in Square Feet, and S. Gallons Capacity for One Foot in Depth.

lon = 231 cubic inches = $\frac{1 \text{ cubic foot}}{7.4805}$ = 0.13368 cubic feet.

rea.	Gals.	Diam.	Area.	Gals.	Diam.	Area.	Gals.
. ft.	1 foot depth.	Ft. In.	Sq. ft.	1 foot depth.	Ft. In.	Sq. ft.	1 foot depth.
.785	5 87	5 8	25.22	188 66	**19**	283.53	2120.9
.922	6.89	5 9	25.97	194.25	19 3	291.04	2177.1
.069	8.00	5 10	26.73	199.92	19 6	298.65	2234.0
.227	9.18	5 11	27.49	205.67	19 9	306.35	2291.7
.396	10.44	**6**	28.27	211.51	**20**	314.16	2350.1
.576	11.79	6 3	30.68	229.50	20 3	322.06	2409.2
.767	13.22	6 6	33.18	248.23	20 6	330.06	2469.1
.969	14.73	6 9	35.78	267.69	20 9	338 16	2529.6
.182	16.32	**7**	38.48	287.88	**21**	346.36	2591.0
.405	17.99	7 3	41.28	308.81	21 3	354.66	2653.0
.640	19.75	7 6	44.18	330.48	21 6	363.05	2715.8
.885	21.58	7 9	47.17	352.88	21 9	371.54	2779.3
.142	23.50	**8**	50.27	376.01	**22**	380.13	2843.6
.409	25.50	8 3	53.46	399.88	22 3	388.82	2908.6
.687	27.58	8 6	56.75	424.48	22 6	397.61	2974.3
.976	29.74	8 9	60.13	449.82	22 9	406.49	3040.8
.276	31.99	**9**	63.62	475.89	**23**	415.48	3108.0
.587	34 31	9 3	67 20	502.70	23 3	424.56	3175.9
.909	36.72	9 6	70.88	530 24	23 6	433.74	3244.6
.241	39.21	9 9	74.66	558.51	23 9	443 01	3314 0
.585	41.78	**10**	78.54	587.52	**24**	452.39	3384.1
.940	44.43	10 3	82.52	617.26	24 3	461.86	3455.0
.305	47.16	10 6	86.59	647.74	24 6	471.44	3526.6
.681	49 98	10 9	90.76	678.95	24 9	481.11	3598.9
7.069	52 88	**11**	95.03	710.90	**25**	490.87	3672.0
7.467	55.86	11 3	99.40	743 58	25 3	500.74	3745.8
7.876	58.92	11 6	103.87	776.99	25 6	510.71	3820 3
8.296	62.06	11 9	108.43	811.14	25 9	520.77	3895.6
8.727	65.28	**12**	113.10	846.03	**26**	530.93	3971.6
9.168	68.58	12 3	117.86	881.65	26 3	541.19	4048.4
9.621	71.97	12 6	122.72	918.00	26 6	551.55	4125.9
0.085	75.44	12 9	127.68	955.09	26 9	562.00	4204.1
0.559	78.99	**13**	132.73	992.91	**27**	572.56	4283.0
1.045	82 62	13 3	137.89	1031.5	27 3	583.21	4362.7
1.541	86.33	13 6	143.14	1070.8	27 6	593.96	4443.1
2.048	90.13	13 9	148.49	1110.8	27 9	604.81	4524.3
2.566	94.00	**14**	153.94	1151.5	**28**	615.75	4606.2
3.095	97.96	14 3	159.48	1193.0	28 3	626.80	4688.8
3.635	102.00	14 6	165.13	1235.3	28 6	637.94	4772 1
4.186	106.12	14 9	170.87	1278.2	28 9	649.18	4856.2
4.748	110.32	**15**	176.71	1321.9	**29**	660.52	4941.0
5.321	114.61	15 3	182 65	1366.4	29 3	671.96	5026 6
5.90	118.97	15 6	188 69	1411.5	29 6	683.49	5112.9
6.50	123.42	15 9	194.83	1457.4	29 9	695.13	5199.9
7.10	127.95	**16**	201.06	1504.1	**30**	706.86	5287.7
7.72	132.56	16 3	207.39	1551.4	30 3	718.69	5376.2
8.35	137.25	16 6	213 82	1599.5	30 6	730.62	5465 4
18.99	142 02	16 9	220.35	1648.4	30 9	742.64	5555.4
19.63	146.88	**17**	226.98	1697.9	**31**	754.77	5646.1
20.29	151.82	17 3	233.71	1748.2	31 3	766.99	5737.5
20.97	156.83	17 6	240.53	1799.3	31 6	779.31	5829.7
21.65	161.93	17 9	247.45	1851.1	31 9	791.73	5922.6
22.34	167.12	**18**	254 47	1903.6	**32**	804 25	6016.2
23.04	*172.38*	*18 3*	*261.59*	*1956.8*	32 3	816.86	6110.6
.78	*177.72*	*18 6*	*268.80*	*2010.8*	32 6	829.58	6205.7
.48	*183.15*	*18 9*	*276.12*	*2065.5*	32 9	842.39	6301.5

GALLONS AND CUBIC FEET.

tates Gallons in a given Number of Cubic Feet.

= 7.480519 U. S. gallons; 1 gallon = 231 cu. in. = .13368056 cu. ft.

Gallons.	Cubic Ft.	Gallons.	Cubic Ft.	Gallons.
0.75	50	374.0	8,000	59,844.2
1.50	60	448.8	9,000	67,324.7
2.24	70	523.6	10,000	74,805.2
2.99	80	598.4	20,000	149,610.4
3.74	90	673.2	30,000	224,415.6
4.49	100	748.0	40,000	299,220.8
5.24	200	1,496.1	50,000	374,025.9
5.98	300	2,244.2	60,000	448,831.1
6.73	400	2,992.2	70,000	523,636.3
7.48	500	3,740.3	80,000	598,441.5
14.96	600	4,488.3	90,000	673,246.7
22.44	700	5,236.4	100,000	748,051.9
29.92	800	5,984.4	200,000	1,496,103.8
37.40	900	6,732.5	300,000	2,244,155.7
44.88	1,000	7,480.5	400,000	2,992,207.6
52.36	2,000	14,961.0	500,000	3,740,259.5
59.84	3,000	22,441.6	600,000	4,488,311.4
67.32	4,000	29,922.1	700,000	5,236,363.3
74.80	5,000	37,402.6	800,000	5,984,415.2
149.6	6,000	44,883.1	900,000	6,732,467.1
224.4	7,000	52,363.6	1,000,000	7,480,519.0
299.2				

ubic Feet in a given Number of Gallons.

Cubic Ft.	Gallons.	Cubic Ft.	Gallons.	Cubic Ft.
.134	1,000	133.681	1,000,000	133,680.6
.267	2,000	267.361	2,000,000	267,361.1
.401	3,000	401.042	3,000,000	401,041.7
.535	4,000	534.722	4,000,000	534,722.2
.668	5,000	668.403	5,000,000	668,402.8
.802	6,000	802.083	6,000,000	802,083.3
.936	7,000	935.764	7,000,000	935,763.9
1.069	8,000	1,069.444	8,000,000	1,069,444.4
1.203	9,000	1,203.125	9,000,000	1,203,125.0
1.337	10,000	1,336.806	10,000,000	1,336,805.6

CAPACITIES OF RECTANGULAR TANKS IN U. S. GALLONS, FOR EACH FOOT IN DEPTH.

1 cubic foot = 7.4805 U. S. gallons.

Width of Tank.	Length of Tank.										
	feet. 2	ft. in. 2 6	feet. 3	ft. in. 3 6	feet. 4	ft. in. 4 6	feet. 5	ft. in. 5 6	feet. 6	ft. in. 6 6	feet. 7
ft. in.											
2	29.92	37.40	44.88	52.36	59.84	67.32	74.81	82.29	89.77	97.25	104.73
2 6	......	46.75	56.10	65.45	74.80	84.16	93.51	102.86	112.21	121.56	130.91
3	......	......	67.32	78.54	89.77	100.99	112.21	123.43	134.65	145.87	157.09
3 6	......	......	......	91.64	104.73	117.82	130.91	144.00	157.09	170.18	183.27
4	......	......	......	......	119.69	134.65	149.61	164.57	179.53	194.49	209.45
4 6	......	......	......	......	......	151.48	168.31	185.14	201.97	218.80	235.63
5	......	......	......	......	......	......	187.01	205.71	224.41	243.11	261.82
5 6	......	......	......	......	......	......	......	226.28	246.86	267.43	288.00
6	......	......	......	......	......	......	......	......	269.30	291.74	314.18
6 6	......	......	......	......	......	......	......	......	......	316.05	340.36
7	......	......	......	......	......	......	......	......	......	......	366.54

Width of Tank.	Length of Tank.										
	ft. in. 7 6	feet. 8	ft. in. 8 6	feet. 9	ft. in. 9 6	feet. 10	ft. in. 10 6	feet. 11	ft. in. 11 6	feet. 12
ft. in.										
2	112.21	119.69	127.17	134.65	142.13	149.61	157.09	164.57	172.05	179 53
2 6	140.26	149.61	158.96	168.31	177.66	187.01	196.36	205.71	215.06	224.41
3	168.31	179.53	190.75	202.97	213.19	224 41	235.63	246.86	258.07	269.30
3 6	196.36	209.45	222.54	235.63	248.73	261.82	274.90	288.00	301.09	314.18
4	224.41	239.37	254.34	269.30	284.26	299.22	314.18	329.14	344.10	359.06
4 6	252.47	269.30	286.13	302.96	319.79	336.62	353.45	370.28	387.11	403.94
5	280.52	299.22	317.92	336.62	355.32	374.03	392.72	411.43	430.13	448.83
5 6	308.57	329.14	349.71	370.28	390.85	411.43	432.00	452.57	473.14	493.71
6	336.62	359.06	381.50	403.94	426.39	448.83	471.27	493.71	516.15	538.59
6 6	364.67	388.98	413.30	437.60	461.92	486.23	510.54	534.85	559.16	583.47
7	392.72	418.91	445 09	471 27	497.45	523.64	549.81	575.99	602.18	628 36
7 6	420.78	448.83	476.88	504.93	532.98	561 04	589.08	617.14	645.19	673.24
8	......	478.75	508.67	538.59	568.51	598.44	628.36	658.28	688.20	718.12
8 6	......	......	540.46	572.25	604.05	635.84	667 63	699.42	731.21	763.00
9	......	......	......	605.92	639.58	673.25	706.90	740.56	774.23	807.89
9 6	......	......	......	......	675.11	710.65	746.17	781.71	817.24	852.77
10	......	......	......	......	......	748.05	785.45	822.86	860.26	897.66
10 6	......	......	......	......	......	......	824.73	864.00	903.26	942.56
11	......	......	......	......	......	......	......	905.14	946.27	987.43
11 6	......	......	......	......	......	......	......	......	989.29	1032.3
12	......	......	......	......	......	......	......	......	......	1077.2

SQUARE FEET IN PLATES—(*Continued.*)

Ft. and Ins. Long.	Ins. Long.	Square Feet.	Ft. and Ins. Long.	Ins. Long.	Square Feet.	Ft. and Ins. Long.	Ins. Long.
17. 6	210	1.458	22. 5	269	1.868	27. 4	328
7	211	1.465	6	270	1.875	5	329
8	212	1.472	7	271	1.882	6	330
9	213	1.479	8	272	1.889	7	331
10	214	1.486	9	273	1.896	8	332
11	215	1.493	10	274	1.903	9	333
18. 0	216	1.5	11	275	1.91	10	334
1	217	1.507	23. 0	276	1.917	11	335
2	218	1.514	1	277	1.924	28. 0	336
3	219	1.521	2	278	1.931	1	337
4	220	1.528	3	279	1.938	2	338
5	221	1.535	4	280	1.944	3	339
6	222	1.542	5	281	1.951	4	340
7	223	1.549	6	282	1.958	5	341
8	224	1.556	7	283	1.965	6	342
9	225	1.563	8	284	1.972	7	343
0	226	1.569	9	285	1.979	8	344
11	227	1.576	10	286	1.986	9	345
19. 0	228	1.583	11	287	1.993	10	346
1	229	1.59	24. 0	288	2.	11	347
2	230	1.597	1	289	2.007	29. 0	348
3	231	1.604	2	290	2.014	1	349
4	232	1.611	3	291	2.021	2	350
5	233	1.618	4	292	2.028	3	351
6	234	1.625	5	293	2.035	4	352
7	235	1.632	6	294	2.042	5	353
8	236	1.639	7	295	2.049	6	354
9	237	1.645	8	296	2.056	7	355
10	238	1.653	9	297	2.063	8	356
11	239	1.659	10	298	2.069	9	357
20. 0	240	1.667	11	299	2.076	10	358
1	241	1.674	25. 0	300	2.083	11	359
2	242	1.681	1	301	2.09	30. 0	360
3	243	1.688	2	302	2.097	1	361
4	244	1.694	3	303	2.104	2	362
5	245	1.701	4	304	2.111	3	363
6	246	1.708	5	305	2.118	4	364
7	247	1.715	6	306	2.125	5	365
8	248	1.722	7	307	2.132	6	366
9	249	1.729	8	308	2.139	7	367
10	250	1.736	9	309	2.146	8	368
11	251	1.743	10	310	2.153	9	369
21. 0	252	1.75	11	311	2.16	10	370
1	253	1.757	26. 0	312	2.167	11	371
2	254	1.764	1	313	2.174	31. 0	372
3	255	1.771	2	314	2.181	1	373
4	256	1.778	3	315	2.188	2	374
5	257	1.785	4	316	2.194	3	375
6	258	1.792	5	317	2.201	4	376
7	259	1.799	6	318	2.208	5	377
8	260	1.806	7	319	2.215	6	378
9	261	1.813	8	320	2.222	7	379
10	262	1.819	9	321	2.229	8	380
11	263	1.826	10	322	2.236	9	381
22. 0	264	1.833	11	323	2.243	10	382
1	265	1.84	27. 0	324	2.25	11	383
2	266	1.847	1	325	2.257	32. 0	384
3	267	1.854	2	326	2.264	1	385
4	268	1.861	3	327	2.271	2	386

NUMBER OF BARRELS (31 1-2 GALLONS) IN CISTERNS AND TANKS.—*Continued.*

Depth in Feet.	Diameter in Feet.							
	23	24	25	26	27	28	29	30
1	98.666	107.432	116.571	126.083	135.968	146.226	157.858	167.862
5	493.3	537.2	582.9	630.4	679.8	731.1	784.3	839.3
6	592 0	644.6	699.4	756.5	815.8	877.4	941.1	1007.2
7	690.7	752.0	816.0	882.6	951.8	1023.6	1098.0	1175.0
8	789.3	859.5	932.6	1008.7	1087.7	1169.8	1254.9	1342.9
9	888.0	966.9	1049.1	1134.7	1223.7	1316.0	1411.7	1510.8
10	986.7	1074.3	1165.7	1260.8	1359.7	1462.2	1568.6	1678 6
11	1085.3	1181.8	1282.3	1386.9	1495.6	1608.5	1725.4	1846.5
12	1184.0	1289.2	1398.8	1513.0	1631.6	1754.7	1882.3	2014.4
13	1282.7	1396.6	1515.4	1639.1	1767.6	1900.9	2039.2	2182.2
14	1381.3	1504 0	1632.0	1765.2	1903.6	2047.2	2196.0	2350.1
15	1480.0	1611.5	1748.6	1891.2	2039.5	2193.4	2352.9	2517.9
16	1578.7	1718.9	1865.1	2017.3	2175.5	2339.6	2509.7	2685.8
17	1677.3	1826.3	1981.7	2143.4	2311.5	2485.8	2666.6	2853.7
18	1776.0	1933.8	2098.3	2269.5	2447.4	2632.0	2823.4	3021.5
19	1874.7	2041.2	2214.8	2395.6	2583.4	2778.3	2980.3	3189.4
20	1973.3	2148.6	2321.4	2521.7	2719.4	2924.5	3137.2	3357.3

LOGARITHMS.

Logarithms (abbreviation *log*).—The log of a number is the exponent of the power to which it is necessary to raise a fixed number to produce the given number. The fixed number is called the *base*. Thus if the base is 10, the log of 1000 is 3, for $10^3 = 1000$. There are two systems of logs in general use, the *common*, in which the base is 10, and the Naperian, or *hyperbolic*, in which the base is 2.718281828 .... The Naperian base is commonly denoted by e, as in the equation $e^y = x$, in which y is the Nap. log of x.

In any system of logs, the log of 1 is 0; the log of the base, taken in that system, is 1. In any system the base of which is greater than 1, the logs of all numbers greater than 1 are positive and the logs of all numbers less than 1 are negative.

The modulus of any system is equal to the reciprocal of the Naperian log of the base of that system. The modulus of the Naperian system is 1, that of the common system is .4342945.

The log of a number in any system equals the modulus of that system × the Naperian log of the number.

The *hyperbolic* or *Naperian* log of any number equals the common log × 2.3025851.

Every log consists of two parts, an entire part called the *characteristic*, or index, and the decimal part, or *mantissa*. The mantissa only is given in the usual tables of common logs, with the decimal point omitted. The characteristic is found by a simple rule, viz., it is one less than the number of figures to the left of the decimal point in the number whose log is to be found. Thus the characteristic of numbers from 1 to 9.99 + is 0, from 10 to 99.99 + is 1, from 100 to 999 + is 2, from .1 to .99 + is − 1, from .01 to .099 + is − 2, etc. Thus

log of 2000 is 3.30103;	log of .2 is − 1.30103;
" " 200 " 2.30103;	" " .02 " − 2.30103;
" " 20 " 1.30103;	" " .002 " − 3.30103;
" " 2 " 0.30103;	" " .0002 " − 4.30103.

The minus sign is frequently written above the characteristic th log .002 = $\bar{3}$.30103. The characteristic only is negative, the decimal part, mantissa, being always positive.

When a log consists of a negative index and a positive mantissa, it is us to write the negative sign over the index, or else to add 10 to the index, a to indicate the subtraction of 10 from the resulting logarithm.

Thus log .2 = $\bar{1}$.30103, and this may be written 9.30103 − 10.

In tables of logarithmic sines, etc., the − 10 is generally omitted, as be understood.

Rules for use of the table of Logarithms.—**To find th log of any whole number.**—For 1 to 100 inclusive the log is giv complete in the small table on page 129.

For 100 to 999 inclusive the decimal part of the log is given opposite given number in the column headed 0 in the table (including the two figur to the left, making six figures). Prefix the characteristic, or index. 2

For 1000 to 9999 inclusive: The last four figures of the log are fou opposite the first three figures of the given number and in the verti column headed with the fourth figure of the given number; prefix the figures under column 0, and the index, which is 3.

For numbers over 10,000 having five or more digits: Find the decimal p of the log for the first four digits as above, multiply the difference fig in the last column by the remaining digit or digits, and divide by 10 if th be only one digit more, by 100 if there be two more, and so on; add quotient to the log of the first four digits and prefix the index, which if there are five digits, 5 if there are six digits, and so on. The table of p portional parts may be used, as shown below.

To find the log of a decimal fraction or of a wh number and a decimal.—First find the log of the quantity as if th were no decimal point, then prefix the index according to rule; the inde one less than the number of figures to the left of the decimal point.

Required log of 3.141593.

log of	3.141	= 0.497068.	Diff. = 138
From proportional parts	5	= 690	
" " "	09	= 1242	
" " "	003	= 041	
log	3.141593	0.4971498	

To find the number corresponding to a given log.—F in the table the log nearest to the decimal part of the given log and take first four digits of the required number from the column N and the top foot of the column containing the log which is the next less than the g log. To find the 5th and 6th digits subtract the log in the table from given log, multiply the difference by 100, and divide by the figure in Diff. column opposite the log; annex the quotient to the four digits alre found, and place the decimal point according to the rule; the number figures to the left of the decimal point is one greater than the index.

Find number corresponding to the log	0.497150
Next lowest log in table corresponds to 3141......	.497068
Diff. =	82

Tabular diff. = 138; 82 ÷ 138 = .59 +

The index being 0, the number is therefore 3.14159 +.

To multiply two numbers by the use of logarithm Add together the logs of the two numbers, and find the number whose is the sum.

To divide two numbers.—Subtract the log of the less from log of the greater, and find the number whose log is the difference.

To raise a number to any given power.—Multiply the lo the number by the exponent of the power, and find the number whose l the product.

To find any root of a given number.—Divide the log of number by the index of the root. The quotient is the log of the root.

To find the reciprocal of a number.—Subtract the dec part of th[illegible] number from 0, add 1 to the index and change the the [illegible] ult is the log of the reciprocal.

Required the reciprocal of 3.141593.

Log of 3.141593, as found above..............................	0.4971498
Subtract decimal part from 0 gives..........................	0.5028502
Add 1 to the index, and changing sign of the index gives..	$\overline{1}$.5028502

which is the log of 0.31831.

To find the fourth term of a proportion by logarithms. Add the logarithms of the second and third terms, and from their sum subtract the logarithm of the first term.

When one logarithm is to be subtracted from another, it may be more convenient to convert the subtraction into an addition, which may be done by first subtracting the given logarithm from 10, adding the difference to the other logarithm, and afterwards rejecting the 10.

The difference between a given logarithm and 10 is called its *arithmetical complement*, or *cologarithm*.

To subtract one logarithm from another is the same as to add its complement and then reject 10 from the result. For $a - b = 10 - b + a - 10$.

To work a proportion, then, by logarithms, add the complement of the logarithm of the first term to the logarithms of the second and third terms. The characteristic must afterwards be diminished by 10.

Example in logarithms with a negative index.—Solve by logarithms $\left(\frac{526}{1011}\right)^{2.45}$, which means divide 526 by 1011 and raise the quotient to the 2.45 power.

$$
\begin{array}{rr}
\log 526 = & 2.720986 \\
\log 1011 = & 3.004751 \\
\hline
\text{log of quotient} = & -\,1.716235 \\
\text{Multiply by} & 2.45 \\
\hline
& -\,2.581175 \\
& -\,2.8\;64940 \\
& -\,1.43\;2470 \\
\hline
& -\,1.30\;477575 = .20173,\ \textit{Ans.}
\end{array}
$$

In multiplying $-\,\overline{1}.7$ by 5, we say: $5 \times 7 = 35$, 3 to carry; $5 \times -1 = -5$ less 3 carried $= -2$. In adding $-2 + 8 + 3 + 1$ carried from previous column, we say: $1 + 3 + 8 = 12$, minus $2 = 10$, set down 0 and carry 1; $1 + 4 - 2 = 3$.

LOGARITHMS OF NUMBERS FROM 1 TO 100.

N.	Log.	N.	Log.	N.	Log.	N.	Log.	N.	Log.
1	0.000000	21	1.322219	41	1.612784	61	1.785330	81	1.908485
2	0.301030	22	1.342423	42	1.623249	62	1.792392	82	1.913814
3	0.477121	23	1.361728	43	1.633468	63	1.799341	83	1.919078
4	0.602060	24	1.380211	44	1.643453	64	1.806180	84	1.924279
5	0.698970	25	1.397940	45	1.653213	65	1.812913	85	1.929419
6	0.778151	26	1.414973	46	1.662758	66	1.819544	86	1.934498
7	0.845098	27	1.431364	47	1.672098	67	1.826075	87	1.939519
8	0.903090	28	1.447158	48	1.681241	68	1.832509	88	1.944483
9	0.954243	29	1.462398	49	1.690196	69	1.838849	89	1.949390
10	1.000000	30	1.477121	50	1.698970	70	1.845098	90	1.954243
11	1.041393	31	1.491362	51	1.707570	71	1.851258	91	1.959041
12	1.079181	32	1.505150	52	1.716003	72	1.857332	92	1.963788
13	1.113943	33	1.518514	53	1.724276	73	1.863323	93	1.968483
14	1.146128	34	1.531479	54	1.732394	74	1.869232	94	1.973128
15	1.176091	35	1.544068	55	1.740363	75	1.875061	95	1.977724
16	1.204120	36	1.556303	56	1.748188	76	1.880814	96	1.982271
17	1.230449	37	1.568202	57	1.755875	77	1.886491	97	1.986772
18	1.255273	38	1.579784	58	1.763428	78	1.892095	98	1.991226
19	1.278754	39	1.591065	59	1.770852	79	1.897627	99	1.995635
20	1.301030	40	1.602060	60	1.778151	80	1.903090	100	[illegible]

No. 100 L. 000.] [No. 109 L.

N.	0	1	2	3	4	5	6	7	8	9
100	000000	0434	0868	1301	1734	2166	2598	3029	3461	3891
1	4321	4751	5181	5609	6038	6466	6894	7321	7748	8174
2	8600	9026	9451	9876	0300	0724	1147	1570	1993	2415
3	012837	3259	3680	4100	4521	4940	5360	5779	6197	6616
4	7033	7451	7868	8284	8700	9116	9532	9947	0361	0775
5	021189	1603	2016	2428	2841	3252	3664	4075	4486	4896
6	5306	5715	6125	6533	6942	7350	7757	8164	8571	8978
7	9384	9789	0195	0600	1004	1408	1812	2216	2619	3021
8	033424	3826	4227	4628	5029	5430	5830	6230	6629	7028
9	7426	7825	8223	8620	9017	9414	9811	0207	0602	0998
	04									

PROPORTIONAL PARTS.

Diff.	1	2	3	4	5	6	7	8
434	43.4	86.8	130.2	173.6	217.0	260.4	303.8	347.2
433	43.3	86.6	129.9	173.2	216.5	259.8	303.1	346.4
432	43.2	86.4	129.6	172.8	216.0	259.2	302.4	345.6
431	43.1	86.2	129.3	172.4	215.5	258.6	301.7	344.8
430	43.0	86.0	129.0	172.0	215.0	258.0	301.0	344.0
429	42.9	85.8	128.7	171.6	214.5	257.4	300.3	343.2
428	42.8	85.6	128.4	171.2	214.0	256.8	299.6	342.4
427	42.7	85.4	128.1	170.8	213.5	256.2	298.9	341.6
426	42.6	85.2	127.8	170.4	213.0	255.6	298.2	340.8
425	42.5	85.0	127.5	170.0	212.5	255.0	297.5	340.0
424	42.4	84.8	127.2	169.6	212.0	254.4	296.8	339.2
423	42.3	84.6	126.9	169.2	211.5	253.8	296.1	338.4
422	42.2	84.4	126.6	168.8	211.0	253.2	295.4	337.6
421	42.1	84.2	126.3	168.4	210.5	252.6	294.7	336.8
420	42.0	84.0	126.0	168.0	210.0	252.0	294.0	336.0
419	41.9	83.8	125.7	167.6	209.5	251.4	293.3	335.2
418	41.8	83.6	125.4	167.2	209.0	250.8	292.6	334.4
417	41.7	83.4	125.1	166.8	208.5	250.2	291.9	333.6
416	41.6	83.2	124.8	166.4	208.0	249.6	291.2	332.8
415	41.5	83.0	124.5	166.0	207.5	249.0	290.5	332.0
414	41.4	82.8	124.2	165.6	207.0	248.4	289.8	331.2
413	41.3	82.6	123.9	165.2	206.5	247.8	289.1	330.4
412	41.2	82.4	123.6	164.8	206.0	247.2	288.4	329.6
411	41.1	82.2	123.3	164.4	205.5	246.6	287.7	328.8
410	41.0	82.0	123.0	164.0	205.0	246.0	287.0	328.0
409	40.9	81.8	122.7	163.6	204.5	245.4	286.3	327.2
408	40.8	81.6	122.4	163.2	204.0	244.8	285.6	326.4
407	40.7	81.4	122.1	162.8	203.5	244.2	284.9	325.6
406	40.6	81.2	121.8	162.4	203.0	243 6	284.2	324.8
405	40.5	81.0	121.5	162.0	202.5	243.0	283.5	324.0
404	40.4	80.8	121.2	161.6	202.0	242.4	282.8	323.2
403	40.3	80.6	120.9	161.2	201.5	241.8	282.1	322.4
402	40.2	80.4	120.6	160.8	201.0	241 2	281.4	321.6
401	40.1	80.2	120.3	160.4	200.5	240.6	280.7	320.8
400	40.0	80·0	120.0	160.0	200.0	240.0	280.0	320.0
399	39.9	79.8	119.7	159.6	199.5	239.4	279.3	319.2
398	39.8	79.6	119.4	159.2	199.0	238.8	278.6	318.4
[illegible]	[illegible]	[illegible]	119.1	158.8	198.5	238.2	277.9	317.6
[illegible]	[illegible]	[illegible]	[illegible]	158.4	198.0	237.6	277.2	316.8
[illegible]	[illegible]	[illegible]	[illegible]	158.0	197.5	237.0	276.5	316.0

1.] [No. 119 L. 078.

1	2	3	4	5	6	7	8	9	Diff.
1787	2182	2576	2969	3362	3755	4148	4540	4932	393
5714	6105	6495	6885	7275	7664	8053	8442	8830	390
9606	9993	0380	0766	1153	1538	1924	2309	2694	386
3463	3846	4230	4613	4996	5378	5760	6142	6524	383
7286	7666	8046	8426	8805	9185	9563	9942	0320	379
1075	1452	1829	2206	2582	2958	3333	3709	4083	376
4882	5206	5580	5953	6326	6699	7071	7443	7815	373
8557	8928	9298	9668	0038	0407	0776	1145	1514	370
2250	2617	2985	3352	3718	4085	4451	4816	5182	366
5912	6276	6640	7004	7368	7731	8094	8457	8819	363

PROPORTIONAL PARTS.

2	3	4	5	6	7	8	9
79.0	118.5	158.0	197.5	237.0	276.5	316.0	355.5
78.8	118.2	157.6	197.0	236.4	275.8	315.2	354.6
8.6	117.9	157.2	196.5	235.8	275.1	314.4	353.7
8.4	117.6	156.8	196.0	235.2	274.4	313.6	352.8
8.2	117.3	156.4	195.5	234.6	273.7	312.8	351.9
8.0	117.0	156.0	195.0	234.0	273.0	312.0	351.0
7.8	116.7	155.6	194.5	233.4	272.3	311.2	350.1
.6	116.4	155.2	194.0	232.8	271.6	310.4	349.2
.4	116.1	154.8	193.5	232.2	270.9	309.6	348.3
.2	115.8	154.4	193.0	231.6	270.2	308.8	347.4
.0	115.5	154.0	192.5	231.0	269.5	308.0	346.5
.8	115.2	153.6	192.0	230.4	268.8	307.2	345.6
6	114.9	153.2	191.5	229.8	268.1	306.4	344.7
4	114.6	152.8	191.0	229.2	267.4	305.6	343.8
2	114.3	152.4	190.5	228.6	266.7	304.8	342.9
0	114.0	152.0	190.0	228.0	266.0	304.0	342.0
8	113.7	151.6	189.5	227.4	265.3	303.2	341.1
6	113.4	151.2	189.0	226.8	264.6	302.4	340.2
4	113.1	150.8	188.5	226.2	263.9	301.6	339.3
2	112.8	150.4	188.0	225.6	263.2	300.8	338.4
[illegible]	112.5	150.0	187.5	225.0	262.5	300.0	337.5
	112.2	149.6	187.0	224.4	261.8	299.2	336.6
	111.9	149.2	186.5	223.8	261.1	298.4	335.7
	111.6	148.8	186.0	223.2	260.4	297.6	334.8
	111.3	148.4	185.5	222.6	259.7	296.8	333.9
	111.0	148.0	185.0	222.0	259.0	296.0	333.0
	110.7	147.6	184.5	221.4	258.3	295.2	332.1
	110.4	147.2	184.0	220.8	257.6	294.4	331.2
	110.1	146.8	183.5	220.2	256.9	293.6	330.3
	109.8	146.4	183.0	219.6	256.2	292.8	329.4
	109.5	146.0	182.5	219.0	255.7	292.0	328.5
	109.2	145.6	182.0	218.4	254.8	291.2	327.6
	108.9	145.2	181.5	217.8	254.1	290.4	326.7
	108.6	144.8	181.0	217.2	253.4	289.6	325.8
	108.3	144.4	180.5	216.6	252.7	288.8	324.9
	108.0	144.0	180.0	216.0	252.0	288.0	324.0
	107.7	143.6	179.5	215.4	251.3	287.2	323.1
	07.4	143.2	179.0	214.8	250.6	286.4	322.2
	7.1	142.8	178.5	214.2	249.9	285.6	321.3
	.8	142.4	178.0	213.6	249.2	284.8	320.4

No. 120 L. 079.] [No. 134 L. 130

N.	0	1	2	3	4	5	6	7	8	9	Diff.
120	079181	9543	9904	0266	0626	0987	1347	1707	2067	2426	360
1	082785	3144	3503	3861	4219	4576	4934	5291	5647	6004	357
2	6360	6716	7071	7426	7781	8136	8490	8845	9198	9552	355
3	9905	0258	0611	0963	1315	1667	2018	2370	2721	3071	352
4	093422	3772	4122	4471	4820	5169	5518	5866	6215	6562	349
5	6910	7257	7604	7951	8298	8644	8990	9335	9681	0026	346
6	100371	0715	1059	1403	1747	2091	2434	2777	3119	3462	343
7	3804	4146	4487	4828	5169	5510	5851	6191	6531	6871	341
8	7210	7549	7888	8227	8565	8903	9241	9579	9916	0253	338
9	110590	0926	1263	1599	1934	2270	2605	2940	3275	3609	335
130	3943	4277	4611	4944	5278	5611	5943	6276	6608	6940	333
1	7271	7603	7934	8265	8595	8926	9256	9586	9915	0245	330
2	120574	0903	1231	1560	1888	2216	2544	2871	3198	3525	328
3	3852	4178	4504	4830	5156	5481	5806	6131	6456	6781	325
4	7105	7429	7753	8076	8399	8722	9045	9368	9690	0012	323
	13										

PROPORTIONAL PARTS.

Diff.	1	2	3	4	5	6	7	8	9
355	35.5	71.0	106.5	142.0	177.5	213.0	248.5	284.0	319.5
354	35.4	70.8	106.2	141.6	177.0	212.4	247.8	283.2	318.6
353	35.3	70.6	105.9	141.2	176.5	211.8	247.1	282.4	317.7
352	35.2	70.4	105.6	140.8	176.0	211.2	246.4	281.6	316.8
351	35.1	70.2	105.3	140.4	175.5	210.6	245.7	280.8	315.9
350	35.0	70.0	105.0	140.0	175.0	210.0	245.0	280.0	315.0
349	34.9	69.8	104.7	139.6	174.5	209.4	244.3	279.2	314.1
348	34.8	69.6	104.4	139.2	174.0	208.8	243.6	278.4	313.2
347	34.7	69.4	104.1	138.8	173.5	208.2	242.9	277.6	312.3
346	34.6	69.2	103.8	138.4	173.0	207.6	242.2	276.8	311.4
345	34.5	69.0	103.5	138.0	172.5	207.0	241.5	276.0	310.5
344	34.4	68.8	103.2	137.6	172.0	206.4	240.8	275.2	309.6
343	34.3	68.6	102.9	137.2	171.5	205.8	240.1	274.4	308.7
342	34.2	68.4	102.6	136.8	171.0	205.2	239.4	273.6	307.8
341	34.1	68.2	102.3	136.4	170.5	204.6	238.7	272.8	306.9
340	34.0	68.0	102.0	136.0	170.0	204.0	238.0	272.0	306.0
339	33.9	67.8	101.7	135.6	169.5	203.4	237.3	271.2	305.1
338	33.8	67.6	101.4	135.2	169.0	202.8	236.6	270.4	304.2
337	33.7	67.4	101.1	134.8	168.5	202.2	235.9	269.6	303.3
336	33.6	67.2	100.8	134.4	168.0	201.6	235.2	268.8	302.4
335	33.5	67.0	100.5	134.0	167.5	201.0	234.5	268.0	301.5
334	33.4	66.8	100.2	133.6	167.0	200.4	233.8	267.2	300.6
333	33.3	66.6	99.9	133.2	166.5	199.8	233.1	266.4	299.7
332	33.2	66.4	99.6	132.8	166.0	199.2	232.4	265.6	298.8
331	33.1	66.2	99.3	132.4	165.5	198.6	231.7	264.8	297.9
330	33.0	66.0	99.0	132.0	165.0	198.0	231.0	264.0	297.0
329	32.9	65.8	98.7	131.6	164.5	197.4	230.3	263.2	
328	32.8	65.6	98.4	131.2	164.0	196.8	229.6	262.4	
327	32.7	65.4	98.1	130.8	163.5	196.2	228.9	261.6	
326	32.6	65.2	97.8	130.4	163.0	195.6	228.2	260.8	
325	32.5	65.0	97.5	130.0	162.5	195.0	227.5	260.0	
24	32.4	64.8	97.2	129.6	162.0	194.4	226.8	259.2	
3	32.3	64.6	96.9	129.2	161.5	193.8	226.1	258.4	
2	32.2	64.4	96.6	128.8	161.0	193.2	225.4	257.6	

1	2	3	4	5	6	7	8	9	Diff.
0655	0977	1298	1619	1939	2260	2580	2900	3219	321
3858	4177	4496	4814	5133	5451	5769	6086	6403	318
7037	7354	7671	7987	8303	8618	8934	9249	9564	316
0194	0508	0822	1136	1450	1763	2076	2389	2702	314
3327	3639	3951	4263	4574	4885	5196	5507	5818	311
6438	6748	7058	7367	7676	7985	8294	8603	8911	309
9527	9835	0142	0449	0756	1063	1370	1676	1982	307
2594	2900	3205	3510	3815	4120	4424	4728	5032	305
5640	5943	6246	6549	6852	7154	7457	7759	8061	303
8664	8965	9266	9567	9868	0168	0469	0769	1068	301
1667	1967	2266	2564	2863	3161	3460	3758	4055	299
4650	4947	5244	5541	5838	6134	6430	6726	7022	297
7613	7908	8203	8497	8792	9086	9380	9674	9968	295
0555	0848	1141	1434	1726	2019	2311	2603	2895	293
3478	3769	4060	4351	4641	4932	5222	5512	5802	291

Proportional Parts.

2	3	4	5	6	7	8	9
64.2	96.3	128.4	160.5	192.6	224.7	256.8	288.9
64.0	96.0	128.0	160.0	192.0	224.0	256.0	288.0
63.8	95.7	127.6	159.5	191.4	223.3	255.2	287.1
63.6	95.4	127.2	159.0	190.8	222.6	254.4	286.2
63.4	95.1	126.8	158.5	190.2	221.9	253.6	285.3
63.2	94.8	126.4	158.0	189.6	221.2	252.8	284.4
63.0	94.5	126.0	157.5	189.0	220.5	252.0	283.5
62.8	94.2	125.6	157.0	188.4	219.8	251.2	282.6
62.6	93.9	125.2	156.5	187.8	219.1	250.4	281.7
2.4	93.6	124.8	156.0	187.2	218.4	249.6	280.8
2.2	93.3	124.4	155.5	186.6	217.7	248.8	279.9
2.0	93.0	124.0	155.0	186.0	217.0	248.0	279.0
1.8	92.7	123.6	154.5	185.4	216.3	247.2	278.1
1.6	92.4	123.2	154.0	184.8	215.6	246.4	277.2
1.4	92.1	122.8	153.5	184.2	214.9	245.6	276.3
1.2	91.8	122.4	153.0	183.6	214.2	244.8	275.4
1.0	91.5	122.0	152.5	183.0	213.5	244.0	274.5
.8	91.2	121.6	152.0	182.4	212.8	243.2	273.6
.6	90.9	121.2	151.5	181.8	212.1	242.4	272.7
.4	90.6	120.8	151.0	181.2	211.4	241.6	271.8
.2	90.3	120.4	150.5	180.6	210.7	240.8	270.9
.0	90.0	120.0	150.0	180.0	210.0	240.0	270.0
.8	89.7	119.6	149.5	179.4	209.3	239.2	269.1
.6	89.4	119.2	149.0	178.8	208.6	238.4	268.2
.4	89.1	118.8	148.5	178.2	207.9	237.6	267.3
.2	88.8	118.4	148.0	177.6	207.2	236.8	266.4
.0	88.5	118.0	147.5	177.0	206.5	236.0	265.5
.8	88.2	117.6	147.0	176.4	205.8	235.2	264.6
.6	87.9	117.2	146.5	175.8	205.1	234.4	263.7
.4	87.6	116.8	146.0	175.2	204.4	233.6	262.8
.2	87.3	116.4	145.5	174.6	203.7	232.8	261.9
.0	87.0	116.0	145.0	174.0	203.0	232.0	261.0
.8	86.7	115.6	144.5	173.4	202.3	231.2	260.1
6	86.4	115.2	144.0	172.8	201.6	[illegible]	[illegible]
4	86.1	114.8	143.5	172.2	200.9	[illegible]	[illegible]
[illegible]	85.8	114.4	143.0	171.6	200.2	[illegible]	[illegible]

No. 150 L. 176.] [No

N.	0	1	2	3	4	5	6	7	8
150	176091	6381	6670	6959	7248	7536	7825	8113	8401
1	8977	9264	9552	9839	0126	0413	0699	0986	1272
2	181844	2129	2415	2700	2985	3270	3555	3839	4123
3	4691	4975	5259	5542	5825	6108	6391	6674	6956
4	7521	7803	8084	8366	8647	8928	9209	9490	9771
5	190332	0612	0892	1171	1451	1730	2010	2289	2567
6	3125	3403	3681	3959	4237	4514	4792	5069	5346
7	5900	6176	6453	6729	7005	7281	7556	7832	8107
8	8657	8932	9206	9481	9755	0029	0303	0577	0850
9	201397	1670	1943	2216	2488	2761	3033	3305	3577
160	4120	4391	4663	4934	5204	5475	5746	6016	6286
1	6826	7096	7365	7634	7904	8173	8441	8710	8979
2	9515	9783	0051	0319	0586	0853	1121	1388	1654
3	212188	2454	2720	2986	3252	3518	3783	4049	4314
4	4844	5109	5373	5638	5902	6166	6430	6694	6957
5	7484	7747	8010	8273	8536	8798	9060	9323	9585
6	220108	0370	0631	0892	1153	1414	1675	1936	2196
7	2716	2976	3236	3496	3755	4015	4274	4533	4792
8	5309	5568	5826	6084	6342	6600	6858	7115	7372
9	7887	8144	8400	8657	8913	9170	9426	9682	9938
	23								

PROPORTIONAL PARTS.

Diff.	1	2	3	4	5	6	7
285	28.5	57.0	85.5	114.0	142.5	171.0	199.5
284	28.4	56.8	85.2	113.6	142.0	170.4	198.8
283	28.3	56.6	84.9	113.2	141.5	169.8	198.1
282	28.2	56.4	84.6	112.8	141.0	169.2	197.4
281	28.1	56.2	84.3	112.4	140.5	168.6	196.7
280	28.0	56.0	84.0	112.0	140.0	168.0	196.0
279	27.9	55.8	83.7	111.6	139.5	167.4	195.3
278	27.8	55.6	83.4	111.2	139.0	166.8	194.6
277	27.7	55.4	83.1	110.8	138.5	166.2	193.9
276	27.6	55.2	82.8	110.4	138.0	165.6	193.2
275	27.5	55.0	82.5	110.0	137.5	165.0	192.5
274	27.4	54.8	82.2	109.6	137.0	164.4	191.8
273	27.3	54.6	81.9	109.2	136.5	163.8	191.1
272	27.2	54.4	81.6	108.8	136.0	163.2	190.4
271	27.1	54.2	81.3	108.4	135.5	162.6	189.7
270	27.0	54.0	81.0	108.0	135.0	162.0	189.0
269	26.9	53.8	80.7	107.6	134.5	161.4	188.3
268	26.8	53.6	80.4	107.2	134.0	160.8	187.6
267	26.7	53.4	80.1	106.8	133.5	160.2	186.9
266	26.6	53.2	79.8	106.4	133.0	159.6	186.2
265	26.5	53.0	79.5	106.0	132.5	159.0	185.5
264	26.4	52.8	79.2	105.6	132.0	158.4	184.8
263	26.3	52.6	78.9	105.2	131.5	157.8	184.1
262	26.2	52.4	78.6	104.8	131.0	157.2	183.4
261	26.1	52.2	78.3	104.4	130.5	156.6	182.7
260	26.0	52.0	78.0	104.0	130.0	156.0	182.0
259	25.9	51.8	77.7	103.6	129.5	155.4	181.3
258	25.8	51.6	77.4	103.2	129.0	154.8	180.6
257	[illegible]	[illegible]	77.1	102.8	128.5	154.2	179.9
256	[illegible]	[illegible]	76.8	102.4	128.0	153.6	179.2
5	[illegible]	[illegible]	76.5	102.0	127.5	153.0	178.5

1	2	3	4	5	6	7	8	9	Diff.
0704	0960	1215	1470	1724	1979	2234	2488	2742	255
3250	3504	3757	4011	4264	4517	4770	5023	5276	253
5781	6033	6285	6537	6789	7041	7292	7544	7795	252
8297	8548	8799	9049	9299	9550	9800	0050	0300	250
0799	1048	1297	1546	1795	2044	2293	2541	2790	249
3286	3534	3782	4030	4277	4525	4772	5019	5266	248
5759	6006	6252	6499	6745	6991	7237	7482	7728	246
8219	8464	8709	8954	9198	9443	9687	9932	0176	245
0664	0908	1151	1395	1638	1881	2125	2368	2610	243
3096	3338	3580	3822	4064	4306	4548	4790	5031	242
5514	5755	5996	6237	6477	6718	6958	7198	7439	241
7918	8158	8398	8637	8877	9116	9355	9594	9833	239
0310	0548	0787	1025	1263	1501	1739	1976	2214	238
2688	2925	3162	3399	3636	3873	4109	4346	4582	237
5054	5290	5525	5761	5906	6232	6467	6702	6937	235
7406	7641	7875	8110	8344	8578	8812	9046	9279	234
9746	9980	0213	0446	0679	0912	1144	1377	1609	233
2074	2306	2538	2770	3001	3233	3464	3696	3927	232
4389	4620	4850	5081	5311	5542	5772	6002	6232	230
6692	6921	7151	7380	7609	7838	8067	8296	8525	229

PROPORTIONAL PARTS.

2	3	4	5	6	7	8	9
1.0	76.5	102.0	127.5	153.0	178.5	204.0	229.5
0.8	76.2	101.6	127.0	152.4	177.8	203.2	228.6
0.6	75.9	101.2	126.5	151.8	177.1	202.4	227.7
.4	75.6	100.8	126.0	151.2	176.4	201.6	226.8
.2	75.3	100.4	125.5	150.6	175.7	200.8	225.9
.0	75.0	100.0	125.0	150.0	175.0	200.0	225.0
.8	74.7	99.6	124.5	149.4	174.3	199.2	224.1
.6	74.4	99.2	124.0	148.8	173.6	198.4	223.2
4	74.1	98.8	123.5	148.2	172.9	197.6	222.3
2	73.8	98.4	123.0	147.6	172.2	196.8	221.4
0	73.5	98.0	122.5	147.0	171.5	196.0	220.5
	73.2	97.6	122.0	146.4	170.8	195.2	219.6
	72.9	97.2	121.5	145.8	170.1	194.4	218.7
	72.6	96.8	121.0	145.2	169.4	193.6	217.8
	72.3	96.4	120.5	144.6	168.7	192.8	216.9
	72.0	96.0	120.0	144.0	168.0	192.0	216.0
	71.7	95.6	119.5	143.4	167.3	191.2	215.1
	71.4	95.2	119.0	142.8	166.6	190.4	214.2
	71.1	94.8	118.5	142.2	165.9	189.6	213.3
	70.8	94.4	118.0	141.6	165.2	188.8	212.4
	70.5	94.0	117.5	141.0	164.5	188.0	211.5
	70.2	93.6	117.0	140.4	163.8	187.2	210.6
	69.9	93.2	116.5	139.8	163.1	186.4	209.7
	69.6	92.8	116.0	139.2	162.4	185.6	208.8
	69.3	92.4	115.5	138.6	161.7	184.8	207.9
	69.0	92.0	115.0	138.0	161.0	184.0	207.0
	68.7	91.6	114.5	137.4	160.3	183.2	206.1
	68.4	91.2	114.0	136.8	159.6	182.4	205.2
	68.1	90.8	113.5	136.2	158.9	181.6	204.3
	67.8	90.4	113.0	135.6	158.2	180.8	203.4

No. 190 L. 278.] [No. 214

N.	0	1	2	3	4	5	6	7	8	9
190	278754	8982	9211	9439	9667	9895	0123	0351	0578	0806
1	281033	1261	1488	1715	1942	2169	2396	2622	2849	3075
2	3301	3527	3753	3979	4205	4431	4656	4882	5107	5332
3	5557	5782	6007	6232	6456	6681	6905	7130	7354	7578
4	7802	8026	8249	8473	8696	8920	9143	9366	9589	9812
5	290035	0257	0480	0702	0925	1147	1369	1591	1813	2034
6	2256	2478	2699	2920	3141	3363	3584	3804	4025	4246
7	4466	4687	4907	5127	5347	5567	5787	6007	6226	6446
8	6665	6884	7104	7323	7542	7761	7979	8198	8416	8635
9	8853	9071	9289	9507	9725	9943	0161	0378	0595	0813
200	301030	1247	1464	1681	1898	2114	2331	2547	2764	2980
1	3196	3412	3628	3844	4059	4275	4491	4706	4921	5136
2	5351	5566	5781	5996	6211	6425	6639	6854	7068	7282
3	7496	7710	7924	8137	8351	8564	8778	8991	9204	9417
4	9630	9843	0056	0268	0481	0693	0906	1118	1330	1542
5	311754	1966	2177	2389	2600	2812	3023	3234	3445	3656
6	3867	4078	4289	4499	4710	4920	5130	5340	5551	5760
7	5970	6180	6390	6599	6809	7018	7227	7436	7646	7854
8	8063	8272	8481	8689	8898	9106	9314	9522	9730	9938
9	320146	0354	0562	0769	0977	1184	1391	1598	1805	2012
210	2219	2426	2633	2839	3046	3252	3458	3665	3871	4077
1	4282	4488	4694	4899	5105	5310	5516	5721	5926	6131
2	6336	6541	6745	6950	7155	7359	7563	7767	7972	8176
3	8380	8583	8787	8991	9194	9398	9601	9805	0008	0211
4	330414	0617	0819	1022	1225	1427	1630	1832	2034	2236

PROPORTIONAL PARTS.

Diff.	1	2	3	4	5	6	7	8
225	22.5	45.0	67.5	90.0	112.5	135.0	157.5	180.0
224	22.4	44.8	67.2	89.6	112.0	134.4	156.8	179.2
223	22.3	44.6	66.9	89.2	111.5	133.8	156.1	178.4
222	22.2	44.4	66.6	88.8	111.0	133.2	155.4	177.6
221	22.1	44.2	66.3	88.4	110.5	132.6	154.7	176.8
220	22.0	44.0	66.0	88.0	110.0	132.0	154.0	176.0
219	21.9	43.8	65.7	87.6	109.5	131.4	153.3	175.2
218	21.8	43.6	65.4	87.2	109.0	130.8	152.6	174.4
217	21.7	43.4	65.1	86.8	108.5	130.2	151.9	173.6
216	21.6	43.2	64.8	86.4	108.0	129.6	151.2	172.8
215	21.5	43.0	64.5	86.0	107.5	129.0	150.5	172.0
214	21.4	42.8	64.2	85.6	107.0	128.4	149.8	171.2
213	21.3	42.6	63.9	85.2	106.5	127.8	149.1	170.4
212	21.2	42.4	63.6	84.8	106.0	127.2	148.4	169.6
211	21.1	42.2	63.3	84.4	105.5	126.6	147.7	168.8
210	21.0	42.0	63.0	84.0	105.0	126.0	147.0	168.0
209	20.9	41.8	62.7	83.6	104.5	125.4	146.3	167.2
208	20.8	41.6	62.4	83.2	104.0	124.8	145.6	166.4
207	20.7	41.4	62.1	82.8	103.5	124.2	144.9	165.6
″	20.6	41.2	61.8	82.4	103.0	123.6	144.2	164.8
			61.5	82.0	102.5	123.0	143.5	164.0
			61.2	81.6	102.0	122.4	142.8	163.2
			60.9	81.2	101.5	121.8	142.1	[illegible]
			[illegible]	[illegible]	101.0	121.2	141.4	[illegible]

2.] [No. 239 L. 380.

1	2	3	4	5	6	7	8	9	Diff.
2640	2842	3044	3246	3447	3649	3850	4051	4253	202
4655	4856	5057	5257	5458	5658	5859	6059	6260	201
6660	6860	7060	7260	7459	7659	7858	8058	8257	200
8656	8855	9054	9253	9451	9650	9849	0047	0246	199
0642	0841	1039	1237	1435	1632	1830	2028	2225	198
2620	2817	3014	3212	3409	3606	3802	3999	4196	197
4589	4785	4981	5178	5374	5570	5766	5962	6157	196
6549	6744	6939	7135	7330	7525	7720	7915	8110	195
8500	8694	8889	9083	9278	9472	9666	9860	0054	194
0442	0636	0829	1023	1216	1410	1603	1796	1989	193
2375	2568	2761	2954	3147	3339	3532	3724	3916	193
4301	4493	4685	4876	5068	5260	5452	5643	5834	192
6217	6408	6599	6790	6981	7172	7363	7554	7744	191
8125	8316	8506	8696	8886	9076	9266	9456	9646	190
0025	0215	0404	0593	0783	0972	1161	1350	1539	189
1917	2105	2294	2482	2671	2859	3048	3236	3424	188
3800	3988	4176	4363	4551	4739	4926	5113	5301	188
5675	5862	6049	6236	6423	6610	6796	6983	7169	187
7542	7729	7915	8101	8287	8473	8659	8845	9030	186
9401	9587	9772	9958	0143	0328	0513	0698	0883	185
1253	1437	1622	1806	1991	2175	2360	2544	2728	184
3096	3280	3464	3647	3831	4015	4198	4382	4565	184
4932	5115	5298	5481	5664	5846	6029	6212	6394	183
6759	6942	7124	7306	7488	7670	7852	8034	8216	182
8580	8761	8943	9124	9306	9487	9668	9849	0030	181

PROPORTIONAL PARTS.

2	3	4	5	6	7	8	9
0.4	60.6	80.8	101.0	121.2	141.4	161.6	181.8
0.2	60.3	80.4	100.5	120.6	140.7	160.8	180.9
0.0	60.0	80.0	100.0	120.0	140.0	160.0	180.0
9.8	59.7	79.6	99.5	119.4	139.3	159.2	179.1
9.6	59.4	79.2	99.0	118.8	138.6	158.4	178.2
.4	59.1	78.8	98.5	118.2	137.9	157.6	177.3
.2	58.8	78.4	98.0	117.6	137.2	156.8	176.4
.0	58.5	78.0	97.5	117.0	136.5	156.0	175.5
.8	58.2	77.6	97.0	116.4	135.8	155.2	174.6
.6	57.9	77.2	96.5	115.8	135.1	154.4	173.7
.4	57.6	76.8	96.0	115.2	134.4	153.6	172.8
.2	57.3	76.4	95.5	114.6	133.7	152.8	171.9
0	57.0	76.0	95.0	114.0	133.0	152.0	171.0
8	56.7	75.6	94.5	113.4	132.3	151.2	170.1
6	56.4	75.2	94.0	112.8	131.6	150.4	169.2
4	56.1	74.8	93.5	112.2	130.9	149.6	168.3
2	55.8	74.4	93.0	111.6	130.2	148.8	167.4
	55.5	74.0	92.5	111.0	129.5	148.0	166.5
	55.2	73.6	92.0	110.4	128.8	147.2	165.6
	54.9	73.2	91.5	109.8	128.1	146.4	164.7
	54.6	72.8	91.0	109.2	127.4	145.6	163.8
	54.3	72.4	90.5	108.6	126.7	144.8	162.9
	54.0	72.0	90.0	108.0	126.0	144.0	162.0
	53.7	71.6	89.5	107.4	125.3	143.2	161.1

No. 240 L. 380.] [No. 26

N.	0	1	2	3	4	5	6	7	8	9
240	380211	0392	0573	0754	0934	1115	1296	1476	1656	183
1	2017	2197	2377	2557	2737	2917	3097	3277	3456	363
2	3815	3995	4174	4353	4533	4712	4891	5070	5249	542
3	5606	5785	5964	6142	6321	6499	6677	6856	7034	721
4	7390	7568	7746	7924	8101	8279	8456	8634	8811	898
5	9166	9343	9520	9698	9875					
						0051	0228	0405	0582	075
6	390935	1112	1288	1464	1641	1817	1993	2169	2345	252
7	2697	2873	3048	3224	3400	3575	3751	3926	4101	427
8	4452	4627	4802	4977	5152	5326	5501	5676	5850	602
9	6199	6374	6548	6722	6896	7071	7245	7419	7592	776
250	7940	8114	8287	8461	8634	8808	8981	9154	9328	950
1	9674	9847								
			0020	0192	0365	0538	0711	0883	1056	122
2	401401	1573	1745	1917	2089	2261	2433	2605	2777	294
3	3121	3292	3464	3635	3807	3978	4149	4320	4492	466
4	4834	5005	5176	5346	5517	5688	5858	6029	6199	637
5	6540	6710	6881	7051	7221	7391	7561	7731	7901	807
6	8240	8410	8579	8749	8918	9087	9257	9426	9595	976
7	9933									
		0102	0271	0440	0609	0777	0946	1114	1283	145
8	411620	1788	1956	2124	2293	2461	2629	2796	2964	313
9	3300	3467	3635	3803	3970	4137	4305	4472	4639	480
260	4973	5140	5307	5474	5641	5808	5974	6141	6308	647
1	6641	6807	6973	7139	7306	7472	7638	7804	7970	813
2	8301	8467	8633	8798	8964	9129	9295	9460	9625	979
3	9956									
		0121	0286	0451	0616	0781	0945	1110	1275	143
4	421604	1768	1933	2097	2261	2426	2590	2754	2918	308
5	3246	3410	3574	3737	3901	4065	4228	4392	4555	471
6	4882	5045	5208	5371	5534	5697	5860	6023	6186	634
7	6511	6674	6836	6999	7161	7324	7486	7648	7811	797
8	8135	8297	8459	8621	8783	8944	9106	9268	9429	959
9	9752	9914								
	43		0075	0236	0398	0559	0720	0881	1042	120

PROPORTIONAL PARTS.

Diff.	1	2	3	4	5	6	7	8
178	17.8	35.6	53.4	71.2	89.0	106.8	124.6	142.
177	17.7	35.4	53.1	70.8	88.5	106.2	123.9	141.
176	17.6	35.2	52.8	70.4	88.0	105.6	123.2	140.
175	17.5	35.0	52.5	70.0	87.5	105.0	122.5	140.
174	17.4	34.8	52.2	69.6	87.0	104.4	121.8	139.
173	17.3	34.6	51.9	69.2	86.5	103.8	121.1	138.
172	17.2	34.4	51.6	68.8	86.0	103.2	120.4	137.
171	17.1	34.2	51.3	68.4	85.5	102.6	119.7	136.
170	17.0	34.0	51.0	68.0	85.0	102.0	119.0	136.
169	16.9	33.8	50.7	67.6	84.5	101.4	118.3	135.
168	16.8	33.6	50.4	67.2	84.0	100.8	117.6	134.
167	16.7	33.4	50.1	66.8	83.5	100.2	116.9	133.
166	16.6	33.2	49.8	66.4	83.0	99.6	116.2	132.
165	16.5	33.0	49.5	66.0	82.5	99.0	115.5	132.
[illegible]	16.4	32.8	49.2	65.6	82.0	98.4	114.8	131.
[illegible]	[illegible]	[illegible]	48.9	65.2	81.5	97.8	114.1	130.
[illegible]	[illegible]	[illegible]	[illegible]	64.8	81.0	97.2	113.4	129
[illegible]	[illegible]	[illegible]	[illegible]	64.4	80.5	96.6	112.7	12

0	1	2	3	4	5	6	7	8	9	Diff.
431364	1525	1685	1846	2007	2167	2328	2488	2649	2809	161
2969	3130	3290	3450	3610	3770	3930	4090	4249	4409	160
4569	4729	4888	5048	5207	5367	5526	5685	5844	6004	159
6163	6322	6481	6640	6799	6957	7116	7275	7433	7592	159
7751	7909	8067	8226	8384	8542	8701	8859	9017	9175	158
9333	9491	9648	9806	9964	0122	0279	0437	0594	0752	158
440909	1066	1224	1381	1538	1695	1852	2009	2166	2323	157
2480	2637	2793	2950	3106	3263	3419	3576	3732	3889	157
4045	4201	4357	4513	4669	4825	4981	5137	5293	5449	156
5604	5760	5915	6071	6226	6382	6537	6692	6848	7003	155
7158	7313	7468	7623	7778	7933	8088	8242	8397	8552	155
8706	8861	9015	9170	9324	9478	9633	9787	9941	0095	154
450249	0403	0557	0711	0865	1018	1172	1326	1479	1633	154
1786	1940	2093	2247	2400	2553	2706	2859	3012	3165	153
3318	3471	3624	3777	3930	4082	4235	4387	4540	4692	153
4845	4997	5150	5302	5454	5606	5758	5910	6062	6214	152
6366	6518	6670	6821	6973	7125	7276	7428	7579	7731	152
7882	8033	8184	8336	8487	8638	8789	8940	9091	9242	151
9392	9543	9694	9845	9995	0146	0296	0447	0597	0748	151
460898	1048	1198	1348	1499	1649	1799	1948	2098	2248	150
2398	2548	2697	2847	2997	3146	3296	3445	3594	3744	150
3893	4042	4191	4340	4490	4639	4788	4936	5085	5234	149
5383	5532	5680	5829	5977	6126	6274	6423	6571	6719	149
6868	7016	7164	7312	7460	7608	7756	7904	8052	8200	148
8347	8495	8643	8790	8938	9085	9233	9380	9527	9675	148
9822	9969	0116	0263	0410	0557	0704	0851	0998	1145	147
471292	1438	1585	1732	1878	2025	2171	2318	2464	2610	146
2756	2903	3049	3195	3341	3487	3633	3779	3925	4071	146
4216	4362	4508	4653	4799	4944	5090	5235	5381	5526	146
5671	5816	5962	6107	6252	6397	6542	6687	6832	6976	145

PROPORTIONAL PARTS.

Diff.	1	2	3	4	5	6	7	8	9
161	16.1	32.2	48.3	64.4	80.5	96.6	112.7	128.8	144.9
160	16.0	32.0	48.0	64.0	80.0	96.0	112.0	128.0	144.0
159	15.9	31.8	47.7	63.6	79.5	95.4	111.3	127.2	143.1
158	15.8	31.6	47.4	63.2	79.0	94.8	110.6	126.4	142.2
157	15.7	31.4	47.1	62.8	78.5	94.2	109.9	125.6	141.3
156	15.6	31.2	46.8	62.4	78.0	93.6	109.2	124.8	140.4
155	15.5	31.0	46.5	62.0	77.5	93.0	108.5	124.0	139.5
154	15.4	30.8	46.2	61.6	77.0	92.4	107.8	123.2	138.6
153	15.3	30.6	45.9	61.2	76.5	91.8	107.1	122.4	137.7
152	15.2	30.4	45.6	60.8	76.0	91.2	106.4	121.6	136.8
151	15.1	30.2	45.3	60.4	75.5	90.6	105.7	120.8	135.9
150	15.0	30.0	45.0	60.0	75.0	90.0	105.0	120.0	135.0
149	14.9	29.8	44.7	59.6	74.5	89.4	104.3	119.2	134.1
148	14.8	29.6	44.4	59.2	74.0	88.8	103.6	118.4	133.2
147	14.7	29.4	44.1	58.8	73.5	88.2	102.9	117.6	132.3
146	14.6	29.2	43.8	58.4	73.0	87.6	102.2	116.8	131.4
145	14.5	29.0	43.5	58.0	72.5	87.0	101.5	116.0	130.5
144	14.4	28.8	43.2	57.6	72.0	86.4	100.8	115.2	129.6
143	14.3	28.6	42.9	57.2	71.5	85.8	100.1	114.4	128.7
142	14.2	28.4	42.6	56.8	71.0	85.2	99.4	113.6	127.8
141	14.1	28.2	42.3	56.4	70.5	84.6	98.7	112.8	126.
140	14.0	28.0	42.0	56.0	70.0	84.0	98.0	112.0	126

No. 300 L. 477.] [No. 339 L. 5

N.	0	1	2	3	4	5	6	7	8	9
300	477121	7266	7411	7555	7700	7844	7989	8133	8278	8422
1	8566	8711	8855	8999	9143	9287	9431	9575	9719	9863
2	480007	0151	0294	0438	0582	0725	0869	1012	1156	1299
3	1443	1586	1729	1872	2016	2159	2302	2445	2588	2731
4	2874	3016	3159	3302	3445	3587	3730	3872	4015	4157
5	4300	4442	4585	4727	4869	5011	5153	5295	5437	5579
6	5721	5863	6005	6147	6289	6430	6572	6714	6855	6997
7	7138	7280	7421	7563	7704	7845	7986	8127	8269	8410
8	8551	8692	8833	8974	9114	9255	9396	9537	9677	9818
9	9958									
		0099	0239	0380	0520	0661	0801	0941	1081	1222
310	491362	1502	1642	1782	1922	2062	2201	2341	2481	2621
1	2760	2900	3040	3179	3319	3458	3597	3737	3876	4015
2	4155	4294	4433	4572	4711	4850	4989	5128	5267	5406
3	5544	5683	5822	5960	6099	6238	6376	6515	6653	6791
4	6930	7068	7206	7344	7483	7621	7759	7897	8035	8173
5	8311	8448	8586	8724	8862	8999	9137	9275	9412	9550
6	9687	9824	9962							
				0099	0236	0374	0511	0648	0785	0922
7	501059	1196	1333	1470	1607	1744	1880	2017	2154	2291
8	2427	2564	2700	2837	2973	3109	3246	3382	3518	3655
9	3791	3927	4063	4199	4335	4471	4607	4743	4878	5014
320	5150	5286	5421	5557	5693	5828	5964	6099	6234	6370
1	6505	6640	6776	6911	7046	7181	7316	7451	7586	7721
2	7856	7991	8126	8260	8395	8530	8664	8799	8934	9068
3	9203	9337	9471	9606	9740	9874				
							0009	0143	0277	0411
4	510545	0679	0813	0947	1081	1215	1349	1482	1616	1750
5	1883	2017	2151	2284	2418	2551	2684	2818	2951	3084
6	3218	3351	3484	3617	3750	3883	4016	4149	4282	4415
7	4548	4681	4813	4946	5079	5211	5344	5476	5609	5741
8	5874	6006	6139	6271	6403	6535	6668	6800	6932	7064
9	7196	7328	7460	7592	7724	7855	7987	8119	8251	8382
330	8514	8646	8777	8909	9040	9171	9303	9434	9566	9697
1	9828	9959								
			0090	0221	0353	0484	0615	0745	0876	1007
2	521138	1269	1400	1530	1661	1792	1922	2053	2183	2314
3	2444	2575	2705	2835	2966	3096	3226	3356	3486	3616
4	3746	3876	4006	4136	4266	4396	4526	4656	4785	4915
5	5045	5174	5304	5434	5563	5693	5822	5951	6081	6210
6	6339	6469	6598	6727	6856	6985	7114	7243	7372	7501
7	7630	7759	7888	8016	8145	8274	8402	8531	8660	8788
8	8917	9045	9174	9302	9430	9559	9687	9815	9943	
										0072
9	530200	0328	0456	0584	0712	0840	0968	1096	1223	1351

PROPORTIONAL PARTS.

Diff.	1	2	3	4	5	6	7	8
139	13.9	27.8	41.7	55.6	69.5	83.4	97.3	111.2
138	13.8	27.6	41.4	55.2	69.0	82.8	96.6	110.4
137	13.7	27.4	41.1	54.8	68.5	82.2	95.9	109.6
136	13.6	27.2	40.8	54.4	68.0	81.6	95.2	108.8
135	13.5	27.0	40.5	54.0	67.5	81.0	94.5	108.0
134	13.4	26.8	40.2	53.6	67.0	80.4	93.8	107.2
133	13.3	26.6	39.9	53.2	66.5	79.8	93.1	106.4
132	13.2	26.4	39.6	52.8	66.0	79.2	92.4	105.6
131	13.1	26.2	39.3	52.4	65.5	78.6	91.7	104.8
130	13.0	26.0	39.0	52.0	65.0	78.0	91.0	104.0
129	12.9	25.8	38.7	51.6	64.5	77.4	90.3	103.
128	12.8	25.6	38.4	51.2	64.0	76.8	89.6	102.

0	1	2	3	4	5	6	7	8	9	Diff.
531479	1607	1734	1862	1990	2117	2245	2372	2500	2627	128
2754	2882	3009	3136	3264	3391	3518	3645	3772	3899	127
4026	4153	4280	4407	4534	4661	4787	4914	5041	5167	127
5294	5421	5547	5674	5800	5927	6053	6180	6306	6432	126
6558	6685	6811	6937	7063	7189	7315	7441	7567	7693	126
7819	7945	8071	8197	8322	8448	8574	8699	8825	8951	126
9076	9202	9327	9452	9578	9703	9829	9954	0079	0204	125
540329	0455	0580	0705	0830	0955	1080	1205	1330	1454	125
1579	1704	1829	1953	2078	2203	2327	2452	2576	2701	125
2825	2950	3074	3199	3323	3447	3571	3696	3820	3944	124
4068	4192	4316	4440	4564	4688	4812	4936	5060	5183	124
5307	5431	5555	5678	5802	5925	6049	6172	6296	6419	124
6543	6666	6789	6913	7036	7159	7282	7405	7529	7652	123
7775	7898	8021	8144	8267	8389	8512	8635	8758	8881	123
9003	9126	9249	9371	9494	9616	9739	9861	9984	0106	123
550228	0351	0473	0595	0717	0840	0962	1084	1206	1328	122
1450	1572	1694	1816	1938	2060	2181	2303	2425	2547	122
2668	2790	2911	3033	3155	3276	3398	3519	3640	3762	121
3883	4004	4126	4247	4368	4489	4610	4731	4852	4973	121
5094	5215	5336	5457	5578	5699	5820	5940	6061	6182	121
6303	6423	6544	6664	6785	6905	7026	7146	7267	7387	120
7507	7627	7748	7868	7988	8108	8228	8349	8469	8589	120
8709	8829	8948	9068	9188	9308	9428	9548	9667	9787	120
9907	0026	0146	0265	0385	0504	0624	0743	0863	0982	119
561101	1221	1340	1459	1578	1698	1817	1936	2055	2174	119
2293	2412	2531	2650	2769	2887	3006	3125	3244	3362	119
3481	3600	3718	3837	3955	4074	4192	4311	4429	4548	119
4666	4784	4903	5021	5139	5257	5376	5494	5612	5730	118
5848	5966	6084	6202	6320	6437	6555	6673	6791	6909	118
7026	7144	7262	7379	7497	7614	7732	7849	7967	8084	118
8202	8319	8436	8554	8671	8788	8905	9023	9140	9257	117
9374	9491	9608	9725	9842	9959	0076	0193	0309	0426	117
570543	0660	0776	0893	1010	1126	1243	1359	1476	1592	117
1709	1825	1942	2058	2174	2291	2407	2523	2639	2755	116
2872	2988	3104	3220	3336	3452	3568	3684	3800	3915	116
4031	4147	4263	4379	4494	4610	4726	4841	4957	5072	116
5188	5303	5419	5534	5650	5765	5880	5996	6111	6226	115
6341	6457	6572	6687	6802	6917	7032	7147	7262	7377	115
7492	7607	7722	7836	7951	8066	8181	8295	8410	8525	115
8639	8754	8868	8983	9097	9212	9326	9441	9555	9669	114

PROPORTIONAL PARTS.

Diff.	1	2	3	4	5	6	7	8	9
128	12.8	25.6	38.4	51.2	64.0	76.8	89.6	102.4	115.2
127	12.7	25.4	38.1	50.8	63.5	76.2	88.9	101.6	114.3
126	12.6	25.2	37.8	50.4	63.0	75.6	88.2	100.8	113.4
125	12.5	25.0	37.5	50.0	62.5	75.0	87.5	100.0	112.5
124	12.4	24.8	37.2	49.6	62.0	74.4	86.8	99.2	111.6
123	12.3	24.6	36.9	49.2	61.5	73.8	86.1	98.4	110.7
122	12.2	24.4	36.6	48.8	61.0	73.2	85.4	97.6	109.8
121	12.1	24.2	36.3	48.4	60.5	72.6	84.7	96.8	108.9
120	12.0	24.0	36.0	48.0	60.0	72.0	84.0	96.0	108.0
119	11.9	23.8	35.7	47.6	59.5	71.4	83.3	95.2	107.1

No. 300 L. 477.] [No. 339 L.

N.	0	1	2	3	4	5	6	7	8	9
300	477121	7266	7411	7555	7700	7844	7989	8133	8278	8422
1	8566	8711	8855	8999	9143	9287	9431	9575	9719	9863
2	480007	0151	0294	0438	0582	0725	0869	1012	1156	1299
3	1443	1586	1729	1872	2016	2159	2302	2445	2588	2731
4	2874	3016	3159	3302	3445	3587	3730	3872	4015	4157
5	4300	4442	4585	4727	4869	5011	5153	5295	5437	5579
6	5721	5863	6005	6147	6289	6430	6572	6714	6855	6997
7	7138	7280	7421	7563	7704	7845	7986	8127	8269	8410
8	8551	8692	8833	8974	9114	9255	9396	9537	9677	9818
9	9958	0099	0239	0380	0520	0661	0801	0941	1081	1222
310	491362	1502	1642	1782	1922	2062	2201	2341	2481	2621
1	2760	2900	3040	3179	3319	3458	3597	3737	3876	4015
2	4155	4294	4433	4572	4711	4850	4989	5128	5267	5406
3	5544	5683	5822	5960	6099	6238	6376	6515	6653	6791
4	6930	7068	7206	7344	7483	7621	7759	7897	8035	8173
5	8311	8448	8586	8724	8862	8999	9137	9275	9412	9550
6	9687	9824	9962	0099	0236	0374	0511	0648	0785	0922
7	501059	1196	1333	1470	1607	1744	1880	2017	2154	2291
8	2427	2564	2700	2837	2973	3109	3246	3382	3518	3655
9	3791	3927	4063	4199	4335	4471	4607	4743	4878	5014
320	5150	5286	5421	5557	5693	5828	5964	6099	6234	6370
1	6505	6640	6776	6911	7046	7181	7316	7451	7586	7721
2	7856	7991	8126	8260	8395	8530	8664	8799	8934	9068
3	9203	9337	9471	9606	9740	9874	0009	0143	0277	0411
4	510545	0679	0813	0947	1081	1215	1349	1482	1616	1750
5	1883	2017	2151	2284	2418	2551	2684	2818	2951	3084
6	3218	3351	3484	3617	3750	3883	4016	4149	4282	4415
7	4548	4681	4813	4946	5079	5211	5344	5476	5609	5741
8	5874	6006	6139	6271	6403	6535	6668	6800	6932	7064
9	7196	7328	7460	7592	7724	7855	7987	8119	8251	8382
330	8514	8646	8777	8909	9040	9171	9303	9434	9566	9697
1	9828	9959	0090	0221	0353	0484	0615	0745	0876	1007
2	521138	1269	1400	1530	1661	1792	1922	2053	2183	2314
3	2444	2575	2705	2835	2966	3096	3226	3356	3486	3616
4	3746	3876	4006	4136	4266	4396	4526	4656	4785	4915
5	5045	5174	5304	5434	5563	5693	5822	5951	6081	6210
6	6339	6469	6598	6727	6856	6985	7114	7243	7372	7501
7	7630	7759	7888	8016	8145	8274	8402	8531	8660	8788
8	8917	9045	9174	9302	9430	9559	9687	9815	9943	0072
9	530200	0328	0456	0584	0712	0840	0968	1096	1223	1351

PROPORTIONAL PARTS.

Diff.	1	2	3	4	5	6	7	8
139	13.9	27.8	41.7	55.6	69.5	83.4	97.3	111.2
138	13.8	27.6	41.4	55.2	69.0	82.8	96.6	110.4
137	13.7	27.4	41.1	54.8	68.5	82.2	95.9	109.6
136	13.6	27.2	40.8	54.4	68.0	81.6	95.2	108.8
135	13.5	27.0	40.5	54.0	67.5	81.0	94.5	108.0
134	13.4	26.8	40.2	53.6	67.0	80.4	93.8	107.2
133	13.3	26.6	39.9	53.2	66.5	79.8	93.1	106.4
132	13.2	26.4	39.6	52.8	66.0	79.2	92.4	105.6
131	13.1	26.2	39.3	52.4	65.5	78.6	91.7	104.8
[illegible]	[illegible]	[illegible]	39.0	52.0	65.0	78.0	91.0	104.0
[illegible]	[illegible]	[illegible]	38.7	51.6	64.5	77.4	90.3	103.2
[illegible]	[illegible]	[illegible]	38.4	51.2	64.0	76.8	89.6	102.[illegible]

[No. 459 L. 662

		3	4	5	6	7	8	9	Diff.
		8362	8466	8571	8676	8780	8884	8989	105
		9406	9511	9615	9719	9824	9928	0032	
		448	0552	0656	0760	0864	0968	1072	104
		1488	1592	1695	1799	1903	2007	2110	
		525	2628	2732	2835	2939	3042	3146	
		559	3663	3766	3869	3973	4076	4179	
		591	4695	4798	4901	5004	5107	5210	103
		621	5724	5827	5929	6032	6135	6238	
		648	6751	6853	6956	7058	7161	7263	
		673	7775	7878	7980	8082	8185	8287	
		695	8797	8900	9002	9104	9206	9308	102
		715	9817	9919	0021	0123	0224	0326	
		733	0835	0936	1038	1139	1241	1342	
		1748	1849	1951	2052	2153	2255	2356	
		2761	2862	2963	3064	3165	3266	3367	
		3771	3872	3973	4074	4175	4276	4376	101
		4779	4880	4981	5081	5182	5283	5383	
		5785	5886	5986	6087	6187	6287	6388	
		6789	6889	6989	7089	7189	7290	7390	
		7790	7890	7990	8090	8190	8290	8389	100
		8789	8888	8988	9088	9188	9287	9387	
		9785	9885	9984	0084	0183	0283	0382	
		0779	0879	0978	1077	1177	1276	1375	
		1771	1871	1970	2069	2168	2267	2366	
		2761	2860	2959	3058	3156	3255	3354	99
		3749	3847	3946	4044	4143	4242	4340	
		4734	4832	4931	5029	5127	5226	5324	
	)	5717	5815	5913	6011	6110	6208	6306	
	0	6698	6796	6894	6992	7089	7187	7285	98
	9	7676	7774	7872	7969	8067	8165	8262	
	5	8653	8750	8848	8945	9043	9140	9237	
	30	9627	9724	9821	9919	0016	0113	0210	
	502	0599	0696	0793	0890	0987	1084	1181	
	472	1569	1666	1762	1859	1956	2053	2150	97
	440	2536	2633	2730	2826	2923	3019	3116	
	3405	3502	3598	3695	3791	3888	3984	4080	
	4369	4465	4562	4658	4754	4850	4946	5042	
	5331	5427	5523	5619	5715	5810	5906	6002	96
	6290	6386	6482	6577	6673	6769	6864	6960	
	7247	7343	7438	7534	7629	7725	7820	7916	
	8202	8298	8393	8488	8584	8679	8774	8870	
)	9155	9250	9346	9441	9536	9631	9726	9821	
11	0106	0201	0296	0391	0486	0581	0676	0771	95
60	1055	1150	1245	1339	1434	1529	1623	1718	
07	2002	2096	2191	2286	2380	2475	2569	2663	

PROPORTIONAL PARTS.

2	3	4	5	6	7	8	9
21.0	31.5	42.0	52.5	63.0	73.5	84.0	94.5
20.8	31.2	41.6	52.0	62.4	72.8	83.2	93.6
20.6	30.9	41.2	51.5	61.8	72.1	82.4	92.7
0.4	30.6	40.8	51.0	61.2	71.4	81.6	91.8
.2	30.3	40.4	50.5	60.6	70.7	80.8	90.9
0	30.0	40.0	50.0	60.0	70.0	80.0	90.0
	29.7	39.6	49.5	59.4	69.3	79.2	89.1

No. 460 L. 662.] [No. 499 L.

N	0	1	2	3	4	5	6	7	8	9
460	662758	2852	2947	3041	3135	3230	3324	3418	3512	3607
1	3701	3795	3889	3983	4078	4172	4266	4360	4454	4548
2	4642	4736	4830	4924	5018	5112	5206	5299	5393	5487
3	5581	5675	5769	5862	5956	6050	6143	6237	6331	6424
4	6518	6612	6705	6799	6892	6986	7079	7173	7266	7360
5	7453	7546	7640	7733	7826	7920	8013	8106	8199	8293
6	8386	8479	8572	8665	8759	8852	8945	9038	9131	9224
7	9317	9410	9503	9596	9689	9782	9875	9967	0060	0153
8	670246	0339	0431	0524	0617	0710	0802	0895	0988	1080
9	1173	1265	1358	1451	1543	1636	1728	1821	1913	2005
470	2098	2190	2283	2375	2467	2560	2652	2744	2836	2929
1	3021	3113	3205	3297	3390	3482	3574	3666	3758	3850
2	3942	4034	4126	4218	4310	4402	4494	4586	4677	4769
3	4861	4953	5045	5137	5228	5320	5412	5503	5595	5687
4	5778	5870	5962	6053	6145	6236	6328	6419	6511	6602
5	6694	6785	6876	6968	7059	7151	7242	7333	7424	7516
6	7607	7698	7789	7881	7972	8063	8154	8245	8336	8427
7	8518	8609	8700	8791	8882	8973	9064	9155	9246	9337
8	9428	9519	9610	9700	9791	9882	9973	0063	0154	0245
9	680336	0426	0517	0607	0698	0789	0879	0970	1060	1151
480	1241	1332	1422	1513	1603	1693	1784	1874	1964	2055
1	2145	2235	2326	2416	2506	2596	2686	2777	2867	2957
2	3047	3137	3227	3317	3407	3497	3587	3677	3767	3857
3	3947	4037	4127	4217	4307	4396	4486	4576	4666	4756
4	4845	4935	5025	5114	5204	5294	5383	5473	5563	5652
5	5742	5831	5921	6010	6100	6189	6279	6368	6458	6547
6	6636	6726	6815	6904	6994	7083	7172	7261	7351	7440
7	7529	7618	7707	7796	7886	7975	8064	8153	8242	8331
8	8420	8509	8598	8687	8776	8865	8953	9042	9131	9220
9	9309	9398	9486	9575	9664	9753	9841	9930	0019	0107
490	690196	0285	0373	0462	0550	0639	0728	0816	0905	0993
1	1081	1170	1258	1347	1435	1524	1612	1700	1789	1877
2	1965	2053	2142	2230	2318	2406	2494	2583	2671	2759
3	2847	2935	3023	3111	3199	3287	3375	3463	3551	3639
4	3727	3815	3903	3991	4078	4166	4254	4342	4430	4517
5	4605	4693	4781	4868	4956	5044	5131	5219	5307	5394
6	5482	5569	5657	5744	5832	5919	6007	6094	6182	6269
7	6356	6444	6531	6618	6706	6793	6880	6968	7055	7142
8	7229	7317	7404	7491	7578	7665	7752	7839	7926	8014
9	8100	8188	8275	8362	8449	8535	8622	8709	8796	8883

PROPORTIONAL PARTS.

Diff.	1	2	3	4	5	6	7	8
98	9.8	19.6	29.4	39.2	49.0	58.8	68.6	78.4
97	9.7	19.4	29.1	38.8	48.5	58.2	67.9	77.6
96	9.6	19.2	28.8	38.4	48.0	57.6	67.2	76.8
95	9.5	19.0	28.5	38.0	47.5	57.0	66.5	76.0
94	9.4	18.8	28.2	37.6	47.0	56.4	65.8	75.2
93	9.3	18.6	27.9	37.2	46.5	55.8	65.1	74.4
92	9.2	18.4	27.6	36.8	46.0	55.2	64.4	73.6
91	9.1	18.2	27.3	36.4	45.5	54.6	63.7	72.8
90	9.0	18.0	27.0	36.0	45.0	54.0	63.0	72.0
[illegible]	[illegible]	[illegible]	26.7	35.6	44.5	53.4	62.3	71.2
[illegible]	[illegible]	[illegible]	26.4	35.2	44.0	52.8	61.6	70.4
[illegible]	[illegible]	[illegible]	.1	34.8	43.5	52.2	60.9	69.6
[illegible]	[illegible]	[illegible]	.8	34.4	43.0	51.6	60.2	68.8

[No. 544 L. 736.

2	3	4	5	6	7	8	9	Diff.
144	9231	9317	9404	9491	9578	9664	9751	
011	0098	0184	0271	0358	0444	0531	0617	
77	0963	1050	1136	1222	1309	1395	1482	
41	1827	1913	1999	2086	2172	2258	2344	
03	2689	2775	2861	2947	3033	3119	3205	
63	3549	3635	3721	3807	3893	3979	4065	86
22	4408	4494	4579	4665	4751	4837	4922	
79	5265	5350	5436	5522	5607	5693	5778	
35	6120	6206	6291	6376	6462	6547	6632	
88	6974	7059	7144	7229	7315	7400	7485	
40	7826	7911	7996	8081	8166	8251	8336	
91	8676	8761	8846	8931	9015	9100	9185	85
40	9524	9609	9694	9779	9863	9948	0033	
87	0371	0456	0540	0625	0710	0794	0879	
32	1217	1301	1385	1470	1554	1639	1723	
76	2060	2144	2229	2313	2397	2481	2566	
18	2902	2986	3070	3154	3238	3323	3407	
59	3742	3826	3910	3994	4078	4162	4246	84
97	4581	4665	4749	4833	4916	5000	5084	
35	5418	5502	5586	5669	5753	5836	5920	
70	6254	6337	6421	6504	6588	6671	6754	
04	7088	7171	7254	7338	7421	7504	7587	
37	7920	8003	8086	8169	8253	8336	8419	
68	8751	8834	8917	9000	9083	9165	9248	83
97	9580	9663	9745	9828	9911	9994	0077	
325	0407	0490	0573	0655	0738	0821	0903	
151	1233	1316	1398	1481	1563	1646	1728	
975	2058	2140	2222	2305	2387	2469	2552	
798	2881	2963	3045	3127	3209	3291	3374	
620	3702	3784	3866	3948	4030	4112	4194	82
440	4522	4604	4685	4767	4849	4931	5013	
258	5340	5422	5503	5585	5667	5748	5830	
075	6156	6238	6320	6401	6483	6564	6646	
890	6972	7053	7134	7216	7297	7379	7460	
704	7785	7866	7948	8029	8110	8191	8273	
8516	8597	8678	8759	8841	8922	9003	9084	
9327	9408	9489	9570	9651	9732	9813	9893	81
0136	0217	0298	0378	0459	0540	0621	0702	
0944	1024	1105	1186	1266	1347	1428	1508	
1750	1830	1911	1991	2072	2152	2233	2313	
2555	2635	2715	2796	2876	2956	3037	3117	
3358	3438	3518	3598	3679	3759	3839	3919	
4160	4240	4320	4400	4480	4560	4640	4720	80
4960	5040	5120	5200	5279	5359	5439	5519	
5759	5838	5918	5998	6078	6157	6237	6317	

PROPORTIONAL PARTS.

	3	4	5	6	7	8	9
.4	26.1	34.8	43.5	52.2	60.9	69.6	78.3
.2	25.8	34.4	43.0	51.6	60.2	68.8	77.4
0	25.5	34.0	42.5	51.0	59.5	68.0	76.5
	25.2	33.6	42.0	50.4	58.8	67.2	75.6

0	1	2	3	4	5	6	7	8	9	Diff.
736397	6476	6556	6635	6715	6795	6874	6954	7034	7113	
7193	7272	7352	7431	7511	7590	7670	7749	7829	7908	
7987	8067	8146	8225	8305	8384	8463	8543	8622	8701	
8781	8860	8939	9018	9097	9177	9256	9335	9414	9493	
9572	9651	9731	9810	9889	9968	0047	0126	0205	0284	79
740363	0442	0521	0600	0678	0757	0836	0915	0994	1073	
1152	1230	1309	1388	1467	1546	1624	1703	1782	1860	
1939	2018	2096	2175	2254	2332	2411	2489	2568	2647	
2725	2804	2882	2961	3039	3118	3196	3275	3353	3431	
3510	3588	3667	3745	3823	3902	3980	4058	4136	4215	
4293	4371	4449	4528	4606	4684	4762	4840	4919	4997	
5075	5153	5231	5309	5387	5465	5543	5621	5699	5777	78
5855	5933	6011	6089	6167	6245	6323	6401	6479	6556	
6634	6712	6790	6868	6945	7023	7101	7179	7256	7334	
7412	7489	7567	7645	7722	7800	7878	7955	8033	8110	
8188	8266	8343	8421	8498	8576	8653	8731	8808	8885	
8963	9040	9118	9195	9272	9350	9427	9504	9582	9659	
9736	9814	9891	9968	0045	0123	0200	0277	0354	0431	
750508	0586	0663	0740	0817	0894	0971	1048	1125	1202	
1279	1356	1433	1510	1587	1664	1741	1818	1895	1972	77
2048	2125	2202	2279	2356	2433	2509	2586	2663	2740	
2816	2893	2970	3047	3123	3200	3277	3353	3430	3506	
3583	3660	3736	3813	3889	3966	4042	4119	4195	4272	
4348	4425	4501	4578	4654	4730	4807	4883	4960	5036	
5112	5189	5265	5341	5417	5494	5570	5646	5722	5799	
5875	5951	6027	6103	6180	6256	6332	6408	6484	6560	
6636	6712	6788	6864	6940	7016	7092	7168	7244	7320	76
7396	7472	7548	7624	7700	7775	7851	7927	8003	8079	
8155	8230	8306	8382	8458	8533	8609	8685	8761	8836	
8912	8988	9063	9139	9214	9290	9366	9441	9517	9592	
9668	9743	9819	9894	9970	0045	0121	0196	0272	0347	
760422	0498	0573	0649	0724	0799	0875	0950	1025	1101	
1176	1251	1326	1402	1477	1552	1627	1702	1778	1853	
1928	2003	2078	2153	2228	2303	2378	2453	2529	2604	75
2679	2754	2829	2904	2978	3053	3128	3203	3278	3353	
3428	3503	3578	3653	3727	3802	3877	3952	4027	4101	
4176	4251	4326	4400	4475	4550	4624	4699	4774	4848	
4923	4998	5072	5147	5221	5296	5370	5445	5520	5594	
5669	5743	5818	5892	5966	6041	6115	6190	6264	6338	
6413	6487	6562	6636	6710	6785	6859	6933	7007	7082	

PROPORTIONAL PARTS.

ff.	1	2	3	4	5	6	7	8	9
3	8.3	16.6	24.9	33.2	41.5	49.8	58.1	66.4	74.7
2	8.2	16.4	24.6	32.8	41.0	49.2	57.4	65.6	73.8
1	8.1	16.2	24.3	32.4	40.5	48.6	56.7	64.8	72.9
0	8.0	16.0	24.0	32.0	40.0	48.0	56.0	64.0	72.0
9	7.9	15.8	23.7	31.6	39.5	47.4	55.3	63.2	71.1
8	7.8	15.6	23.4	31.2	39.0	46.8	54.6	62.4	70.2
[illegible]	7.7	15.4	23.1	30.8	38.5	46.2	53.9	61.6	[illegible]
[illegible]	[illegible]	[illegible]	22.8	30.4	38.0	45.6	53.2	60.8	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	30.0	37.5	45.0	52.5	60.0	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	29.6	37.0	44.4	51.8	59.2	[illegible]

2	3	4	5	6	7	8	9	Diff.
7304	7379	7453	7527	7601	7675	7749	7823	
8046	8120	8194	8268	8342	8416	8490	8564	74
8786	8860	8934	9008	9082	9156	9230	9303	
9525	9599	9673	9746	9820	9894	9968	0042	
0263	0336	0410	0484	0557	0631	0705	0778	
0999	1073	1146	1220	1293	1367	1440	1514	
1734	1808	1881	1955	2028	2102	2175	2248	
2468	2542	2615	2688	2762	2835	2908	2981	
3201	3274	3348	3421	3494	3567	3640	3713	
3933	4006	4079	4152	4225	4298	4371	4444	73
4663	4736	4809	4882	4955	5028	5100	5173	
5392	5465	5538	5610	5683	5756	5829	5902	
6120	6193	6265	6338	6411	6483	6556	6629	
6846	6919	6992	7064	7137	7209	7282	7354	
7572	7644	7717	7789	7862	7934	8006	8079	
8296	8368	8441	8513	8585	8658	8730	8802	
9019	9091	9163	9236	9308	9380	9452	9524	
9741	9813	9885	9957	0029	0101	0173	9245	72
0461	0533	0605	0677	0749	0821	0893	0965	
1181	1253	1324	1396	1468	1540	1612	1684	
1899	1971	2042	2114	2186	2258	2329	2401	
2616	2688	2759	2831	2902	2974	3046	3117	
3332	3403	3475	3546	3618	3689	3761	3832	
4046	4118	4189	4261	4332	4403	4475	4546	
4760	4831	4902	4974	5045	5116	5187	5259	
5472	5543	5615	5686	5757	5828	5899	5970	
6183	6254	6325	6396	6467	6538	6609	6680	71
6893	6964	7035	7106	7177	7248	7319	7390	
7602	7673	7744	7815	7885	7956	8027	8098	
8310	8381	8451	8522	8593	8663	8734	8804	
9016	9087	9157	9228	9299	9369	9440	9510	
9722	9792	9863	9933	0004	0074	0144	0215	
0426	0496	0567	0637	0707	0778	0848	0918	
1129	1199	1269	1340	1410	1480	1550	1620	
1831	1901	1971	2041	2111	2181	2252	2322	
2532	2602	2672	2742	2812	2882	2952	3022	70
3231	3301	3371	3441	3511	3581	3651	3721	
3930	4000	4070	4139	4209	4279	4349	4418	
4627	4697	4767	4836	4906	4976	5045	5115	
5324	5393	5463	5532	5602	5672	5741	5811	
6019	6088	6158	6227	6297	6366	6436	6505	
6713	6782	6852	6921	6990	7060	7129	7198	
7406	7475	7545	7614	7683	7752	7821	7890	
8098	8167	8236	8305	8374	8443	8513	8582	
8789	8858	8927	8996	9065	9134	9203	9272	69

PROPORTIONAL PARTS.

	3	4	5	6	7	8	9
0	22.5	30.0	37.5	45.0	52.5	60.0	67.5
2	22.2	29.6	37.0	44.4	51.8	59.2	66.6
	21.9	29.2	36.5	43.8	51.1	58.4	65.7
	21.6	28.8	36.0	43.2	50.4	57.6	64.8
	21.3	28.4	35.5	42.6	49.7	56.8	63.9
	21.0	28.0	35.0	42.0	49.0	56.0	63.0
	20.7	27.6	34.5	41.4	48.3	55.2	62.1

No. 630 L. 799.] [No. 674 L.

N.	0	1	2	3	4	5	6	7	8	9
630	799341	9409	9478	9547	9616	9685	9754	9823	9892	9961
1	800029	0098	0167	0236	0305	0373	0442	0511	0580	0648
2	0717	0786	0854	0923	0992	1061	1129	1198	1266	1335
3	1404	1472	1541	1609	1678	1747	1815	1884	1952	2021
4	2089	2158	2226	2295	2363	2432	2500	2568	2637	2705
5	2774	2842	2910	2979	3047	3116	3184	3252	3321	3389
6	3457	3525	3594	3662	3730	3798	3867	3935	4003	4071
7	4139	4208	4276	4344	4412	4480	4548	4616	4685	4753
8	4821	4889	4957	5025	5093	5161	5229	5297	5365	5433
9	5501	5569	5637	5705	5773	5841	5908	5976	6044	6112
640	806180	6248	6316	6384	6451	6519	6587	6655	6723	6790
1	6858	6926	6994	7061	7129	7197	7264	7332	7400	7467
2	7535	7603	7670	7738	7806	7873	7941	8008	8076	8143
3	8211	8279	8346	8414	8481	8549	8616	8684	8751	8818
4	8886	8953	9021	9088	9156	9223	9290	9358	9425	9492
5	9560	9627	9694	9762	9829	9896	9964	0031	0098	0165
6	810233	0300	0367	0434	0501	0569	0636	0703	0770	0837
7	0904	0971	1039	1106	1173	1240	1307	1374	1441	1508
8	1575	1642	1709	1776	1843	1910	1977	2044	2111	2178
9	2245	2312	2379	2445	2512	2579	2646	2713	2780	2847
650	2913	2980	3047	3114	3181	3247	3314	3381	3448	3514
1	3581	3648	3714	3781	3848	3914	3981	4048	4114	4181
2	4248	4314	4381	4447	4514	4581	4647	4714	4780	4847
3	4913	4980	5046	5113	5179	5246	5312	5378	5445	5511
4	5578	5644	5711	5777	5843	5910	5976	6042	6109	6175
5	6241	6308	6374	6440	6506	6573	6639	6705	6771	6838
6	6904	6970	7036	7102	7169	7235	7301	7367	7433	7499
7	7565	7631	7698	7764	7830	7896	7962	8028	8094	8160
8	8226	8292	8358	8424	8490	8556	8622	8688	8754	8820
9	8885	8951	9017	9083	9149	9215	9281	9346	9412	9478
660	9544	9610	9676	9741	9807	9873	9939	0004	0070	0136
1	820201	0267	0333	0399	0464	0530	0595	0661	0727	0792
2	0858	0924	0989	1055	1120	1186	1251	1317	1382	1448
3	1514	1579	1645	1710	1775	1841	1906	1972	2037	2103
4	2168	2233	2299	2364	2430	2495	2560	2626	2691	2756
5	2822	2887	2952	3018	3083	3148	3213	3279	3344	3409
6	3474	3539	3605	3670	3735	3800	3865	3930	3996	4061
7	4126	4191	4256	4321	4386	4451	4516	4581	4646	4711
8	4776	4841	4906	4971	5036	5101	5166	5231	5296	5361
9	5426	5491	5556	5621	5686	5751	5815	5880	5945	6010
670	6075	6140	6204	6269	6334	6399	6464	6528	6593	6658
1	6723	6787	6852	6917	6981	7046	7111	7175	7240	7305
2	7369	7434	7499	7563	7628	7692	7757	7821	7886	7951
3	8015	8080	8144	8209	8273	8338	8402	8467	8531	8595
4	8660	8724	8789	8853	8918	8982	9046	9111	9175	9239

PROPORTIONAL PARTS.

Diff.	1	2	3	4	5	6	7	8
68	6.8	13.6	20.4	27.2	34.0	40.8	47.6	54.4
67	[illegible]	[illegible]	20.1	26.8	33.5	40.2	46.9	53.6
[illegible]	[illegible]	[illegible]	19.8	26.4	33.0	39.6	46.2	52.8
[illegible]	[illegible]	[illegible]	19.5	26.0	32.5	39.0	45.5	52.0
[illegible]	[illegible]	[illegible]	19.2	25.6	32.0	38.4	44.8	51.2

[No. 719 L. 857.

		4	5	6	7	8	9	Diff.
	7	9561	9625	9690	9754	9818	9882	
	9	0204	0268	0332	0396	0460	0525	
	1	0845	0909	0973	1037	1102	1166	
	2	1486	1550	1614	1678	1742	1806	64
	2	2126	2189	2253	2317	2381	2445	
	0	2764	2828	2892	2956	3020	3083	
	8	3402	3466	3530	3593	3657	3721	
	5	4039	4103	4166	4230	4294	4357	
	1	4675	4739	4802	4866	4929	4993	
	47	5310	5373	5437	5500	5564	5627	
	51	5944	6007	6071	6134	6197	6261	
	14	6577	6641	6704	6767	6830	6894	
	46	7210	7273	7336	7399	7462	7525	
	78	7841	7904	7967	8030	8093	8156	
	08	8471	8534	8597	8660	8723	8786	63
	038	9101	9164	9227	9289	9352	9415	
	567	9729	9792	9855	9918	9981	0043	
	294	0357	0420	0482	0545	0608	0671	
	921	0984	1046	1109	1172	1234	1297	
	1547	1610	1672	1735	1797	1860	1922	
	2172	2235	2297	2360	2422	2484	2547	
	2796	2859	2921	2983	3046	3108	3170	
	3420	3482	3544	3606	3669	3731	3793	
	4042	4104	4166	4229	4291	4353	4415	
	4664	4726	4788	4850	4912	4974	5036	
	5284	5346	5408	5470	5532	5594	5656	62
	5904	5966	6028	6090	6151	6213	6275	
	6523	6585	6646	6708	6770	6832	6894	
	7141	7202	7264	7326	7388	7449	7511	
5	7758	7819	7881	7943	8004	8066	8128	
2	8374	8435	8497	8559	8620	8682	8743	
8	8989	9051	9112	9174	9235	9297	9358	
2	9604	9665	9726	9788	9849	9911	9972	
56	0217	0279	0340	0401	0462	0524	0585	
69	0830	0891	0952	1014	1075	1136	1197	
81	1442	1503	1564	1625	1686	1747	1809	
92	2053	2114	2175	2236	2297	2358	2419	61
302	2663	2724	2785	2846	2907	2968	3029	
211	3272	3333	3394	3455	3516	3577	3637	
820	3881	3941	4002	4063	4124	4185	4245	
428	4488	4549	4610	4670	4731	4792	4852	
034	5095	5156	5216	5277	5337	5398	5459	
5640	5701	5761	5822	5882	5943	6003	6064	
6245	6306	6366	6427	6487	6548	6608	6668	
6850	6910	6970	7031	7091	7152	7212	7272	

PROPORTIONAL PARTS.

3	4	5	6	7	8	9
19.5	26.0	32.5	39.0	45.5	52.0	58.5
19.2	25.6	32.0	38.4	44.8	51.2	57.6
18.9	25.2	31.5	37.8	44.1	50.4	56.7
18.6	24.8	31.0	37.2	43.4	49.6	55.8
18.3	24.4	30.5	36.6	42.7	48.8	54.9
18.0	24.0	30.0	36.0	42.0	48.0	54.0

N.	0	1	2	3	4	5	6	7	8	9
720	857332	7393	7453	7513	7574	7634	7694	7755	7815	7875
1	7935	7995	8056	8116	8176	8236	8297	8357	8417	8477
2	8537	8597	8657	8718	8778	8838	8898	8958	9018	9078
3	9138	9198	9258	9318	9379	9439	9499	9559	9619	9679
4	9739	9799	9859	9918	9978	0038	0098	0158	0218	0278
5	860338	0398	0458	0518	0578	0637	0697	0757	0817	0877
6	0937	0996	1056	1116	1176	1236	1295	1355	1415	1475
7	1534	1594	1654	1714	1773	1833	1893	1952	2012	2072
8	2131	2191	2251	2310	2370	2430	2489	2549	2608	2668
9	2728	2787	2847	2906	2966	3025	3085	3144	3204	3263
730	3323	3382	3442	3501	3561	3620	3680	3739	3799	3858
1	3917	3977	4036	4096	4155	4214	4274	4333	4392	4452
2	4511	4570	4630	4689	4748	4808	4867	4926	4985	5045
3	5104	5163	5222	5282	5341	5400	5459	5519	5578	5637
4	5696	5755	5814	5874	5933	5992	6051	6110	6169	6228
5	6287	6346	6405	6465	6524	6583	6642	6701	6760	6819
6	6878	6937	6996	7055	7114	7173	7232	7291	7350	7409
7	7467	7526	7585	7644	7703	7762	7821	7880	7939	7998
8	8056	8115	8174	8233	8292	8350	8409	8468	8527	8586
9	8644	8703	8762	8821	8879	8938	8997	9056	9114	9173
740	9232	9290	9349	9408	9466	9525	9584	9642	9701	9760
1	9818	9877	9935	9994	0053	0111	0170	0228	0287	0345
2	870404	0462	0521	0579	0638	0696	0755	0813	0872	0930
3	0989	1047	1106	1164	1223	1281	1339	1398	1456	1515
4	1573	1631	1690	1748	1806	1865	1923	1981	2040	2098
5	2156	2215	2273	2331	2389	2448	2506	2564	2622	2681
6	2739	2797	2855	2913	2972	3030	3088	3146	3204	3262
7	3321	3379	3437	3495	3553	3611	3669	3727	3785	3844
8	3902	3960	4018	4076	4134	4192	4250	4308	4366	4424
9	4482	4540	4598	4656	4714	4772	4830	4888	4945	5003
750	5061	5119	5177	5235	5293	5351	5409	5466	5524	5582
1	5640	5698	5756	5813	5871	5929	5987	6045	6102	6160
2	6218	6276	6333	6391	6449	6507	6564	6622	6680	6737
3	6795	6853	6910	6968	7026	7083	7141	7199	7256	7314
4	7371	7429	7487	7544	7602	7659	7717	7774	7832	7889
5	7947	8004	8062	8119	8177	8234	8292	8349	8407	8464
6	8522	8579	8637	8694	8752	8809	8866	8924	8981	9039
7	9096	9153	9211	9268	9325	9383	9440	9497	9555	9612
8	9669	9726	9784	9841	9898	9956	0013	0070	0127	0185
9	880242	0299	0356	0413	0471	0528	0585	0642	0699	0756
760	0814	0871	0928	0985	1042	1099	1156	1213	1271	1328
1	1385	1442	1499	1556	1613	1670	1727	1784	1841	1898
2	1955	2012	2069	2126	2183	2240	2297	2354	2411	2468
3	2525	2581	2638	2695	2752	2809	2866	2923	2980	3037
4	3093	3150	3207	3264	3321	3377	3434	3491	3548	3605

PROPORTIONAL PARTS.

Diff.	1	2	3	4	5	6	7	8
[illegible]	[illegible]	[illegible]	17.7	23.6	29.5	35.4	41.3	47.2
[illegible]	[illegible]	[illegible]	[illegible]	23.2	29.0	34.8	40.6	46.4
[illegible]	[illegible]	[illegible]	[illegible]	22.8	28.5	34.2	39.9	45.6
[illegible]	[illegible]	[illegible]	[illegible]	22.4	28.0	33.6	39.2	44.8

o. 765 L. 883.] [No. 809 L. 908.

N.	0	1	2	3	4	5	6	7	8	9	Diff.
5	883661	3718	3775	3832	3888	3945	4002	4059	4115	4172	
6	4229	4285	4342	4399	4455	4512	4569	4625	4682	4739	
7	4795	4852	4909	4965	5022	5078	5135	5192	5248	5305	
8	5361	5418	5474	5531	5587	5644	5700	5757	5813	5870	
9	5926	5983	6039	6096	6152	6209	6265	6321	6378	6434	
0	6491	6547	6604	6660	6716	6773	6829	6885	6942	6998	
1	7054	7111	7167	7223	7280	7336	7392	7449	7505	7561	
2	7617	7674	7730	7786	7842	7898	7955	8011	8067	8123	
3	8179	8236	8292	8348	8404	8460	8516	8573	8629	8685	
4	8741	8797	8853	8909	8965	9021	9077	9134	9190	9246	
5	9302	9358	9414	9470	9526	9582	9638	9694	9750	9806	56
6	9862	9918	9974	0030	0086	0141	0197	0253	0309	0365	
7	890421	0477	0533	0589	0645	0700	0756	0812	0868	0924	
8	0980	1035	1091	1147	1203	1259	1314	1370	1426	1482	
9	1537	1593	1649	1705	1760	1816	1872	1928	1983	2039	
0	2095	2150	2206	2262	2317	2373	2429	2484	2540	2595	
1	2651	2707	2762	2818	2873	2929	2985	3040	3096	3151	
2	3207	3262	3318	3373	3429	3484	3540	3595	3651	3706	
3	3762	3817	3873	3928	3984	4039	4094	4150	4205	4261	
4	4316	4371	4427	4482	4538	4593	4648	4704	4759	4814	
5	4870	4925	4980	5036	5091	5146	5201	5257	5312	5367	
6	5423	5478	5533	5588	5644	5699	5754	5809	5864	5920	
7	5975	6030	6085	6140	6195	6251	6306	6361	6416	6471	
8	6526	6581	6636	6692	6747	6802	6857	6912	6967	7022	
9	7077	7132	7187	7242	7297	7352	7407	7462	7517	7572	55
0	7627	7682	7737	7792	7847	7902	7957	8012	8067	8122	
1	8176	8231	8286	8341	8396	8451	8506	8561	8615	8670	
2	8725	8780	8835	8890	8944	8999	9054	9109	9164	9218	
3	9273	9328	9383	9437	9492	9547	9602	9656	9711	9766	
4	9821	9875	9930	9985	0039	0094	0149	0203	0258	0312	
5	900367	0422	0476	0531	0586	0640	0695	0749	0804	0859	
6	0913	0968	1022	1077	1131	1186	1240	1295	1349	1404	
7	1458	1513	1567	1622	1676	1731	1785	1840	1894	1948	
8	2003	2057	2112	2166	2221	2275	2329	2384	2438	2492	
9	2547	2601	2655	2710	2764	2818	2873	2927	2981	3036	
00	3090	3144	3199	3253	3307	3361	3416	3470	3524	3578	
1	3633	3687	3741	3795	3849	3904	3958	4012	4066	4120	
2	4174	4229	4283	4337	4391	4445	4499	4553	4607	4661	
3	4716	4770	4824	4878	4932	4986	5040	5094	5148	5202	54
4	5256	5310	5364	5418	5472	5526	5580	5634	5688	5742	
5	5796	5850	5904	5958	6012	6066	6119	6173	6227	6281	
6	6335	6389	6443	6497	6551	6604	6658	6712	6766	6820	
7	6874	6927	6981	7035	7089	7143	7196	7250	7304	7358	
8	7411	7465	7519	7573	7626	7680	7734	7787	7841	7895	
9	7949	8002	8056	8110	8163	8217	8270	8324	8378	8431	

PROPORTIONAL PARTS.

Diff.	1	2	3	4	5	6	7	8	9
57	5.7	11.4	17.1	22.8	28.5	34.2	39.9	45.6	51.3
56	5.6	11.2	16.8	22.4	28.0	33.6	39.2	44.8	50.4
5	5.5	11.0	16.5	22.0	27.5	33.0	38.5	44.0	49.
5	5.4	10.8	16.2	21.6	27.0	32.4	37.8	43.2	48

No. 810 L. 908.] [No. 854 L.

N.	0	1	2	3	4	5	6	7	8	9	D
810	908485	8539	8592	8646	8699	8753	8807	8860	8914	8967	
1	9021	9074	9128	9181	9235	9289	9342	9396	9449	9503	
2	9556	9610	9663	9716	9770	9823	9877	9930	9984	0037	
3	910091	0144	0197	0251	0304	0358	0411	0464	0518	0571	
4	0624	0678	0731	0784	0838	0891	0944	0998	1051	1104	
5	1158	1211	1264	1317	1371	1424	1477	1530	1584	1637	
6	1690	1743	1797	1850	1903	1956	2009	2063	2116	2169	
7	2222	2275	2328	2381	2435	2488	2541	2594	2647	2700	
8	2753	2806	2859	2913	2966	3019	3072	3125	3178	3231	
9	3284	3337	3390	3443	3496	3549	3602	3655	3708	3761	
820	3814	3867	3920	3973	4026	4079	4132	4184	4237	4290	
1	4343	4396	4449	4502	4555	4608	4660	4713	4766	4819	
2	4872	4925	4977	5030	5083	5136	5189	5241	5294	5347	
3	5400	5453	5505	5558	5611	5664	5716	5769	5822	5875	
4	5927	5980	6033	6085	6138	6191	6243	6296	6349	6401	
5	6454	6507	6559	6612	6664	6717	6770	6822	6875	6927	
6	6980	7033	7085	7138	7190	7243	7295	7348	7400	7453	
7	7506	7558	7611	7663	7716	7768	7820	7873	7925	7978	
8	8030	8083	8135	8188	8240	8293	8345	8397	8450	8502	
9	8555	8607	8659	8712	8764	8816	8869	8921	8973	9026	
830	9078	9130	9183	9235	9287	9340	9392	9444	9496	9549	
1	9601	9653	9706	9758	9810	9862	9914	9967	0019	0071	
2	920123	0176	0228	0280	0332	0384	0436	0489	0541	0593	
3	0645	0697	0749	0801	0853	0906	0958	1010	1062	1114	
4	1166	1218	1270	1322	1374	1426	1478	1530	1582	1634	
5	1686	1738	1790	1842	1894	1946	1998	2050	2102	2154	
6	2206	2258	2310	2362	2414	2466	2518	2570	2622	2674	
7	2725	2777	2829	2881	2933	2985	3037	3089	3140	3192	
8	3244	3296	3348	3399	3451	3503	3555	3607	3658	3710	
9	3762	3814	3865	3917	3969	4021	4072	4124	4176	4228	
840	4279	4331	4383	4434	4486	4538	4589	4641	4693	4744	
1	4796	4848	4899	4951	5003	5054	5106	5157	5209	5261	
2	5312	5364	5415	5467	5518	5570	5621	5673	5725	5776	
3	5828	5879	5931	5982	6034	6085	6137	6188	6240	6291	
4	6342	6394	6445	6497	6548	6600	6651	6702	6754	6805	
5	6857	6908	6959	7011	7062	7114	7165	7216	7268	7319	
6	7370	7422	7473	7524	7576	7627	7678	7730	7781	7832	
7	7883	7935	7986	8037	8088	8140	8191	8242	8293	8345	
8	8396	8447	8498	8549	8601	8652	8703	8754	8805	8857	
9	8908	8959	9010	9061	9112	9163	9215	9266	9317	9368	
850	9419	9470	9521	9572	9623	9674	9725	9776	9827	9879	
1	9930	9981	0032	0083	0134	0185	0236	0287	0338	0389	
2	930440	0491	0542	0592	0643	0694	0745	0796	0847	0898	
3	0949	1000	1051	1102	1153	1204	1254	1305	1356	1407	
4	1458	1509	1560	1610	1661	1712	1763	1814	1865	1915	

PROPORTIONAL PARTS.

Diff.	1	2	3	4	5	6	7	8
53	5.3	10.6	15.9	21.2	26.5	31.8	37.1	42.4
52	5.2	10.4	15.6	20.8	26.0	31.2	36.4	41.6
[illegible]	[illegible]	[illegible]	15.3	20.4	25.5	30.6	35.7	40.8
[illegible]	[illegible]	[illegible]	15.0	20.0	25.0	30.0	35.0	40.0

	1	2	3	4	5	6	7	8	9	
	017	2068	2118	2169	2220	2271	2322	2372	2423	
	524	2575	2626	2677	2727	2778	2829	2879	2930	
	3031	3082	3133	3183	3234	3285	3335	3386	3437	
	3538	3589	3639	3690	3740	3791	3841	3892	3943	
	4044	4094	4145	4195	4246	4296	4347	4397	4448	
	4549	4599	4650	4700	4751	4801	4852	4902	4953	
	5054	5104	5154	5205	5255	5306	5356	5406	5457	
	5558	5608	5658	5709	5759	5809	5860	5910	5960	
1	6061	6111	6162	6212	6262	6313	6363	6413	6463	
4	6564	6614	6665	6715	6765	6815	6865	6916	6966	
16	7066	7116	7167	7217	7267	7317	7367	7418	7468	
18	7568	7618	7668	7718	7769	7819	7869	7919	7969	
19	8069	8119	8169	8219	8269	8320	8370	8420	8470	5
520	8570	8620	8670	8720	8770	8820	8870	8920	8970	
020	9070	9120	9170	9220	9270	9320	9369	9419	9469	
9519	9569	9619	9669	9719	9769	9819	9869	9918	9968	
0018	0068	0118	0168	0218	0267	0317	0367	0417	0467	
0516	0566	0616	0666	0716	0765	0815	0865	0915	0964	
1014	1064	1114	1163	1213	1263	1313	1362	1412	1462	
1511	1561	1611	1660	1710	1760	1809	1859	1909	1958	
2008	2058	2107	2157	2207	2256	2306	2355	2405	2455	
2504	2554	2603	2653	2702	2752	2801	2851	2901	2950	
000	3049	3099	3148	3198	3247	3297	3346	3396	3445	
495	3544	3593	3643	3692	3742	3791	3841	3890	3939	
989	4038	4088	4137	4186	4236	4285	4335	4384	4433	
83	4532	4581	4631	4680	4729	4779	4828	4877	4927	
76	5025	5074	5124	5173	5222	5272	5321	5370	5419	
9	5518	5567	5616	5665	5715	5764	5813	5862	5912	
	6010	6059	6108	6157	6207	6256	6305	6354	6403	
	6501	6551	6600	6649	6698	6747	6796	6845	6894	
	6992	7041	7090	7139	7189	7238	7287	7336	7385	
	7483	7532	7581	7630	7679	7728	7777	7826	7875	49
	7973	8022	8070	8119	8168	8217	8266	8315	8364	
	8462	8511	8560	8608	8657	8706	8755	8804	8853	
	8951	8999	9048	9097	9146	9195	9244	9292	9341	
	9439	9488	9536	9585	9634	9683	9731	9780	9829	
	9926	9975	0024	0073	0121	0170	0219	0267	0316	
	14	0462	0511	0560	0608	0657	0706	0754	0803	
	00	0949	0997	1046	1095	1143	1192	1240	1289	
	86	1435	1483	1532	1580	1629	1677	1726	1775	
	2	1920	1969	2017	2066	2114	2163	2211	2260	
	6	2405	2453	2502	2550	2599	2647	2696	2744	
	1	2889	2938	2986	3034	3083	3131	3180	3228	
	5	3373	3421	3470	3518	3566	3615	3663	3711	
	3	3856	3905	3953	4001	4049	4098	4146	4194	

PROPORTIONAL PARTS.

3	4	5	6	7	8	9
15.3	20.4	25.5	30.6	35.7	40.8	45.9
15.0	20.0	25.0	30.0	35.0	40.0	
14.7	19.6	24.5	29.4	[illegible]		
14.4	19.2	24.0	[illegible]			

N.	0	1	2	3	4	5	6	7	8	9	D
900	954243	4291	4339	4387	4435	4484	4532	4580	4628	4677	
1	4725	4773	4821	4869	4918	4966	5014	5062	5110	5158	
2	5207	5255	5303	5351	5399	5447	5495	5543	5592	5640	
3	5688	5736	5784	5832	5880	5928	5976	6024	6072	6120	
4	6168	6216	6265	6313	6361	6409	6457	6505	6553	6601	
5	6649	6697	6745	6793	6840	6888	6936	6984	7032	7080	
6	7128	7176	7224	7272	7320	7368	7416	7464	7512	7559	
7	7607	7655	7703	7751	7799	7847	7894	7942	7990	8038	
8	8086	8134	8181	8229	8277	8325	8373	8421	8468	8516	
9	8564	8612	8659	8707	8755	8803	8850	8898	8946	8994	
910	9041	9089	9137	9185	9232	9280	9328	9375	9423	9471	
1	9518	9566	9614	9661	9709	9757	9804	9852	9900	9947	
2	9995	0042	0090	0138	0185	0233	0280	0328	0376	0423	
3	960471	0518	0566	0613	0661	0709	0756	0804	0851	0899	
4	0946	0994	1041	1089	1136	1184	1231	1279	1326	1374	
5	1421	1469	1516	1563	1611	1658	1706	1753	1801	1848	
6	1895	1943	1990	2038	2085	2132	2180	2227	2275	2322	
7	2369	2417	2464	2511	2559	2606	2653	2701	2748	2795	
8	2843	2890	2937	2985	3032	3079	3126	3174	3221	3268	
9	3316	3363	3410	3457	3504	3552	3599	3646	3693	3741	
920	3788	3835	3882	3929	3977	4024	4071	4118	4165	4212	
1	4260	4307	4354	4401	4448	4495	4542	4590	4637	4684	
2	4731	4778	4825	4872	4919	4966	5013	5061	5108	5155	
3	5202	5249	5296	5343	5390	5437	5484	5531	5578	5625	
4	5672	5719	5766	5813	5860	5907	5954	6001	6048	6095	
5	6142	6189	6236	6283	6329	6376	6423	6470	6517	6564	
6	6611	6658	6705	6752	6799	6845	6892	6939	6986	7033	
7	7080	7127	7173	7220	7267	7314	7361	7408	7454	7501	
8	7548	7595	7642	7688	7735	7782	7829	7875	7922	7969	
9	8016	8062	8109	8156	8203	8249	8296	8343	8390	8436	
930	8483	8530	8576	8623	8670	8716	8763	8810	8856	8903	
1	8950	8996	9043	9090	9136	9183	9229	9276	9323	9369	
2	9416	9463	9509	9556	9602	9649	9695	9742	9789	9835	
3	9882	9928	9975	0021	0068	0114	0161	0207	0254	0300	
4	970347	0393	0440	0486	0533	0579	0626	0672	0719	0765	
5	0812	0858	0904	0951	0997	1044	1090	1137	1183	1229	
6	1276	1322	1369	1415	1461	1508	1554	1601	1647	1693	
7	1740	1786	1832	1879	1925	1971	2018	2064	2110	2157	
8	2203	2249	2295	2342	2388	2434	2481	2527	2573	2619	
9	2666	2712	2758	2804	2851	2897	2943	2989	3035	3082	
940	3128	3174	3220	3266	3313	3359	3405	3451	3497	3543	
1	3590	3636	3682	3728	3774	3820	3866	3913	3959	4005	
2	4051	4097	4143	4189	4235	4281	4327	4374	4420	4466	
3	4512	4558	4604	4650	4696	4742	4788	4834	4880	4926	
4	4972	5018	5064	5110	5156	5202	5248	5294	5340	5386	

PROPORTIONAL PARTS.

Diff.	1	2	3	4	5	6	7	8
47	4.7	9.4	14.1	18.8	23.5	28.2	32.9	37.6
46	4.6	[illegible]	[illegible]	18.4	23.0	27.6	32.2	36.8

1	2	3	4	5	6	7	8	9	Diff.
5478	5524	5570	5616	5662	5707	5753	5799	5845	
5937	5983	6029	6075	6121	6167	6212	6258	6304	
6396	6442	6488	6533	6579	6625	6671	6717	6763	
6854	6900	6946	6992	7037	7083	7129	7175	7220	
7312	7358	7403	7449	7495	7541	7586	7632	7678	
7769	7815	7861	7906	7952	7998	8043	8089	8135	
8226	8272	8317	8363	8409	8454	8500	8546	8591	
8683	8728	8774	8819	8865	8911	8956	9002	9047	
9138	9184	9230	9275	9321	9366	9412	9457	9503	
9594	9639	9685	9730	9776	9821	9867	9912	9958	
0049	0094	0140	0185	0231	0276	0322	0367	0412	
0503	0549	0594	0640	0685	0730	0776	0821	0867	
0957	1003	1048	1093	1139	1184	1229	1275	1320	
1411	1456	1501	1547	1592	1637	1683	1728	1773	
1864	1909	1954	2000	2045	2090	2135	2181	2226	
2316	2362	2407	2452	2497	2543	2588	2633	2678	
2769	2814	2859	2904	2949	2994	3040	3085	3130	
3220	3265	3310	3356	3401	3446	3491	3536	3581	
3671	3716	3762	3807	3852	3897	3942	3987	4032	
4122	4167	4212	4257	4302	4347	4392	4437	4482	
4572	4617	4662	4707	4752	4797	4842	4887	4932	45
5022	5067	5112	5157	5202	5247	5292	5337	5382	
5471	5516	5561	5606	5651	5696	5741	5786	5830	
5920	5965	6010	6055	6100	6144	6189	6234	6279	
6369	6413	6458	6503	6548	6593	6637	6682	6727	
6817	6861	6906	6951	6996	7040	7085	7130	7175	
7264	7309	7353	7398	7443	7488	7532	7577	7622	
7711	7756	7800	7845	7890	7934	7979	8024	8068	
8157	8202	8247	8291	8336	8381	8425	8470	8514	
8604	8648	8693	8737	8782	8826	8871	8916	8960	
9049	9094	9138	9183	9227	9272	9316	9361	9405	
9494	9539	9583	9628	9672	9717	9761	9806	9850	
9939	9983	0028	0072	0117	0161	0206	0250	0294	
0383	0428	0472	0516	0561	0605	0650	0694	0738	
0827	0871	0916	0960	1004	1049	1093	1137	1182	
1270	1315	1359	1403	1448	1492	1536	1580	1625	
1713	1758	1802	1846	1890	1935	1979	2023	2067	
2156	2200	2244	2288	2333	2377	2421	2465	2509	
2598	2642	2686	2730	2774	2819	2863	2907	2951	
3039	3083	3127	3172	3216	3260	3304	3348	3392	
3480	3524	3568	3613	3657	3701	3745	3789	3833	
3921	3965	4009	4053	4097	4141	4185	4229	4273	
4361	4405	4449	4493	4537	4581	4625	4669	4713	44
4801	4845	4889	4933	4977	5021	5065	5108	5152	
5240	5284	5328	5372	5416	5460	5504	5547	5591	

PROPORTIONAL PARTS.

2	3	4	5	6	7	8	9
9.2	*13.8*	*18.4*	*23.0*	*27.6*	32.2	36.8	41.4
9.0	*13.5*	*18.0*	*22.5*	*27.0*	31.5	36.0	40.5
8.8	*13.2*	*17.6*	*22.0*	*26.4*	30.8	35.2	[illegible]
8.6	*12.9*	*17.2*	*21.5*	*25.8*	30.1	34.4	[illegible]

N.	0	1	2	3	4	5	6	7	8	9
990	995635	5679	5723	5767	5811	5854	5898	5942	5986	6030
1	6074	6117	6161	6205	6249	6293	6337	6380	6424	6468
2	6512	6555	6599	6643	6687	6731	6774	6818	6862	6906
3	6949	6993	7037	7080	7124	7168	7212	7255	7299	7343
4	7386	7430	7474	7517	7561	7605	7648	7692	7736	7779
5	7823	7867	7910	7954	7998	8041	8085	8129	8172	8216
6	8259	8303	8347	8390	8434	8477	8521	8564	8608	8652
7	8695	8739	8782	8826	8869	8913	8956	9000	9043	9087
8	9131	9174	9218	9261	9305	9348	9392	9435	9479	9522
9	9565	9609	9652	9696	9739	9783	9826	9870	9913	9957

HYPERBOLIC LOGARITHMS.

No.	Log.	No.	Log.	No.	Log.	No.	Log.	No.	L
1.01	.0099	1.45	.3716	1.89	.6366	2.33	.8458	2.77	[illegible]
1.02	.0198	1.46	.3784	1.90	.6419	2.34	.8502	2.78	[illegible]
1.03	.0296	1.47	.3853	1.91	.6471	2.35	.8544	2.79	[illegible]
1.04	.0392	1.48	.3920	1.92	.6523	2.36	.8587	2.80	[illegible]
1.05	.0488	1.49	.3988	1.93	.6575	2.37	.8629	2.81	[illegible]
1.06	.0583	1.50	.4055	1.94	.6627	2.38	.8671	2.82	[illegible]
1.07	.0677	1.51	.4121	1.95	.6678	2.39	.8713	2.83	[illegible]
1.08	.0770	1.52	.4187	1.96	.6729	2.40	.8755	2.84	[illegible]
1.09	.0862	1.53	.4253	1.97	.6780	2.41	.8796	2.85	[illegible]
1.10	.0953	1.54	.4318	1.98	.6831	2.42	.8838	2.86	[illegible]
1.11	.1044	1.55	.4383	1.99	.6881	2.43	.8879	2.87	[illegible]
1.12	.1133	1.56	.4447	2.00	.6931	2.44	.8920	2.88	[illegible]
1.13	.1222	1.57	.4511	2.01	.6981	2.45	.8961	2.89	[illegible]
1.14	.1310	1.58	.4574	2.02	.7031	2.46	.9002	2.90	[illegible]
1.15	.1398	1.59	.4637	2.03	.7080	2.47	.9042	2.91	[illegible]
1.16	.1484	1.60	.4700	2.04	.7129	2.48	.9083	2.92	[illegible]
1.17	.1570	1.61	.4762	2.05	.7178	2.49	.9123	2.93	[illegible]
1.18	.1655	1.62	.4824	2.06	.7227	2.50	.9163	2.94	[illegible]
1.19	.1740	1.63	.4886	2.07	.7275	2.51	.9203	2.95	
1.20	.1823	1.64	.4947	2.08	.7324	2.52	.9243	2.96	
1.21	.1906	1.65	.5008	2.09	.7372	2.53	.9282	2.97	
1.22	.1988	1.66	.5068	2.10	.7419	2.54	.9322	2.98	
1.23	.2070	1.67	.5128	2.11	.7467	2.55	.9361	2.99	
1.24	.2151	1.68	.5188	2.12	.7514	2.56	.9400	3.00	
1.25	.2231	1.69	.5247	2.13	.7561	2.57	.9439	3.01	
1.26	.2311	1.70	.5306	2.14	.7608	2.58	.9478	3.02	
1.27	.2390	1.71	.5365	2.15	.7655	2.59	.9517	3.03	
1.28	.2469	1.72	.5423	2.13	.7701	2.60	.9555	3.04	
1.29	.2546	1.73	.5481	2.17	.7747	2.61	.9594	3.05	
1.30	.2624	1.74	.5539	2.18	.7793	2.62	.9632	3.06	
1.31	.2700	1.75	.5596	2.19	.7839	2.63	.9670	3.07	
1.32	.2776	1.76	.5653	2.20	.7885	2.64	.9708	3.08	
1.33	.2852	1.77	.5710	2.21	.7930	2.65	.9746	3.09	
1.34	.2927	1.78	.5766	2.22	.7975	2.66	.9783	3.10	
1.35	.3001	1.79	.5822	2.23	.8020	2.67	.9821	3.11	
1.36	.3075	1.80	.5878	2.24	.8065	2.68	.9858	3.12	
1.37	.3148	1.81	.5933	2.25	.8109	2.69	.9895	3.13	
1.38	.3221	1.82	.5988	2.26	.8154	2.70	.9933	3.14	
1.39	.3293	1.83	.6043	2.27	.8198	2.71	.9969	3.15	
1.40	.3365	1.84	.6098	2.28	.8242	2.72	1.0006	3.16	
1.41	.3436	1.85	.6152	2.29	.8286	2.73	1.0043	3.17	
1.42	.3507	1.86	.6206	2.30	.8329	2.74	1.0080	3.18	
1.43	.3577	1.87	.6259	2.31	.8372	2.75	1.0116	3.19	
1.44	[illegible]	[illegible]	.6313	2.32	.8416	2.76	1.0152	3.20	

No.	Log.	No.	Log.	No.	Log.
4.53	1.5107	5.19	1.6467	5.85	1.76
4.54	1.5129	5.20	1.6487	5.86	1.76
4.55	1.5151	5.21	1.6506	5.87	1.769
4.56	1.5173	5.22	1.6525	5.88	1.771
4.57	1.5195	5.23	1.6514	5.89	1.773
4.58	1.5217	5.24	1.6563	5.90	1.7750
4.59	1.5239	5.25	1.6582	5.91	1.7766
4.60	1.5261	5.26	1.6601	5.92	1.7783
4.61	1.5282	5.27	1.6620	5.93	1.780
.62	1.5304	5.28	1.6639	5.94	1.7817
63	1.5326	5.29	1.6658	5.95	1.7834
64	1.5347	5.30	1.6677	5.96	1.7851
65	1.5369	5.31	1.6696	5.97	1.7867
66	1.5390	5.32	1.6715	5.98	1.7884
67	1.5412	5.33	1.6734	5.99	1.7901
	1.5433	5.34	1.6752	6.00	1.7918
	1.5454	5.35	1.6771	6.01	1.7934
	1.5476	5.36	1.6790	6.02	1.7951
	1.5497	5.37	1.6808	6.03	1.7967
	1.5518	5.38	1.6827	6.04	1.7984
	1.5539	5.39	1.6845	6.05	1.8001
	1.5560	5.40	1.6864	6.06	1.8017
	1.5581	5.41	1.6882	6.07	1.8034
	1.5602	5.42	1.6901	6.08	1.8050
	1.5623	5.43	1.6919	6.09	1.8066
	1.5644	5.44	1.6938	6.10	1.8083
	1.5665	5.45	1.6956	6.11	1.8099
	1.5686	5.46	1.6974	6.12	1.8116
	1.5707	5.47	1.6993	6.13	1.8132
	1.5728	5.48	1.7011	6.14	1.8148
	1.5748	5.49	1.7029	6.15	1.8165
	1.5769	5.50	1.7047	6.16	1.8181
	1.5790	5.51	1.7066	6.17	1.8197
	1.5810	5.52	1.7084	6.18	1.8213
	1.5831	5.53	1.7102	6.19	1.8229
	1.5851	5.54	1.7120	6.20	1.8245
	1.5872	5.55	1.7138	6.21	1.8262
	1.5892	5.56	1.7156	6.22	1.8278
	1.5913	5.57	1.7174	6.23	1.8294
	1.5933	5.58	1.7192	6.24	1.8310
	1.5953	5.59	1.7210	6.25	1.8326
	1.5974	5.60	1.7228	6.26	1.8342
	1.5994	5.61	1.7246	6.27	1.8358
	1.6014	5.62	1.7263	6.28	1.8374
	1.6034	5.63	1.7281	6.29	1.8390
	1.6054	5.64	1.7299	6.30	1.8405
	1.6074	5.65	1.7317	6.31	1.8421
	1.6094	5.66	1.7334	6.32	1.8437
	1.6114	5.67	1.7352	6.33	1.8453
	1.6134	5.68	1.7370	6.34	1.8469
	1.6154	5.69	1.7387	6.35	1.8485
	1.6174	5.70	1.7405	6.36	1.8500
	1.6194	5.71	1.7422	6.37	1.8516
	1.6214	5.72	1.7440	6.38	1.8532
	1.6233	5.73	1.7457	6.39	1.8547
	1.6253	5.74	1.7475	6.40	1.8563
	1.6273	5.75	1.7492	6.41	1.8579
	1.6292	5.76	1.7509	6.42	1.8594
	1.6312	5.77	1.7527	6.43	1.8610
	1.6332	5.78	1.7544	6.44	1.8625
	1.6351	5.79	1.7561	6.45	1.8641
	1.6371	5.80	1.7579	6.46	1.8656
	1.6390	5.81	1.7596	6.47	1.8672
	1.6409	5.82	1.7613	6.48	1.8687
	1.6429	5.83	1.7630	6.49	1.8703
	1.6448	5.84	1.7647	6.50	1.8718

No.	Log.	No.	Log.	No.	Log.	No.	Log.	No.	Log.
6.51	1.8733	7.15	1.9671	7.79	2.0528	8.66	2.1587	9.94	2.2966
6.52	1.8749	7.16	1.9685	7.80	2.0541	8.68	2.1610	9.96	2.2986
6.53	1.8764	7.17	1.9699	7.81	2.0554	8.70	2.1633	9.98	2.3006
6.54	1.8779	7.18	1.9713	7.82	2.0567	8.72	2.1656	10.00	2.3026
6.55	1.8795	7.19	1.9727	7.83	2.0580	8.74	2.1679	10.25	2.3273
6.56	1.8810	7.20	1.9741	7.84	2.0592	8.76	2.1702	10.50	2.3514
6.57	1.8825	7.21	1.9754	7.85	2.0605	8.78	2.1725	10.75	2.3749
6.58	1.8840	7.22	1.9769	7.86	2.0618	8.80	2.1748	11.00	2.3979
6.59	1.8856	7.23	1.9782	7.87	2.0631	8.82	2.1770	11.25	2.4201
6.60	1.8871	7.24	1.9796	7.88	2.0643	8.84	2.1793	11.50	2.4430
6.61	1.8886	7.25	1.9810	7.89	2.0656	8.86	2.1815	11.75	2.4636
6.62	1.8901	7.26	1.9824	7.90	2.0669	8.88	2.1838	12.00	2.4849
6.63	1.8916	7.27	1.9838	7.91	2.0681	8.90	2.1861	12.25	2.5052
6.64	1.8931	7.28	1.9851	7.92	2.0694	8.92	2.1883	12.50	2.5252
6.65	1.8946	7.29	1.9865	7.93	2.0707	8.94	2.1905	12.75	2.5455
6.66	1.8961	7.30	1.9879	7.94	2.0719	8.96	2.1928	13.00	2.5649
6.67	1.8976	7.31	1.9892	7.95	2.0732	8.98	2.1950	13.25	2.5840
6.68	1.8991	7.32	1.9906	7.96	2.0744	9.00	2.1972	13.50	2.6027
6.69	1.9006	7.33	1.9920	7.97	2.0757	9.02	2.1994	13.75	2.6211
6.70	1.9021	7.34	1.9933	7.98	2.0769	9.04	2.2017	14.00	2.6391
6.71	1.9036	7.35	1.9947	7.99	2.0782	9.06	2.2039	14.25	2.6567
6.72	1.9051	7.36	1.9961	8.00	2.0794	9.08	2.2061	14.50	2.6740
6.73	1.9066	7.37	1.9974	8.01	2.0807	9 10	2.2083	14.75	2.6913
6.74	1.9081	7.38	1.9988	8.02	2.0819	9.12	2.2105	15.00	2.7081
6.75	1.9095	7.39	2.0001	8.03	2.0832	9.14	2.2127	15.50	2.7408
6.76	1.9110	7.40	2.0015	8.04	2.0844	9.16	2.2148	16.00	2.7726
6.77	1.9125	7.41	2.0028	8.05	2.0857	9.18	2.2170	16.50	2.8034
6.78	1.9140	7.42	2.0041	8.06	2.0869	9.20	2.2192	17.00	2.8332
6.79	1.9155	7.43	2.0055	8.07	2.0882	9.22	2.2214	17.50	2.8621
6.80	1.9169	7.44	2.0069	8.08	2.0894	9.24	2.2235	18.00	2.8904
6.81	1.9184	7.45	2.0082	8.09	2.0906	9.26	2.2257	18.50	2.9173
6.82	1.9199	7.46	2.0096	8.10	2.0919	9.28	2.2279	19.00	2.9444
6.83	1.9213	7.47	2.0108	8.11	2.0931	9.30	2.2300	19.50	2.9703
6.84	1.9228	7.48	2.0122	8.12	2.0943	9.32	2.2322	20.00	2.9957
6.85	1.9242	7.49	2.0136	8.13	2.0956	9.34	2.2343	21	3.0445
6.86	1.9257	7.50	2.0149	8.14	2.0968	9.36	2.2364	22	3.0910
6.87	1.9272	7.51	2.0162	8.15	2.0980	9.38	2.2386	23	3.1355
6.88	1.9286	7.52	2.0176	8.16	2.0992	9.40	2.2407	24	3.1781
6.89	1.9301	7.53	2.0189	8.17	2.1005	9.42	2.2428	25	3.2189
6.90	1.9315	7.54	2.0202	8.18	2.1017	9.44	2.2450	26	3.2581
6.91	1.9330	7.55	2.0215	8.19	2.1029	9.46	2.2471	27	3.2958
6.92	1.9344	7.56	2.0229	8.20	2.1041	9.48	2.2492	28	3.3322
6.93	1.9359	7.57	2.0242	8.22	2.1066	9.50	2.2513	29	3.3673
6.94	1.9373	7.58	2.0255	8.24	2.1090	9.52	2.2534	30	3.4012
6.95	1.9387	7.59	2.0268	8.26	2.1114	9.54	2.2555	31	3.4340
6.96	1.9402	7.60	2.0281	8.28	2.1138	9.56	2.2576	32	3.4657
6.97	1.9416	7.61	2.0295	8.30	2.1163	9.58	2.2597	33	3.4965
6.98	1.9430	7.62	2.0308	8.32	2.1187	9.60	2.2618	34	3.5263
6.99	1.9445	7.63	2.0321	8.34	2.1211	9.62	2.2638	35	3.5553
7.00	1.9459	7.64	2.0334	8.36	2.1235	9.64	2.2659	36	3.5835
7.01	1.9473	7.65	2.0347	8.38	2.1258	9.66	2.2680	37	3.6109
7.02	1.9488	7.66	2.0360	8.40	2.1282	9.68	2.2701	38	3.6376
7.03	1.9502	7.67	2.0373	8.42	2.1306	9.70	2.2721	39	3.6636
7.04	1.9516	7.68	2.0386	8.44	2.1330	9.72	2.2742	40	3.6889
7.05	1.9530	7.69	2.0399	8.46	2.1353	9.74	2.2762	41	3.7136
7.06	1.9544	7.70	2.0412	8.48	2.1377	9.76	2.2783	42	3.7377
7.07	1.9559	7.71	2.0425	8.50	2.1401	9.78	2.2803	43	3.7612
7.08	1.9573	7.72	2.0438	8.52	2.1424	9.80	2.2824	44	3.7842
7.09	1.9587	7.73	2.0451	8.54	2.1448	9.82	2.2844	45	3.8067
7.10	1.9601	7.74	2.0464	8.56	2.1471	9.84	2.2865	46	3.8286
7.11	1.9615	7.75	2.0477	8.58	2.1494	9.86	2.2885	47	3.8501
7.12	1.9629	7.76	2.0490	8.60	2.1518	9.88	2.2905	48	3.8712
7.13	1.9643	7.77	2.0503	8.62	2.1541	9.90	2.2925	49	3.8918
7.14	1.9657	7.78	2.0516	8.64	2.1564	9.92	2.2946	50	3.9120

ONOMETRICAL FUNCTIONS.

		Tang.	Cotan.	Secant.	Ver. Sin.	Cosine.		
		.00000	Infinite	1.0000	.00000	1.0000	90	0
		.00436	229.18	1.0000	.00001	.99999		45
		.00873	114.59	1.0000	.00004	.99996		30
		.01309	76.390	1.0001	.00009	.99991		15
		.01745	57.290	1.0001	.00015	.99985	89	0
		.02182	45.829	1.0002	.00024	.99976		45
		.02618	38.188	1.0003	.00034	.99966		30
		.03055	32.730	1.0005	.00047	.99953		15
		.03492	28.636	1.0006	.00061	.99939	88	0
		.03929	25.452	1.0008	.00077	.99923		45
		.04366	22.904	1.0009	.00095	.99905		30
		.04803	20.819	1.0011	.00115	.99885		15
		.05241	19.081	1.0014	.00137	.99863	87	0
		.05678	17.611	1.0016	.00161	.99839		45
		.06116	16.350	1.0019	.00187	.99813		30
		.06554	15.257	1.0021	.00214	.99786		15
		.06993	14.301	1.0024	.00244	.99756	86	0
		.07431	13.457	1.0028	.00275	.99725		45
		.07870	12.706	1.0031	.00308	.99692		30
		.08309	12.035	1.0034	.00343	.99656		15
		.08749	11.430	1.0038	.00381	.99619	85	0
		.09189	10.883	1.0042	.00420	.99580		45
		.09629	10.385	1.0046	.00460	.99540		30
	2	.10069	9.9310	1.0051	.00503	.99497		15
	8	.10510	9.5144	1.0055	.00548	.99452	84	0
	5	.10952	9.1309	1.0060	.00594	.99406		45
	7	.11393	8.7769	1.0065	.00643	.99357		30
	9	.11836	8.4400	1.0070	.00693	.99307		15
	55	.12278	8.1443	1.0075	.00745	.99255	83	0
	40	.12722	7.8606	1.0081	.00800	.99200		45
	13	.13165	7.5958	1.0086	.00856	.99144		30
	56	.13609	7.3479	1.0092	.00913	.99086		15
	53	.14054	7.1154	1.0098	.00973	.99027	82	0
	90	.14499	6.8969	1.0105	.01035	.98965		45
	55	.14945	6.6912	1.0111	.01098	.98902		30
	736	.15391	6.4971	1.0118	.01164	.98836		15
	924	.15838	6.3138	1.0125	.01231	.98769	81	0
	211	.16286	6.1402	1.0132	.01300	.98700		45
	0589	.16734	5.9758	1.0139	.01371	.98629		30
	9049	.17183	5.8197	1.0147	.01444	.98556		15
	7588	.17633	5.6713	1.0154	.01519	.98481	80	0
	.6198	.18083	5.5301	1.0162	.01596	.98404		45
	.4874	.18534	5.3955	1.0170	.01675	.98325		30
	.3612	.18986	5.2672	1.0179	.01755	.98245		15
	.2408	.19438	5.1446	1.0187	.01837	.98163	79	0
	.1258	.19891	5.0273	1.0196	.01921	.98079		45
	5.0158	.20345	4.9152	1.0205	.02008	.97992		30
	4.9106	.20800	4.8077	1.0214	.02095	.97905	*	15
	4.8097	.21256	4.7046	1.0223	.02185	.97815	78	0
	4.7130	.21712	4.6057	1.0233	.02277	.97723		45
	4.6202	.22169	4.5107	1.0243	.02370	.97630		30
	4.5311	.22628	4.4194	1.0253	.02466	.97534		15
	4.4454	.23087	4.3315	1.0263	.02563	.97437	77	0
	4.3630	.23547	4.2468	1.0273	.02662	.97338		45
	4.2837	.24008	4.1653	1.0284	.02763	.97237		30
	4.2072	.24470	4.0867	1.0295	.02866	.97134		15
	4.1336	.24933	4.0108	1.0306	.02970	.97030	76	0
5	4.0625	.25397	3.9375	1.0317	.03077	.96923		45
2	3.9939	.25862	3.8667	1.0329	.03185	.96815		30
0	3.9277	.26328	3.7983	1.0341	.03295	.96705		15
18	3.8637	.26795	3.7320	1.0353	.03407	.96593	75	0
In.	Secant.	Cotan.	Tang.	Cosec.	Co-Vers.	Sine.	°	M.

90° read from bottom of table upwards.

°	M.	Sine.	Co-Vers.	Cosec.	Tang.	Cotan.	Secant.	Ver. Sin.	Cosine.	
15	0	.25882	.74118	3.8637	.26795	3.7320	1.0353	.03407	.96593	75
	15	.26303	.73697	3.8018	.27262	3.6680	1.0365	.03521	.96479	
	30	.26724	.73276	3.7420	.27732	3.6059	1.0377	.03637	.96363	
	45	.27144	.72856	3.6840	.28203	3.5457	1.0390	.03754	.96246	
16	0	.27564	.72436	3.6280	.28674	3.4874	1.0403	.03874	.96126	74
	15	.27983	.72017	3.5736	.29147	3.4308	1.0416	.03995	.96005	
	30	.28402	.71598	3.5209	.29621	3.3759	1.0429	.04118	.95882	
	45	.28820	.71180	3.4699	.30096	3.3226	1.0443	.04243	.95757	
17	0	.29237	.70763	3.4203	.30573	3.2709	1.0457	.04370	.95630	73
	15	.29654	.70346	3.3722	.31051	3.2205	1.0471	.04498	.95502	
	30	.30070	.69929	3.3255	.31530	3.1716	1.0485	.04628	.95372	
	45	.30486	.69514	3.2801	.32010	3.1240	1.0500	.04760	.95240	
18	0	.30902	.69098	3.2361	.32492	3.0777	1.0515	.04894	.95106	72
	15	.31316	.68684	3.1932	.32975	3.0326	1.0530	.05030	.94970	
	30	.31730	.68270	3.1515	.33459	2.9887	1.0545	.05168	.94832	
	45	.32144	.67856	3.1110	.33945	2.9459	1.0560	.05307	.94693	
19	0	.32557	.67443	3.0715	.34433	2.9042	1.0576	.05448	.94552	71
	15	.32969	.67031	3.0331	.34921	2.8636	1.0592	.05591	.94409	
	30	.33381	.66619	2.9957	.35412	2.8239	1.0608	.05736	.94264	
	45	.33792	.66208	2.9593	.35904	2.7852	1.0625	.05882	.94118	
20	0	.34202	.65798	2.9238	.36397	2.7475	1.0642	.06031	.93969	70
	15	.34612	.65388	2.8892	.36892	2.7106	1.0659	.06181	.93819	
	30	.35021	.64979	2.8554	.37388	2.6746	1.0676	.06333	.93607	
	45	.35429	.64571	2.8225	.37887	2.6395	1.0694	.06486	.93514	
21	0	.35837	.64163	2.7904	.38386	2.6051	1.0711	.06642	.93358	69
	15	.36244	.63756	2.7591	.38888	2.5715	1.0729	.06799	.93201	
	30	.36650	.63350	2.7285	.39391	2.5386	1.0748	.06958	.93042	
	45	.37056	.62944	2.6986	.39896	2.5065	1.0766	.07119	.92881	
22	0	.37461	.62539	2.6695	.40403	2.4751	1.0785	.07282	.92718	68
	15	.37865	.62135	2.6410	.40911	2.4443	1.0804	.07446	.92554	
	30	.38268	.61732	2.6131	.41421	2.4142	1.0824	.07612	.92388	
	45	.38671	.61329	2.5859	.41933	2.3847	1.0844	.07780	.92220	
23	0	.39073	.60927	2.5593	.42447	2.3559	1.0864	.07950	.92050	67
	15	.39474	.60526	2.5333	.42963	2.3276	1.0884	.08121	.91879	
	30	.39875	.60125	2.5078	.43481	2.2998	1.0904	.08294	.91706	
	45	.40275	.59725	2.4829	.44001	2.2727	1.0925	.08469	.91531	
24	0	.40674	.59326	2.4586	.44523	2.2460	1.0946	.08645	.91355	66
	15	.41072	.58928	2.4348	.45047	2.2199	1.0968	.08824	.91176	
	30	.41469	.58531	2.4114	.45573	2.1943	1.0989	.09004	.90996	
	45	.41866	.58134	2.3886	.46101	2.1692	1.1011	.09186	.90814	
25	0	.42262	.57738	2.3662	.46631	2.1445	1.1034	.09369	.90631	65
	15	.42657	.57343	2.3443	.47163	2.1203	1.1056	.09554	.90446	
	30	.43051	.56949	2.3228	.47697	2.0965	1.1079	.09741	.90259	
	45	.43445	.56555	2.3018	.48234	2.0732	1.1102	.09930	.90070	
26	0	.43837	.56163	2.2812	.48773	2.0503	1.1126	.10121	.89879	64
	15	.44229	.55771	2.2610	.49314	2.0278	1.1150	.10313	.89687	
	30	.44620	.55380	2.2412	.49858	2.0057	1.1174	.10507	.89493	
	45	.45010	.54990	2.2217	.50404	1.9840	1.1198	.10702	.89298	
27	0	.45399	.54601	2.2027	.50952	1.9626	1.1223	.10899	.89101	63
	15	.45787	.54213	2.1840	.51503	1.9416	1.1248	.11098	.88902	
	30	.46175	.53825	2.1657	.52057	1.9210	1.1274	.11299	.88701	
	45	.46561	.53439	2.1477	.52612	1.9007	1.1300	.11501	.88499	
28	0	.46947	.53053	2.1300	.53171	1.8807	1.1326	.11705	.88295	62
	15	.47332	.52668	2.1127	.53732	1.8611	1.1352	.11911	.88089	
	30	.47716	.52284	2.0957	.54295	1.8418	1.1379	.12118	.87882	
	45	.48099	.51901	2.0790	.54862	1.8228	1.1406	.12327	.87673	
29	0	.48481	.51519	2.0627	.55431	1.8040	1.1433	.12538	.87462	61
	15	.48862	.51138	2.0466	.56003	1.7856	1.1461	.12750	.87250	
	30	.49242	.50758	2.0308	.56577	1.7675	1.1490	.12964	.87036	
	45	.49622	.50378	2.0152	.57155	1.7496	1.1518	.13180	.86820	
30	0	.50000	.50000	2.0000	.57735	1.7320	1.1547	.13397	.86603	60
		Cosine.	Ver. Sin.	Secant.	Cotan.	Tang.	Cosec.	Co-Vers.	Sine.	°

75° read from bottom of table upwar

o-Vers.	Cosec.	Tang.	Cotan.	Secant.	Ver. Sin.	Cosine.		
50000	2.0000	.57735	1.7320	1.1547	.13397	.86603	60	0
49623	1.9850	.58318	1.7147	1.1576	.13616	.86384		45
49246	1.9703	.58904	1.6977	1.1606	.13837	.86163		30
8871	1.9558	.59494	1.6808	1.1636	.14059	.85941		15
8496	1.9416	.60086	1.6643	1.1666	.14283	.85717	59	0
8123	1.9276	.60681	1.6479	1.1697	.14509	.85491		45
7750	1.9139	.61280	1.6319	1.1728	.14736	.85264		30
379	1.9004	.61882	1.6160	1.1760	.14965	.85035		15
008	1.8871	.62487	1.6003	1.1792	.15195	.84805	58	0
639	1.8740	.63095	1.5849	1.1824	.15427	.84573		45
270	1.8612	.63707	1.5697	1.1857	.15661	.84339		30
903	1.8485	.64322	1.5547	1.1890	.15896	.84104		15
536	1.8361	.64941	1.5399	1 1924	.16133	.83867	57	0
171	1.8238	.65563	1.5253	1.1958	.16371	.83629		45
806	1.8118	.66188	1.5108	1.1992	.16611	.83389		30
443	1.7999	.66818	1.4966	1.2027	.16853	.83147		15
81	1.7883	.67451	1.4826	1.2062	.17096	.82904	56	0
20	1.7768	.68087	1.4687	1.2098	.17341	.82659		45
59	1.7655	.68728	1.4550	1.2134	.17587	.82413		30
00	1.7544	.69372	1.4415	1.2171	.17835	.82165		15
42	1.7434	.70021	1.4281	1.2208	.18085	.81915	55	0
85	1.7327	.70673	1.4150	1.2245	.18336	.81664		45
30	1.7220	.71329	1.4019	1.2283	.18588	.81412		30
75	1.7116	.71990	1.3891	1.2322	.18843	.81157		15
21	1.7013	.72654	1.3764	1.2361	.19098	.80902	54	0
69	1.6912	.73323	1.3638	1.2400	.19356	.80644		45
8	1.6812	.73996	1.3514	1.2440	.19614	.80386		30
8	1.6713	.74673	1.3392	1.2480	.19875	.80125		15
9	1.6616	.75355	1.3270	1.2521	.20136	.79864	53	0
1	1.6521	.76042	1.3151	1.2563	.20400	.79600		45
4	1.6427	.76733	1.3032	1.2605	.20665	.79335		30
8	1.6334	.77428	1.2915	1.2647	.20931	.79069		15
4	1.6243	.78129	1.2799	1.2690	.21199	.78801	52	0
1	1.6153	.78834	1.2685	1.2734	.21468	.78532		45
9	1.6064	.79543	1.2572	1.2778	.21739	.78261		3°
3	1.5976	.80258	1.2460	1.2822	.22012	.77988		15
3	1.5890	.80978	1.2349	1.2868	.22285	.77715	51	0
9	1.5805	.81703	1.2239	1.2913	.22561	.77439		45
2	1.5721	.82434	1.2131	1.2960	.22838	.77162		30
5	1.5639	.83169	1.2024	1.3007	.23116	.76884		15
1	1.5557	.83910	1.1918	1.3054	.23396	.76604	50	0
3	1.5477	.84656	1.1812	1.3102	.23677	.76323		45
5	1.5398	.85408	1.1708	1.3151	.23959	.76041		30
4	1.5320	.86165	1.1606	1.3200	.24244	.75756		15
4	1.5242	.86929	1.1504	1.3250	.24529	.75471	49	0
5	1.5166	.87698	1.1403	1.3301	.24816	.75184		45
3	1.5092	.88472	1.1303	1.3352	.25104	.74896		30
2	1.5018	.89253	1.1204	1.3404	.25394	.74606		15
7	1.4945	.90040	1.1106	1.3456	.25686	.74314	48	0
3	1.4873	.90834	1.1009	1.3509	.25978	.74022		45
1	1.4802	.91633	1.0913	1.3563	.26272	.73728		30
0	1.4732	.92439	1.0818	1.3618	.26568	.73432		15
0	1.4663	.93251	1.0724	1.3673	.26865	.73135	47	0
2	1.4595	.94071	1.0630	1.3729	.27163	.72837		45
5	1.4527	.94896	1.0538	1.3786	.27463	.72537		30
9	1.4461	.95729	1.0446	1.3843	.27764	.72236		15
4	1.4396	.96569	1.0355	1.3902	.28066	.71934	46	0
1	1.4331	.97416	1.0265	1.3961	.28370	.71630		45
9	1.4267	.98270	1.0176	1.4020	.28675	.71325		30
9	1.4204	.99131	1.0088	1.4081	.28981	.71019		15
9	1.4142	1.0000	1.0000	1.4142	.29289	.70711	45	0
n.	Secant.	Cotan.	Tang.	Cosec.	Co-Vers.	Sine.	°	M.

0° read from bottom of table upwards.

LOGARITHMIC SINES, ETC.

Deg.	Sine.	Cosec.	Versin.	Tangent.	Cotan.	Covers.	Secant.
0	In.Neg.	Infinite.	In.Neg.	In.Neg.	Infinite.	10.00000	10.00000
1	8.24186	11.75814	6.18271	8.24192	11.75808	9.99235	10.00007
2	8.54282	11.45718	6.78474	8.54308	11 45692	9.98457	10.00026
3	8.71880	11.28120	7.13687	8.71940	11.28060	9.97665	10.00060
4	8.84358	11.15642	7.38667	8.84464	11.15536	9.96860	10.00106
5	8.94030	11.05970	7.58039	8.94195	11.05805	9.96040	10.00166
6	9.01923	10.98077	7.73863	9.02162	10.97838	9.95205	10.00239
7	9.08589	10.91411	7.87238	9.08914	10.91086	9.94356	10.00325
8	9.14356	10.85644	7.98820	9.14780	10.85220	9.93492	10.00425
9	9.19433	10.80567	8.09032	9.19971	10.80029	9.92612	10.00538
10	9.23967	10.76033	8.18162	9.24632	10.75368	9.91717	10.00665
11	9.28060	10.71940	8.26418	9 28865	10.71135	9.90805	10.00805
12	9.31788	10.68212	8.33950	9.32747	10.67253	9.89877	10.00960
13	9.35209	10 64791	8.40875	9.36336	10.63664	9.88933	10.01128
14	9.38368	10.61632	8.47282	9.39677	10.60323	9.87971	10.01310
15	9.41300	10.58700	8.53243	9.42805	10.57195	9.86992	10.01506
16	9.44034	10.55966	8.58814	9.45750	10.54250	9.85996	10.01716
17	9.46594	10.53406	8.64043	9.48534	10.51466	9.84981	10.01940
18	9.48998	10.510 2	8.68969	9.51178	10.48822	9.83947	10.02179
19	9.51264	10.48736	8.73625	9.53697	10.46303	9.82894	10.02433
20	9.53405	10.46595	8.78037	9.56107	10.43893	9.81821	10.02701
21	9.55433	10.44567	8.82230	9.58418	10.41582	9.80729	10.02985
22	9.57358	10.42642	8.86223	9.60641	10.39359	9.79615	10.03283
23	9 59188	10.40812	8.90034	9.62785	10.37215	9.78481	10.03597
24	9.60931	10.39069	8.93679	9.64858	10.35142	9.77325	10.03927
25	9.62595	10 37405	8.97170	9.66867	10.33133	9.76146	10.04272
26	9.64184	10.35816	9 00521	9.68818	10.31182	9.74945	10.04634
27	9.65705	10.34295	9.03740	9.70717	10.29283	9.73720	10.05012
28	9.67161	10.32839	9.06838	9.72567	10.27433	9.72471	10.05407
29	9.68557	10.31443	9.09823	9.74375	10.25625	9.71197	10.05818
30	9.69897	10.30103	9.12702	9.76144	10.23856	9 69897	10.06247
31	9.71184	10.28816	9.15483	9.77877	10.22123	9.68571	10 06693
32	9.72421	10.27579	9.18171	9.79579	10.20421	9.67217	10.07158
33	9.73611	10.26389	9.20771	9.81252	10.18748	9.65836	10.07641
34	9.74756	10.25244	9.23290	9.82899	10.17101	9.64425	10.08143
35	9.75859	10.24141	9.25731	9.84523	10.15477	9 62984	10.08664
36	9 76922	10.23078	9.28099	9.86126	10.13874	9.61512	10.09204
37	9.77946	10.22054	9.30398	9.87711	10.12289	9.60008	10.09765
38	9 78934	10.21066	9.32631	9.89281	10.10719	9.58471	10.10347
39	9.79887	10.20113	9.34802	9.90837	10.09163	9.56900	10.10950
40	9.80807	10.19193	9.36913	9.92381	10.07619	9.55293	10.11575
41	9 81694	10.18306	9.38968	9.93916	10.06084	9.53648	10.12222
42	9.82551	10.17449	9.40969	9.95444	10.04556	9.51966	10.12893
43	9 83378	10.16622	9.42918	9.96966	10.03034	9.50243	10.13587
44	9.84177	10.15823	9.44818	9.98484	10.01516	9.48479	10.14307
45	9.84949	10.15052	9.46671	10.00000	10.00000	9 46671	10.15052
	Cosine.	Secant.	Covers.	Cotan.	Tangent.	Versin.	Cosec.

From 45° to 90° read from bottom of table upw

MATERIALS.

THE CHEMICAL ELEMENTS.

The Common Elements (42).

Name.	Atomic Weight.	Chemical Symbol.	Name.	Atomic Weight.	Chemical Symbol.	Name.	Atomic Weight.
Aluminum	27.1	F	Fluorine	19.	Pd	Palladium	106.
Antimony	120.4	Au	Gold	197.2	P	Phosphorus	31.
Arsenic	75.1	H	Hydrogen	1.01	Pt	Platinum	194.9
Barium	137.4	I	Iodine	126.8	K	Potassium	39.1
Bismuth	208.1	Ir	Iridium	193.1	Si	Silicon	28.4
Boron	10.9	Fe	Iron	56.	Ag	Silver	107.9
Bromine	79.9	Pb	Lead	206.9	Na	Sodium	23.
Cadmium	111.9	Li	Lithium	7.03	Sr	Strontium	87.6
Calcium	40.1	Mg	Magnesium	24.3	S	Sulphur	32.1
Carbon	12.	Mn	Manganese	55.	Sn	Tin	119.
Chlorine	35.4	Hg	Mercury	200.	Ti	Titanium	48.1
Chromium	52.1	Ni	Nickel	58.7	W	Tungsten	184.8
Cobalt	59.	N	Nitrogen	14.	Va	Vanadium	51.4
Copper	63.6	O	Oxygen	16.	Zn	Zinc	65.4

atomic weights of many of the elements vary in the decimal place as by different authorities. The above are the most recent values re- to O = 16 and H = 1.008. When H is taken as 1, O = 15.879, and the figures are diminished proportionately. (See *Jour. Am. Chem. Soc.*, 1896.)

The Rare Elements (27).

m, Be.	Glucinum, G.	Rubidium, Rb.	Thallium, Tl.
, Cs.	Indium, In.	Ruthenium, Ru.	Thorium, Th.
, Ce.	Lanthanum, La.	Samarium, Sm.	Uranium, U.
um, D.	Molybdenum, Mo.	Scandium, Sc.	Ytterbium, Yr.
, E.	Niobium, Nb.	Selenium, Se.	Yttrium, Y.
, Ga.	Osmium, Os.	Tantalum, Ta.	Zirconium, Zr.
ium, Ge.	Rhodium, R.	Tellurium, Te.	

SPECIFIC GRAVITY.

pecific gravity of a substance is its weight as compared with the f an equal bulk of pure water.

nd the specific gravity of a substance.
weight of body in air; w = weight of body submerged in water.

$$\text{Specific gravity} = \frac{W}{W - w}.$$

substance be lighter than the water, sink it by means of a heavier e, and deduct the weight of the heavier substance.

c-gravity determinations are usually referred to the standard of the f water at 62° F., 62.355 lbs. per cubic foot. Some experimenters d 60° F. as the standard, and others 32° and 39.1° F. There is no agreement.

sp. gr. referred to water at 39.1° F., to reduce it to the standard of *ltiply it by 1.00112.*

. gr. referred to water at 62° F., to find weight per cubic foot mul- *2.355. Given weight per cubic foot*, to find sp. gr. multiply by *iven sp. gr., to find weight per cubic inch* multiply by .036085.

Weight and Specific Gravity of Metals.

	Specific Gravity. Range according to several Authorities.	Specific Gravity. Approx. Mean Value, used in Calculation of Weight.	Weight per Cubic Foot, lbs.
Aluminum	2.56 to 2.71	2.67	166.5
Antimony	6.66 to 6.86	6.76	421.6
Bismuth	9.74 to 9.90	9.82	612.4
Brass: Copper + Zinc 80 20	7.8 to 8.6	8.60	536.3
70 30		8.40	523.8
60 40		8.36	521.3
50 50		8.20	511.4
Bronze { Copper, 95 to 80 / Tin, 5 to 20 }	8.52 to 8.96	8.853	552.
Cadmium	8.6 to 8.7	8.65	539.
Calcium	1.58		
Chromium	5.0		
Cobalt	8.5 to 8.6		
Gold, pure	19.245 to 19.361	19.258	1200.9
Copper	8.69 to 8.92	8.853	552.
Iridium	22.38 to 23.		1396.
Iron, Cast	6.85 to 7.48	7.218	450.
" Wrought	7.4 to 7.9	7.70	480.
Lead	11.07 to 11.44	11.38	709.7
Manganese	7. to 8.	8.	499.
Magnesium	1.69 to 1.75	1.75	109.
Mercury 32°	13.60 to 13.62	13.62	849.3
Mercury 60°	13.58	13.58	846.8
Mercury 212°	13.37 to 13.38	13.38	834.4
Nickel	8.279 to 8.93	8.8	548.7
Platinum	20.33 to 22.07	21.5	1347.0
Potassium	0.865		
Silver	10.474 to 10.511	10.505	655.1
Sodium	0.97		
Steel	7.69* to 7.932†	7.854	489.6
Tin	7.291 to 7.409	7.350	458.3
Titanium	5.3		
Tungsten	17. to 17.6		
Zinc	6.86 to 7.20	7.00	436.5

* Hard and burned.

† Very pure and soft. The sp. gr. decreases as the carbon is increa[...]

In the first column of figures the lowest are usually those of cast m[...] which are more or less porous; the highest are of metals finely rol[...] drawn into wire.

Specific Gravity of Liquids at 60° F.

Acid, Muriatic	1.200	Oil, Olive	.92
" Nitric	1.217	" Palm	.97
" Sulphuric	1.849	" Petroleum	.78 to
Alcohol, pure	.794	" Rape	.92
" 95 per cent	.816	" Turpentine	.87
" 50 " "	.934	" Whale	.92
Ammonia, 27.9 per cent	.891	Tar	1.
Bromine	2.97	Vinegar	1.08
Carbon disulphide	1.26	Water	1.
Ether, Sulphuric	.72	" sea	1.026 to
Oil, Linseed	.94		

Compression of the following Fluids under a Pressu[re of] 15 lbs. per Square Inch.

Water	.00004663	Ether	[illegible]
Alcohol	.0000216	Mercury	[illegible]

PROPERTIES OF THE USEFUL METALS.

ninum, Al.—Atomic weight 27.1. Specific gravity 2.6 to 2.7. htest of all the useful metals except magnesium. A soft, ductile, le metal, of a white color, approaching silver, but with a bluish cast. n-corrosive. Tenacity about one third that of wrought-iron. For- rare metal, but since 1890 its production and use have greatly in- l on account of the discovery of cheap processes for reducing it from . Melts at about 1160° F. For further description see Aluminum, Strength of Materials.

timony (Stibium), **Sb.**—At. wt. 120.4. Sp. gr. 6.7 to 6.8. A brittle of a bluish-white color and highly crystalline or laminated structure. at 842° F. Heated in the open air it burns with a bluish-white flame. ef use is for the manufacture of certain alloys, as type-metal (anti- 1, lead 4), britannia (antimony 1, tin 9), and various anti-friction s (see Alloys). Cubical expansion by heat from 32° to 212° F., 0.0070. fic heat .050.

smuth, Bi.—At. wt. 208.1. Bismuth is of a peculiar light reddish highly crystalline, and so brittle that it can readily be pulverized. It at 510° F., and boils at about 2300° F. Sp. gr. 9.823 at 54° F., and just above the melting-point. Specific heat about .0301 at ordinary eratures. Coefficient of cubical expansion from 32° to 212°, 0.0040. Con- ity for heat about 1/56 and for electricity only about 1/80 of that of r. Its tensile strength is about 6400 lbs. per square inch. Bismuth ex- s in cooling, and Tribe has shown that this expansion does not take until after solidification. Bismuth is the most diamagnetic element n, a sphere of it being repelled by a magnet.

dmium, Cd.—At. wt. 112. Sp. gr. 8.6 to 8.7. A bluish-white metal. ous, with a fibrous fracture. Melts below 500° F. and volatilizes at t 680° F. It is used as an ingredient in some fusible alloys with lead, and bismuth. Cubical expansion from 32° to 212° F., 0.0094.

pper, Cu.—At. wt. 63.2. Sp. gr. 8.81 to 8.95. Fuses at about 1930° Distinguished from all other metals by its reddish color. Very ductile malleable, and its tenacity is next to iron. Tensile strength 20,000 to 0 lbs. per square inch. Heat conductivity 73.6% of that of silver, and su- r to that of other metals. Electric conductivity equal to that of gold silver. Expansion by heat from 32° to 212° F., 0.0051 of its volume. ific heat .093. (See Copper under Strength of Materials; also Alloys.)

old (Aurum), **Au.**—At. wt. 197.2. Sp. gr., when pure and pressed in a 19.34. Melts at about 1915° F. The most malleable and ductile of all ls. One ounce Troy may be beaten so as to cover 160 sq. ft. of surface. average thickness of gold-leaf is 1/282000 of an inch, or 100 sq. ft. per e. One grain may be drawn into a wire 500 ft. in length. The ductil- destroyed by the presence of 1/2000 part of lead, bismuth, or antimony. is hardened by the addition of silver or of copper. In U. S. gold coin are 90 parts gold and 10 parts of alloy, which is chiefly copper with a silver. By jewelers the fineness of gold is expressed in carats, pure being 24 carats, three fourths fine 18 carats, etc.

dium.—Iridium is one of the rarer metals. It has a white lustre, re- ling that of steel; its hardness is about equal to that of the ruby; in old it is quite brittle, but at a white heat it is somewhat malleable. It of the heaviest of metals, having a specific gravity of 22.38. It is ex- ely infusible and almost absolutely inoxidizable.

uses of iridium, methods of manufacturing it, etc., see paper by W. D. y on the "Iridium Industry," Trans. A. I. M. E. 1884.

n (Ferrum), **Fe.**—At. wt. 56. Sp. gr.: Cast, 6.85 to 7.48; Wrought, 7.9. Pure iron is extremely infusible, its melting point being above F., but its fusibility increases with the addition of carbon, cast iron fus- out 2500° F. Conductivity for heat 11.9, and for electricity 12 to 14.8, being 100. Expansion in bulk by heat: cast iron .0033, and wrought iron from 32° to 212° F. Specific heat: cast iron .1298, wrought iron .1138, 165. Cast iron exposed to continued heat becomes permanently ex- 1 1½ to 3 per cent of its length. Grate-bars should therefore be d about 4 per cent play. (For other properties see Iron and Steel Strength of Materials.)

d (Plumbum), **Pb.**—At. wt. 206.9. Sp. gr. 11.07 to 11.44 by different ties. Melts at about *625° F., softens and becomes pasty at about* *If broken by a sudden blow when just below the melting-point it is ittle and the fracture appears crystalline.* Lead is very malleable

Weight and Specific Gravity of Stones, Brick, Cement, etc.

	Pounds per Cubic Foot.	Specific Gravity.
Asphaltum	87	1.39
Brick, Soft	100	1.6
" Common	112	1.79
" Hard	125	2.0
" Pressed	135	2.16
" Fire	140 to 150	2.24 to 2.[illegible]
Brickwork in mortar	100	1.6
" " cement	112	1.79
Cement, Rosendale, loose	60	.96
" Portland, "	78	1.25
Clay	120 to 150	1.92 to 2.[illegible]
Concrete	120 to 140	1.92 to 2.[illegible]
Earth, loose	72 to 80	1.15 to 1.[illegible]
" rammed	90 to 110	1.44 to 1.[illegible]
Emery	250	4.
Glass	156 to 172	2.5 to 2.[illegible]
" flint	180 to 196	2.88 to 3.[illegible]
Gneiss / Granite	160 to 170	2.56 to 2[illegible]
Gravel	100 to 120	1.6 to 1[illegible]
Gypsum	130 to 150	2.08 to 2[illegible]
Hornblende	200 to 220	3.2 to 3[illegible]
Lime, quick, in bulk	50 to 55	.8 to [illegible]
Limestone	170 to 200	2.72 to [illegible]
Magnesia, Carbonate	150	2.4
Marble	160 to 180	2.56 to [illegible]
Masonry, dry rubble	140 to 160	2.24 to [illegible]
" dressed	140 to 180	2.24 to [illegible]
Mortar	90 to 100	1.44 to [illegible]
Pitch	72	1.15
Plaster of Paris	74 to 80	1.18 to [illegible]
Quartz	165	2.64
Sand	90 to 110	1.44 to [illegible]
Sandstone	140 to 150	2.24 to [illegible]
Slate	170 to 180	2.72 to [illegible]
Stone, various	135 to 200	2.16 to [illegible]
Trap	170 to 200	2.72 to [illegible]
Tile	110 to 120	1.76 to [illegible]
Soapstone	166 to 175	2.65 to [illegible]

Specific Gravity and Weight of Gases at Atmosp[illegible] Pressure and 32° F.

(For other temperatures and pressures see pp. 459, 479.)

	Density, Air = 1.	Grammes per Litre.	Lbs. per Cu. Ft.	Cu[illegible] pe[illegible]
Air	1.0000	1.2931	0.080728	1[illegible]
Oxygen	1.1051	1.4290	0.08921	11[illegible]
Hydrogen	0.0695	0.08987	0.00561	17[illegible]
Nitrogen	0.9714	1.2561	0.07842	1[illegible]
Carbonic oxide, CO	0.9674	1.251	0.07810	1[illegible]
Carbonic acid, CO_2	1.5290	1.977	0.12343	[illegible]
Marsh-gas, methane, CH_4	0.5560	0.719	0.04488	2[illegible]
Ethylene	0.9847	1.273	0.07949	1[illegible]

0.0088. Specific heat .096. Electric conductivity 29, heat conductivity 36, silver being 100. Its principal uses are for coating iron surfaces, called "galvanizing," and for making brass and other alloys.

Table Showing the Order of

Malleability.	Ductility.	Tenacity.	Infusibility.
Gold	Platinum	Iron	Platinum
Silver	Silver	Copper	Iron
Aluminum	Iron	Aluminum	Copper
Copper	Copper	Platinum	Gold
Tin	Gold	Silver	Silver
Lead	Aluminum	Zinc	Aluminum
Zinc	Zinc	Gold	Zinc
Platinum	Tin	Tin	Lead
Iron	Lead	Lead	Tin

FORMULÆ AND TABLE FOR CALCULATING THE WEIGHT OF RODS, BARS, PLATES, TUBES, AND SPHERES OF DIFFERENT MATERIALS.

Notation: b = breadth, t = thickness, s = side of square, d = external diameter, d_1 = internal diameter, all in inches.

Sectional areas: of square bars $= s^2$; of flat bars $= bt$; of round rods $= .7854d^2$; of tubes $= .7854(d^2 - d_1^2) = 3.1416(dt - t^2)$.

Volume of 1 foot in length: of square bars $= 12s^2$; of flat bars $= 12bt$; of round bars $= 9.4248d^2$; of tubes $= 9.4248(d^2 - d_1^2) = 37.6992(dt - t^2)$, in cubic inches.

Weight per foot length = volume × weight per cubic inch of the material.
Weight of a sphere = diam.³ × .5236 × weight per cubic inch.

Material.	Specific Gravity.	Weight per cubic foot, lbs.	Weight of Plates 1 inch thick per per sq. ft., lbs.	Weight of Square Bars per foot length, lbs.	Weight of Flat Bars per foot length, lbs.	Weight per cubic inch, lbs.	Relative Weights. Wrought Iron =1.	Weight of Round Rod per foot length, lbs.	Weight of Spheres or Balls, lbs.
Cast Iron	7.218	450.	37.5	$3\frac{1}{8}s^2$	$3\frac{1}{8}bt$	.2604	15–16	$2.454d^2$	$.1363d^3$
Wrought Iron....	7.7	480.	40.	$3\frac{1}{3}s^2$	$3\frac{1}{3}bt$	.2779	1.	$2.618d^2$	$.1455d^3$
Steel.............	7.854	489.6	40.8	$3.4s^2$	$3.4bt$	.2833	1.02	$2.670d^2$	$.1484d^3$
Copper & Bronze (copper and tin)	8.855	552.	46.	$3.833s^2$	$3.833bt$	.3195	1.15	$3.011d^2$	$.1673d^3$
Brass { 65 Copper.. 35 Zinc.....	8.393	523.2	43.6	$3.633s^2$	$3.633bt$	.3029	1.09	$2.854d^2$	$.1586d^3$
Lead..........	11.38	709.6	59.1	$4.93s^2$	$4.93bt$	.4106	1.48	$3.870d^2$	$.2150d^3$
Aluminum.........	2.67	166.5	13.9	$1.16s^2$	$1.16bt$	.0963	0.347	$0.908d^2$	$.0504d^3$
Glass....	2.62	163.4	13.6	$1.13s^2$	$1.13bt$	.0945	0.34	$0.891d^2$	$.0495d^3$
Pine Wood, dry ...	0.481	30.0	2.5	$0.21s^2$	$0.21bt$	.0174	1–16	$0.164d^2$	$.0091d^3$

For tubes use the coefficient of d^2 in ninth column, as for rods, and multiply it into $(d^2 - d_1^2)$; or take four times this coefficient and multiply it into $(dt - t^2)$.

For hollow spheres use the coefficient of d^3 in the last column and multiply it into $(d^3 - d_1^3)$.

MEASURES AND WEIGHTS OF VARIOUS MATERIALS (APPROXIMATE).

Brickwork.—Brickwork is estimated by the thousand, and for various thicknesses of wall runs as follows:

8¼-in. wall, or 1 brick in thickness, 14 bricks per superficial foot.
12¾ " " " 1½ " " " 21 " " " "
17 " " " 2 " " " 28 " " " "
21½ " " " 2½ " " " 35 " " " "

An ordinary brick measures about 8¼ × 4 × 2 inches, which is equal to 66 cubic inches, or 26.2 bricks to a cubic foot. The average weight is 4½ lbs.

Fuel.—A bushel of bituminous coal weighs 76 pounds and contains 2688 cubic inches = 1.554 cubic feet. 29.47 bushels = 1 gross ton.

A bushel of coke weighs 40 lbs. (35 to 42 lbs.).

One acre of bituminous coal contains 1600 tons of 2240 lbs. per foot of thickness of coal worked. 15 to 25 per cent must be deducted for waste in mining.

41 to 45 cubic feet bituminous coal when broken down	= 1 ton, 2240 lbs.
34 to 41 " " anthracite, prepared for market	= 1 ton, 2240 lbs.
123 " " of charcoal	= 1 ton, 2240 lbs.
70.9 " " " coke	= 1 ton, 2240 lbs.
1 cubic foot of anthracite coal (see also page 625)	= 55 to 66 lbs.
1 " " " bituminous "	= 50 to 55 lbs.
1 " " Cumberland coal	= 53 lbs.
1 " " Cannel coal	= 50.3 lbs.
1 " " charcoal (hardwood)	= 18.5 lbs.
1 " " " (pine)	= 18 lbs.

A bushel of charcoal.—In 1881 the American Charcoal-Iron Workers' Association adopted for use in its official publications for the standard bushel of charcoal 2748 cubic inches, or 20 pounds. A ton of charcoal is to be taken at 2000 pounds. This figure of 20 pounds to the bushel was taken as a fair average of different bushels used throughout the country, and it has since been established by law in some States.

Ores, Earths, etc.

13 cubic feet of ordinary gold or silver ore, in mine	= 1 ton = 2000 lbs.
20 " " " broken quartz	= 1 ton = 2000 lbs.
18 feet of gravel in bank	= 1 ton.
27 cubic feet of gravel when dry	= 1 ton.
25 " " " sand	= 1 ton.
18 " " " earth in bank	= 1 ton.
27 " " " " when dry	= 1 ton.
17 " " " clay	= 1 ton.

Cement.—English Portland, sp. gr. 1.25 to 1.51, per bbl.... 400 to 430 lbs.
Rosendale, U. S., a struck bushel .... 62 to 70 lbs.

Lime.—A struck bushel .... 72 to 75 lbs.

Grain.—A struck bushel of wheat = 60 lbs.; of corn = 56 lbs.; of oats = 30 lbs.

Salt.—A struck bushel of salt, coarse, Syracuse, N. Y. = 56 lbs.; Turk's Island = 76 to 80 lbs.

Weight of Earth Filling.

(From Howe's "Retaining Walls.")

	Average weight in lbs. per cubic foot.
Earth, common loam, loose	72 to 80
" " " shaken	82 to 92
" " " rammed moderately	90 to 100
Gravel	90 to 106
Sand	90 to 106
Soft flowing mud	104 to 120
Sand, perfectly wet	118 to 129

COMMERCIAL SIZES OF IRON BARS.

Flats.

Width.	Thickness.	Width.	Thickness.	Width.	Thickness.
¾	⅛ to ⅝	1⅞	½ to 1½	4	¼ to 2
⅞	⅛ to ¾	2	⅛ to 1¾	4½	¼ to 2
1	⅛ to 15/16	2¼	¼ to 1¾	5	¼ to 2
1⅛	⅛ to 1	2⅜	¼ to 1⅛	5½	¼ to 2
1¼	⅛ to 1⅛	2½	3/16 to 1¾	6	¼ to 2
1⅜	⅛ to 1⅛	2⅝	¼ to 1⅛	6½	¼ to 2
1½	⅛ to 1¼	2¾	¼ to 1⅛	7	¼ to 2
1⅝	½ to 1½	3	¼ to 2	7½	¼ to 2
1¾	3/16 to 1½	3½	¼ to 2		

ls: ¼ to 1⅜ inches, advancing by 16ths, and 1⅜ to 5 inches by

s: 5/16 to 1¼ inches, advancing by 16ths, and 1¼ to 3 inches by

unds: 7/16, ½, ⅝, 11/16, ¾, 1, 1⅛, 1¼, 1½, 1¾, 2 inches.
ns: ⅜ to 1½ inches, advancing by 8ths.
½ × ¼, ⅝ × 5/16, ¾ × ⅜, ⅞ × 7/16 inch.
als: ½ × ⅛, ⅝ × 5/32, ¾ × 3/16, ⅞ × 7/32, 1½ × ½, 1¾ × ⅝,
h.
edge flats: 1½ × ½, 1¾ × ⅝, 1⅞ × ⅝ inch.
½ to 1½ inches, advancing by 8ths, 7 to 16 B. W. gauge.
ches, advancing by 4ths, 7 to 16 gauge up to 3 inches, 4 to 14
5 inches.

ITS OF SQUARE AND ROUND BARS OF IT IRON IN POUNDS PER LINEAL FOOT.

ighing 480 lbs. per cubic foot. For steel add 2 per cent.

Weight of Round Bar One Foot Long.	Thickness or Diameter in Inches.	Weight of Square Bar One Foot Long.	Weight of Round Bar One Foot Long.	Thickness or Diameter in Inches.	Weight of Square Bar One Foot Long.	Weight of Round Bar One Foot Long.
	11/16	24.08	18.91	⅜	96.30	75.64
.010	¾	25.21	19.80	7/16	98.55	77.40
.041	13/16	26.37	20.71	½	100.8	79.19
.092	⅞	27.55	21.64	9/16	103.1	81.00
.164	15/16	28.76	22.59	⅝	105.5	82.83
.256	3	30.00	23.56	11/16	107.8	84.69
.368	1/16	31.26	24.55	¾	110.2	86.56
.501	⅛	32.55	25.57	13/16	112.6	88.45
.654	3/16	33.87	26.60	⅞	115.1	90.36
.828	¼	35.21	27.65	15/16	117.5	92.29
1.023	5/16	36.58	28.73	6	120.0	94.25
1.237	⅜	37.97	29.82	⅛	125.1	98.22
1.473	7/16	39.39	30.94	¼	130.2	102.3
1.728	½	40.83	32.07	⅜	135.5	106.4
2.004	9/16	42.30	33.23	½	140.8	110.6
2.301	⅝	43.80	34.40	⅝	146.3	114.9
2.618	11/16	45.33	35.60	¾	151.9	119.3
2.955	¾	46.88	36.82	⅞	157.6	123.7
3.313	13/16	48.45	38.05	7	163.3	128.3
3.692	⅞	50.05	39.31	⅛	169.2	132.9
4.091	15/16	51.68	40.59	¼	175.2	137.6
4.510	4	53.33	41.89	⅜	181.3	142.4
4.950	1/16	55.01	43.21	½	187.5	147.3
5.410	⅛	56.72	44.55	⅝	193.8	152.2
5.890	3/16	58.45	45.91	¾	200.2	157.2
6.392	¼	60.21	47.29	⅞	206.7	162.4
6.913	5/16	61.99	48.69	8	213.3	167.6
7.455	⅜	63.80	50.11	¼	226.9	178.2
8.018	7/16	65.64	51.55	½	240.8	189.2
8.601	½	67.50	53.01	¾	255.2	200.4
9.204	9/16	69.39	54.50	9	270.0	212.1
9.828	⅝	71.30	56.00	¼	285.2	224.0
10.47	11/16	73.24	57.52	½	300.8	236.3
11.14	¾	75.21	59.07	¾	316.9	248.9
11.82	13/16	77.20	60.63	10	333.3	261.8
12.53	⅞	79.22	62.22	¼	350.2	275.1
13.25	15/16	81.26	63.82	½	367.5	288.6
14.00	5	83.33	65.45	¾	385.2	302.5
14.77	1/16	85.43	67.10	11	403.3	316.8
15.55	⅛	87.55	68.76	¼	421.9	331.3
16.36	3/16	89.70	70.45	½	440.8	346.2
17.19	¼	91.88	72.16	¾	460.2	361.4
18.04	5/16	94.08	73.89	12	480.	377.

WEIGHTS OF FLAT ROLLED IRON IN POUNDS PER LINEAL FOOT.

Widths from 1 In. to 12 In.

Iron weighing 480 lbs. per cubic foot. For steel add 2 per cent.

Thickness in Inches.	Widths. 1″.	1¼″.	1½″.	1¾″.	2″.	2¼″.	2½″.	2¾″.	3″.	3¼″.	3½″.	3¾″.	4″.	4¼″.	4½″.	4¾″.
1-16	.208	.260	.313	.365	.417	.469	.521	.573	.625	.677	.729	.781	.833	.885	.938	.990
⅛	.417	.521	.625	.729	.833	.938	1.04	1.15	1.25	1.35	1.46	1.56	1.67	1.77	1.88	1.98
3-16	.625	.781	.938	1.09	1.25	1.41	1.56	1.72	1.88	2.03	2.19	2.34	2.50	2.66	2.81	2.97
¼	.833	1.04	1.25	1.46	1.67	1.88	2.08	2.29	2.50	2.71	2.92	3.13	3.33	3.54	3.75	3.96
5-16	1.04	1.30	1.56	1.82	2.08	2.34	2.60	2.86	3.13	3.39	3.65	3.91	4.17	4.43	4.69	4.95
⅜	1.25	1.56	1.88	2.19	2.50	2.81	3.13	3.44	3.75	4.06	4.38	4.69	5.00	5.31	5.63	5.94
7-16	1.46	1.82	2.19	2.55	2.92	3.28	3.65	4.01	4.38	4.74	5.10	5.47	5.83	6.20	6.56	6.93
½	1.67	2.08	2.50	2.92	3.33	3.75	4.17	4.58	5.00	5.42	5.83	6.25	6.67	7.08	7.50	7.92
9-16	1.88	2.34	2.81	3.28	3.75	4.22	4.69	5.16	5.63	6.09	6.56	7.03	7.50	7.97	8.44	8.91
⅝	2.08	2.60	3.13	3.65	4.17	4.69	5.21	5.73	6.25	6.77	7.29	7.81	8.33	8.85	9.38	9.90
11-16	2.29	2.86	3.44	4.01	4.58	5.16	5.73	6.30	6.88	7.45	8.02	8.59	9.17	9.74	10.31	10.89
¾	2.50	3.13	3.75	4.38	5.00	5.63	6.25	6.88	7.50	8.13	8.75	9.38	10.00	10.63	11.25	11.88
13-16	2.71	3.39	4.06	4.74	5.42	6.09	6.77	7.45	8.13	8.80	9.48	10.16	10.83	11.51	12.19	12.86
⅞	2.92	3.65	4.38	5.10	5.83	6.56	7.29	8.02	8.75	9.48	10.21	10.94	11.67	12.40	13.13	13.85
15-16	3.13	3.91	4.69	5.47	6.25	7.03	7.81	8.59	9.38	10.16	10.94	11.72	12.50	13.28	14.06	14.84
1	3.33	4.17	5.00	5.83	6.67	7.50	8.33	9.17	10.00	10.83	11.67	12.50	13.33	14.17	15.00	15.83
1 1-16	3.54	4.43	5.31	6.20	7.08	7.97	8.85	9.74	10.63	11.51	12.40	13.28	14.17	15.05	15.94	16.82
1 ⅛	3.75	4.69	5.63	6.56	7.50	8.44	9.38	10.31	11.25	12.19	13.13	14.06	15.00	15.94	16.88	17.81
1 3-16	3.96	4.95	5.94	6.93	7.92	8.91	9.90	10.89	11.88	12.86	13.85	14.84	15.83	16.82	17.81	18.80
1 ¼	4.17	5.21	6.25	7.29	8.33	9.38	10.42	11.46	12.50	13.54	14.58	15.63	16.67	17.71	18.75	19.79
1 5-16	4.37	5.47	6.56	7.66	8.75	9.84	10.94	12.03	13.13	14.22	15.31	16.41	17.50	18.59	19.69	20.78
1 ⅜	4.58	5.73	6.88	8.02	9.17	10.31	11.46	12.60	13.75	14.90	16.04	17.19	18.33	19.48	20.63	21.77
1 7-16	4.79	5.99	7.19	8.39	9.58	10.78	11.98	13.18	14.38	15.57	16.77	17.97	19.17	20.36	21.56	22.76
1 ½	5.00	6.25	7.50	8.75	10.00	11.25	12.50	13.75	15.00	16.25	17.50	18.75	20.00	21.25	22.50	23.75
1 9-16	5.21	6.51	7.81	9.11	10.42	11.72	13.02	14.32	15.63	16.93	18.23	19.53	20.83	22.14	23.44	24.74
1 ⅝	5.42	6.77	8.13	9.48	10.83	12.19	13.54	14.90	16.25	17.60	18.96	20.31	21.67	23.02	24.38	25.73
1 11-16	5.63	7.03	8.44	9.84	11.25	12.66	14.06	15.47	16.88	18.28	19.69	21.09	22.50	23.91	25.31	26.72
1 ¾	5.83	7.29	8.75	10.21	11.67	13.13	14.58	16.04	17.50	18.96	20.42	21.88	23.33	24.79	26.25	27.71
1 13-16	6.04	7.55	9.06	10.57	12.08	13.59	15.10	16.61	18.13	19.64	21.15	22.66	24.17	25.68	27.19	28.70

Thickness in Inches.	Widths. 5″.	5¼″.	5½″.	5¾″.	6″.	6¼″.	6½″.	6¾″.	7″.	7½″.	8″.	8½″.	9″.	10″.	11″.	12″.
1-16	1.04	1.09	1.15	1.20	1.25	1.30	1.35	1.41	1.46	1.56	1.67	1.77	1.88	2.08	2.29	2.50
⅛	2.08	2.19	2.29	2.40	2.50	2.60	2.71	2.81	2.92	3.13	3.33	3.54	3.75	4.17	4.58	5.00
3-16	3.13	3.28	3.44	3.59	3.75	3.91	4.06	4.22	4.38	4.69	5.00	5.31	5.63	6.25	6.88	7.50
¼	4.17	4.38	4.58	4.79	5.00	5.21	5.42	5.63	5.83	6.25	6.67	7.08	7.50	8.33	9.17	10.00
5-16	5.21	5.47	5.73	5.99	6.25	6.51	6.77	7.03	7.29	7.81	8.33	8.85	9.38	10.42	11.46	12.50
⅜	6.25	6.56	6.88	7.19	7.50	7.81	8.13	8.44	8.75	9.38	10.00	10.63	11.25	12.50	13.75	15.00
7-16	7.29	7.66	8.02	8.39	8.75	9.11	9.48	9.84	10.21	10.94	11.67	12.40	13.13	14.58	16.04	17.50
½	8.33	8.75	9.17	9.58	10.00	10.42	10.83	11.25	11.67	12.50	13.33	14.17	15.00	16.67	18.33	20.00
9-16	9.38	9.84	10.31	10.78	11.25	11.72	12.19	12.66	13.13	14.06	15.00	15.94	16.88	18.75	20.63	22.50
⅝	10.42	10.94	11.46	11.98	12.50	13.02	13.54	14.06	14.58	15.63	16.67	17.71	18.75	20.83	22.92	25.00
11-16	11.46	12.03	12.60	13.18	13.75	14.32	14.90	15.47	16.04	17.19	18.33	19.48	20.63	22.92	25.21	27.50
¾	12.50	13.13	13.75	14.38	15.00	15.63	16.25	16.88	17.50	18.75	20.00	21.25	22.50	25.00	27.50	30.00
13-16	13.54	14.22	14.90	15.57	16.25	16.93	17.60	18.28	18.96	20.31	21.67	23.02	24.38	27.08	29.79	32.50
⅞	14.58	15.31	16.04	16.77	17.50	18.23	18.96	19.69	20.42	21.88	23.33	24.79	26.25	29.17	32.08	35.00
15-16	15.63	16.41	17.19	17.97	18.75	19.53	20.31	21.09	21.88	23.44	25.00	26.56	28.13	31.25	34.38	37.50
1	16.67	17.50	18.33	19.17	20.00	20.83	21.67	22.50	23.33	25.00	26.67	28.33	30.00	33.33	36.67	40.00
1⅛	18.75	19.69	20.63	21.56	22.50	23.44	24.38	25.31	26.25	28.13	30.00	31.88	33.75	37.50	41.25	45.00
1¼	20.83	21.88	22.92	23.96	25.00	26.04	27.08	28.13	29.17	31.25	33.33	35.42	37.50	41.67	45.83	50.00
1⅜	22.92	24.06	25.21	26.35	27.50	28.65	29.79	30.94	32.08	34.38	36.67	38.96	41.25	45.83	50.42	55.00
1½	25.00	26.25	27.50	28.75	30.00	31.25	32.50	33.75	35.00	37.50	40.00	42.50	45.00	50.00	55.00	60.00
1⅝	27.08	28.44	29.79	31.15	32.50	33.85	35.21	36.56	37.92	40.63	43.33	46.04	48.75	54.17	59.58	65.00
1¾	29.17	30.63	32.08	33.54	35.00	36.46	37.92	39.38	40.83	43.75	46.67	49.58	52.50	58.33	64.17	70.00
1⅞	31.25	32.81	34.38	35.94	37.50	39.06	40.63	42.19	43.75	46.88	50.00	53.13	56.25	62.50	68.75	75.00
2	33.33	35.00	36.67	38.33	40.00	41.67	43.33	45.00	46.67	50.00	53.33	56.67	60.00	66.67	73.33	80.00

Other sizes.—Weight of other sizes can easily be obtained from the above table by means of combinations or divisions. Thus, for example,

Weight of 12 × 1¼ equals weight of 12 × 1 plus weight of 12 × ¼..................	50.00
Or, twice weight of 12 × ⅝, as it is twice as thick.	50.00
Weight of 6 × 1 15/16 equals midway weight between 6 × 1⅞ and 6 × 2...	38.75
Weight of 24 × 15/16, being twice as wide as 12 × 15/16, weighs........................	75.00

WEIGHTS OF FLAT ROLLED IRON IN POUNDS PER LINEAL FOOT.
Widths from 1 In, to 12 In.

Iron weighing 480 lbs. per cubic foot. For steel add 2 per cent.

	Widths.															
t.	1″.	1¼″.	1½″.	1¾″.	2″.	2¼″.	2½″.	2¾″.	3″.	3¼″.	3½″.	3¾″.	4″.	4¼″.	4½″.	4¾″.
1-16	.208	.260	.313	.365	.417	.469	.521	.573	.625	.677	.729	.781	.833	.885	.938	.990
⅛	.417	.521	.625	.729	.833	.938	1.04	1.15	1.25	1.35	1.46	1.56	1.67	1.77	1.88	1.98
3-16	.625	.781	.938	1.09	1.25	1.41	1.56	1.72	1.88	2.03	2.19	2.34	2.50	2.66	2.81	2.97
¼	.833	1.04	1.25	1.46	1.67	1.88	2.08	2.29	2.50	2.71	2.92	3.13	3.33	3.54	3.75	3.96
5-16	1.04	1.30	1.56	1.82	2.08	2.34	2.60	2.86	3.13	3.39	3.65	3.91	4.17	4.43	4.69	4.95
⅜	1.25	1.56	1.88	2.19	2.50	2.81	3.13	3.44	3.75	4.06	4.38	4.69	5.00	5.31	5.63	5.94
7-16	1.46	1.82	2.19	2.55	2.92	3.28	3.65	4.01	4.38	4.74	5.10	5.47	5.83	6.20	6.56	6.93
½	1.67	2.08	2.50	2.92	3.33	3.75	4.17	4.58	5.00	5.42	5.83	6.25	6.67	7.08	7.50	7.92
9-16	1.88	2.34	2.81	3.28	3.75	4.22	4.69	5.16	5.63	6.09	6.56	7.03	7.50	7.97	8.44	8.91
⅝	2.08	2.60	3.13	3.65	4.17	4.69	5.21	5.73	6.25	6.77	7.29	7.81	8.33	8.85	9.38	9.90
11-16	2.29	2.86	3.44	4.01	4.58	5.16	5.73	6.30	6.88	7.45	8.02	8.59	9.17	9.74	10.31	10.89
¾	2.50	3.13	3.75	4.38	5.00	5.63	6.25	6.88	7.50	8.13	8.75	9.38	10.00	10.63	11.25	11.88
13-16	2.71	3.39	4.06	4.74	5.42	6.09	6.77	7.45	8.13	8.80	9.48	10.16	10.83	11.51	12.19	12.86
⅞	2.92	3.65	4.38	5.10	5.83	6.56	7.29	8.02	8.75	9.48	10.21	10.94	11.67	12.40	13.13	13.85
15-16	3.13	3.91	4.69	5.47	6.25	7.03	7.81	8.59	9.38	10.16	10.94	11.72	12.50	13.28	14.06	14.84
1	3.33	4.17	5.00	5.83	6.67	7.50	8.33	9.17	10.00	10.83	11.67	12.50	13.33	14.17	15.00	15.83
1 1-16	3.54	4.43	5.31	6.20	7.08	7.97	8.85	9.74	10.63	11.51	12.40	13.28	14.17	15.05	15.94	16.82
1 ⅛	3.75	4.69	5.63	6.56	7.50	8.44	9.38	10.31	11.25	12.19	13.13	14.06	15.00	15.94	16.88	17.81
1 3-16	3.96	4.95	5.94	6.93	7.92	8.91	9.90	10.89	11.88	12.86	13.85	14.84	15.83	16.82	17.81	18.80
1 ¼	4.17	5.21	6.25	7.29	8.33	9.38	10.42	11.46	12.50	13.54	14.58	15.63	16.67	17.71	18.75	19.79
1 5-16	4.37	5.47	6.56	7.66	8.75	9.84	10.94	12.03	13.13	14.22	15.31	16.41	17.50	18.59	19.69	20.78
1 ⅜	4.58	5.73	6.88	8.02	9.17	10.31	11.46	12.60	13.75	14.90	16.04	17.19	18.33	19.48	20.63	21.77
1 7-16	4.79	5.99	7.19	8.39	9.58	10.78	11.98	13.18	14.38	15.57	16.77	17.97	19.17	20.36	21.56	22.76
1 ½	5.00	6.25	7.50	8.75	10.00	11.25	12.50	13.75	15.00	16.25	17.50	18.75	20.00	21.25	22.50	23.75
1 9-16	5.21	6.51	7.81	9.11	10.42	11.72	13.02	14.32	15.63	16.93	18.23	19.53	20.83	22.14	23.44	24.74
1 ⅝	5.42	6.77	8.13	9.48	10.83	12.19	13.54	14.90	16.25	17.60	18.96	20.31	21.67	23.02	24.38	25.73
1 11-16	5.63	7.03	8.44	9.84	11.25	12.66	14.06	15.47	16.88	18.28	19.69	21.09	22.50	23.91	25.31	26.72
1 ¾	5.83	7.29	8.75	10.21	11.67	13.13	14.58	16.04	17.50	18.96	20.42	21.88	23.33	24.79	26.25	27.71
1 13-16	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

Thickness in Inches.	Widths. 5″.	5¼″.	5½″.	5¾″.	6″.	6¼″.	6½″.	6¾″.	7″.	7½″.	8″.	8½″.	9″.	10″.	11″.	12″.
1-16	1.04	1.09	1.15	1.20	1.25	1.30	1.35	1.41	1.46	1.56	1.67	1.77	1.88	2.08	2.29	2.50
⅛	2.08	2.19	2.29	2.40	2.50	2.60	2.71	2.81	2.92	3.13	3.33	3.54	3.75	4.17	4.58	5.00
3-16	3.13	3.28	3.44	3.59	3.75	3.91	4.06	4.22	4.38	4.69	5.00	5.31	5.63	6.25	6.88	7.50
¼	4.17	4.38	4.58	4.79	5.00	5.21	5.42	5.63	5.83	6.25	6.67	7.08	7.50	8.33	9.17	10.00
5-16	5.21	5.47	5.73	5.99	6.25	6.51	6.77	7.03	7.29	7.81	8.33	8.85	9.38	10.42	11.46	12.50
⅜	6.25	6.56	6.88	7.19	7.50	7.81	8.13	8.44	8.75	9.38	10.00	10.63	11.25	12.50	13.75	15.00
7-16	7.29	7.66	8.02	8.39	8.75	9.11	9.48	9.84	10.21	10.94	11.67	12.40	13.13	14.58	16.04	17.50
½	8.33	8.75	9.17	9.58	10.00	10.42	10.83	11.25	11.67	12.50	13.33	14.17	15.00	16.67	18.33	20.00
9-16	9.38	9.84	10.31	10.78	11.25	11.72	12.19	12.66	13.13	14.06	15.00	15.94	16.88	18.75	20.63	22.50
⅝	10.42	10.94	11.46	11.98	12.50	13.02	13.54	14.06	14.58	15.63	16.67	17.71	18.75	20.83	22.92	25.00
11-16	11.46	12.03	12.60	13.18	13.75	14.32	14.90	15.47	16.04	17.19	18.33	19.48	20.63	22.92	25.21	27.50
¾	12.50	13.13	13.75	14.38	15.00	15.63	16.25	16.88	17.50	18.75	20.00	21.25	22.50	25.00	27.50	30.00
13-16	13.54	14.22	14.90	15.57	16.25	16.93	17.60	18.28	18.96	20.31	21.67	23.02	24.38	27.08	29.79	32.50
⅞	14.58	15.31	16.04	16.77	17.50	18.23	18.96	19.69	20.42	21.88	23.33	24.79	26.25	29.17	32.08	35.00
15-16	15.63	16.41	17.19	17.97	18.75	19.53	20.31	21.09	21.88	23.44	25.00	26.56	28.13	31.25	34.38	37.50
1	16.67	17.50	18.33	19.17	20.00	20.83	21.67	22.50	23.33	25.00	26.67	28.33	30.00	33.33	36.67	40.00
1⅛	18.75	19.69	20.63	21.56	22.50	23.44	24.38	25.31	26.25	28.13	30.00	31.88	33.75	37.50	41.25	45.00
1¼	20.83	21.88	22.92	23.96	25.00	26.04	27.08	28.13	29.17	31.25	33.33	35.42	37.50	41.67	45.83	50.00
1⅜	22.92	24.06	25.21	26.35	27.50	28.65	29.79	30.94	32.08	34.38	36.67	38.96	41.25	45.83	50.42	55.00
1½	25.00	26.25	27.50	28.75	30.00	31.25	32.50	33.75	35.00	37.50	40.00	42.50	45.00	50.00	55.00	60.00
1⅝	27.08	28.44	29.79	31.15	32.50	33.85	35.21	36.56	37.92	40.63	43.33	46.04	48.75	54.17	59.58	65.00
1¾	29.17	30.63	32.08	33.54	35.00	36.46	37.92	39.38	40.83	43.75	46.67	49.58	52.50	58.33	64.17	70.00
1⅞	31.25	32.81	34.38	35.94	37.50	39.06	40.63	42.19	43.75	46.88	50.00	53.13	56.25	62.50	68.75	75.00
2	33.33	35.00	36.67	38.33	40.00	41.67	43.33	45.00	46.67	50.00	53.33	56.67	60.00	66.67	73.33	80.00

Other sizes.—Weight of other sizes can easily be obtained from the above table by means of combinations or divisions. Thus, for example,

Weight of 12 × 1¼ equals weight of 12 × 1 plus weight of 12 × ¼.................. 50.00
Or, twice weight of 12 × ⅝, as it is twice as thick. 50.00
Weight of 6 × 1 15/16 equals midway weight between 6 × 1⅞ and 6 × 2............. 38.75
Weight of 24 × 15/16 being twice as wide as 12 × 15/16 weighs 75.00

WEIGHT OF IRON AND STEEL SHEETS.

Weights per Square Foot.

(For weights by new U. S. Standard Gauge, see page 31.)

Thickness by Birmingham Gauge.				Thickness by American (Brown Sharpe's) Gauge.			
No. of Gauge.	Thickness in Inches.	Iron.	Steel.	No. of Gauge.	Thickness in Inches.	Iron.	St
0000	.454	18.16	18.52	0000	.46	18.40	18.
000	.425	17.00	17.34	000	.4096	16.38	16.
00	.38	15.20	15.30	00	.3648	14.59	14.
0	.34	13.60	13.87	0	.3249	13.00	13.
1	.3	12.00	12.24	1	.2893	11.57	11.
2	.284	11.36	11.59	2	.2576	10.30	10.
3	.259	10.36	10.57	3	.2294	9.18	9.
4	.238	9.52	9.71	4	.2043	8.17	8.
5	.22	8.80	8.98	5	.1819	7.28	7.
6	.203	8.12	8.28	6	.1620	6.48	6
7	.18	7.20	7.34	7	.1443	5.77	5
8	.165	6.60	6.73	8	.1285	5.14	
9	.148	5.92	6.04	9	.1144	4.58	
10	.134	5.36	5.47	10	.1019	4.08	
11	.12	4.80	4.90	11	.0907	3.63	
12	.109	4.36	4.45	12	.0808	3.23	
13	.095	3.80	3.88	13	.0720	2.88	
14	.083	3.32	3.39	14	.0641	2.56	
15	.072	2.88	2.94	15	.0571	2.28	
16	.065	2.60	2.65	16	.0508	2.03	
17	.058	2.32	2.37	17	.0453	1.81	
18	.049	1.96	2.00	18	.0403	1.61	
19	.042	1.68	1.71	19	.0359	1.44	
20	.035	1.40	1.43	20	.0320	1.28	
21	.032	1.28	1.31	21	.0285	1.14	
22	.028	1.12	1.14	22	.0253	1.01	
23	.025	1.00	1.02	23	.0226	.904	
24	.022	.88	.898	24	.0201	.804	
25	.02	.80	.816	25	.0179	.716	
26	.018	.72	.734	26	.0159	.636	
27	.016	.64	.653	27	.0142	.568	
28	.014	.56	.571	28	.0126	.504	
29	.013	.52	.530	29	.0113	.452	
30	.012	.48	.490	30	.0100	.400	
31	.01	.40	.408	31	.0089	.356	
32	.009	.36	.367	32	.0080	.320	
33	.008	.32	.326	33	.0071	.284	
34	.007	.28	.286	34	.0063	.252	
35	.005	.20	.204	35	.0056	.224	

	Iron.	Steel.
Specific gravity	7.7	7.854
Weight per cubic foot	480.	489.6
" " " inch	.2778	.2833

There are many gauges in use differing from each other, and even ... in specified gauge, as the Birmingham, are not assu... ...cturers, orders for sheets and wires should alw... ...re foot, or the thickness in thousandths of an...

ES AND WEIGHTS OF STRUCTURAL SHAPES.

imum and Maximum Weights and Dimen ions of Carnegie I-Beams.

STEEL BEAMS.

Depth of Beam, in inches.	Weight per Foot, in lbs.		Flange Width.		Web Thickness.		Increase of Web and Flanges for each lb. increase of weight.
	Min.	Max.	Min.	Max.	Min.	Max.	
24	80.00	100.00	6.95	7.20	.50	.75	.0123
20	80.00	100.00	7 00	7.30	.60	.90	.015
20	64.00	75.00	6.25	6.41	.50	.66	.015
15	80.00	100 00	6.41	6.79	.77	1.16	.020
15	60.00	75.00	6.04	6.34	.54	.84	.020
15	50.00	59.00	5.75	5.93	.45	.63	.020
15	41.00	49.00	5.50	5.66	.40	.56	.020
12	40.00	56.70	5.50	5.91	.39	.80	.025
12	32.00	39.00	5.25	5.42	.35	.52	.025
10	33.00	40.00	5.00	5.21	.37	.58	.029
10	25.50	32.00	4.75	4.94	.32	.51	.029
9	27.00	33.00	4.75	4.95	.31	.51	.033
9	21.00	26.00	4.50	4.66	.27	.43	.033
8	22.00	27.00	4.50	4.68	.27	.45	.037
8	18.00	21.70	4.25	4.39	.25	.39	.037
7	20.00	22.00	4.25	4.33	.27	.35	.042
7	15.50	19.00	4.00	4.15	.23	.38	.042
6	16.00	20.00	3.63	3.83	.26	.46	.049
6	13.00	15.00	3.50	3.60	.23	.34	.049
5	13.00	16.00	3.13	3.31	.26	.44	.059
5	10.00	12.00	3.00	3.12	.22	.33	.059
4	10.00	13.00	2.75	2.97	.24	.46	.074
4	7.50	9.00	2.63	2.74	.20	.31	.074
4	6.00	8.00	2.18	2.33	.18	.33	.074

	Iron.	Steel.
weight in pounds per foot, to find sectional area	÷ $3\frac{1}{3}$	3.4
" " " " " " " "	× 0.3	.2941
sectional area, to find weight in lbs. per foot	× $3\frac{1}{3}$	3.4
" " " " " lbs. per yard	× 10	10.2

imum and Minimum Weights and Dimensions of Carnegie Deck Beams.

STEEL.

Depth of Beam, inches.	Weight per Foot, lbs.		Flange Width.		Web Thickness.		Increase of Web and Flanges per lb. increase of weight.
	Min.	Max.	Min.	Max.	Min.	Max.	
10	27.23	35.70	5.25	5.50	.38	.63	.029
9	26.52	30.60	4.94	5.07	.44	.57	.032
8	20.15	24.48	5.00	5.16	.31	.47	.037
7	18.10	23.46	4.87	5.10	.31	.54	.042
6	15.30	18.36	4.38	4.53	.28	.43	.049

WEIGHTS OF STEEL BLOOMS.

Soft steel. 1 cubic inch = 0.284 lb. 1 cubic foot = 490.75 lbs.

Sizes.	Lengths.											
	1″	6″	12″	18″	24″	30″	36″	42″	48″	54″	60″	66″
12″ × 4″	13.63	82	164	245	327	409	491	573	654	736	818	90
11 × 6	18.75	113	225	338	450	563	675	788	900	1013	1125	122
× 5	15.62	94	188	281	375	469	562	656	750	843	937	103
× 4	12.50	75	150	225	300	375	450	525	600	675	750	82
10 × 7	19.88	120	239	358	477	596	715	835	955	1074	1193	131
× 6	17.04	102	204	307	409	511	613	716	818	920	1022	112
× 5	14.20	85	170	256	341	426	511	596	682	767	852	93
× 4	11.36	68	136	205	273	341	409	477	546	614	682	75
× 3	8.52	51	102	153	204	255	306	358	409	460	511	56
9 × 7	17.89	107	215	322	430	537	644	751	859	966	1073	118
× 6	15.34	92	184	276	368	460	552	644	736	828	920	101
× 5	12.78	77	153	230	307	383	460	537	614	690	767	84
× 4	10.22	61	123	184	245	307	368	429	490	552	613	67
8 × 8	18.18	109	218	327	436	545	655	764	873	982	1091	120
× 7	15.9	95	191	286	382	477	572	668	763	859	954	104
× 6	13.63	82	164	245	327	409	491	573	654	736	818	90
× 5	11.36	68	136	205	273	341	409	477	546	614	682	75
× 4	9.09	55	109	164	218	273	327	382	436	491	545	60
7 × 7	13.92	83	167	251	334	418	501	585	668	752	835	91
× 6	11.93	72	143	215	286	358	430	501	573	644	716	78
× 5	9.94	60	119	179	238	298	358	417	477	536	596	65
× 4	7.95	48	96	143	191	239	286	334	382	429	477	52
× 3	5.96	36	72	107	143	179	214	250	286	322	358	39
6½ × 6½	12.	72	144	216	288	360	432	504	576	648	720	79
× 4	7.38	44	89	133	177	221	266	310	354	399	443	48
6 × 6	10.22	61	123	184	245	307	368	429	490	551	613	67
× 5	8.52	51	102	153	204	255	307	358	409	460	511	56
× 4	6.82	41	82	123	164	204	245	286	327	368	409	45
× 3	5.11	31	61	92	123	153	184	214	245	276	307	33
5½ × 5½	8.59	52	103	155	206	258	309	361	412	464	515	56
× 4	6.25	37	75	112	150	188	225	262	300	337	375	41
5 × 5	7.10	43	85	128	170	213	256	298	341	383	426	46
× 4	5.68	34	68	102	136	170	205	239	273	307	341	37
4½ × 4½	5.75	35	69	104	138	173	207	242	276	311	345	38
× 4	5.11	31	61	92	123	153	184	215	246	276	307	33
4 × 4	4.54	27	55	82	109	136	164	191	218	246	272	30
× 3½	3.97	24	48	72	96	119	143	167	181	215	238	26
× 3	3.40	20	41	61	82	102	122	143	163	184	204	22
3½ × 3½	3.48	21	42	63	84	104	125	146	167	188	209	23
× 3	2.98	18	36	54	72	89	107	125	143	161	179	19
3 × 3	2.56	15	31	46	61	77	92	108	123	138	154	16

or Various

16	3/4	7/8	1
[illegible]	[illegible]	[illegible]	1.00
26.5	29.1	34.2	[illegible]
22.8	25.0	29.2	[illegible]
17.7	19.3		

Pounds per Foot for Various
[illegible] in Inches.

	7/16	1/2	9/16	5/8	11/16	3/4	7/8	1
	.4375	.50	.5625	.625	.6875	.75	.875	1.00
		17.0	18.9	20.9	22.8	24.8	28.[illegible]	[illegible]
[illegible]	15.0	17.1	19.[illegible]	21.4	23.6	25.7	30.[illegible]	[illegible]
12.2	14.4	16.4	18.6	20.7	22.8	24.9	[illegible]	[illegible]
11.5	13.6	15.6	17.6	19.7	21.7	23.8	27.8	[illegible]
11.0	12.8	14.6	16.4	18.2				
11.0	12.[illegible]	14.6	16.4	18.2	20.0	21.8		
10.3	12.0	13.6	15.2	16.8	18.5	20.1		
9.7	11.2	12.8	14.3	15.8	17.[illegible]	18.9		
9.2	10.6	12.1	13.6	15.0	16.5	18.0		
9.2	10.6	12.1	13.6	15.0	16.5	18.0		
8.6	10.0	11.4	12.8	14.2				
7.9	9.2	10.5	11.8	13.1				
7.1	8.3	9.4						
6.7								
6.7	7.8	8.9						
6.1	7.1	8.2						
5.4	6.3	7.2						
4.6								
4.0								

Weights and Dimensions of Carnegie Steel Chann[els]

Section Index	Depth of Channel, in inches.	Weight per Foot, in lbs.		Flange Width.		Web Thickness.		Incr[ease] of [Web] a[nd] Fla[nge] for [1] lb. [in]crea[se of] wei[ght]
		Min.	Max.	Min.	Max.	Min.	Max.	
C1	15	32.00	51.00	3.40	3.78	.40	.78	.0[illegible]
C2	12	20.00	30.25	2.90	3.15	.30	.55	.0[illegible]
C3	10	15.25	23.75	2.66	2.91	.26	.51	.0[illegible]
C4	9	12.75	20.50	2.44	2.69	.24	.49	.0[illegible]
C5	8	10.00	17.25	2.20	2.47	.20	.47	.0[illegible]
C6	7	8.50	14.50	2.00	2.25	.20	.45	.0[illegible]
C7	6	7.00	12.00	1.89	2.14	.19	.44	.0[illegible]
C8	5	6.00	10.25	1.78	2.03	.18	.43	.05
C9	4	5.00	8.25	1.67	1.91	.17	.41	.07

Weights and Dimensions of Carnegie Z-Bars.

Section Index.	Thickness of Metal.	Size.			Weight.	
		Flange.	Web.	Flange.	Iron.	Ste[el]
Z 1	⅜	3 ½	6	3 ½	15.3	[illegible]
"	7-16	3 9-16	6 1-16	3 9-16	18.0	[illegible]
"	½	3 ⅝	6 ⅛	3 ⅝	20.6	[illegible]
Z 2	9-16	3 ½	6	3 ½	22.3	[illegible]
"	⅝	3 9-16	6 1-16	3 9-16	24.9	[illegible]
"	11-16	3 ⅝	6 ⅛	3 ⅝	27.5	[illegible]
Z 3	¾	3 ½	6	3 ½	28.8	[illegible]
"	13-16	3 9-16	6 1-16	3 9-16	31.3	[illegible]
"	⅞	3 ⅝	6 ⅛	3 ⅝	33.9	[illegible]
Z 4	5-16	3 ¼	5	3 ¼	11.3	
"	⅜	3 5-16	5 1-16	3 5-16	13.7	
"	7-16	3 ⅜	5 ⅛	3 ⅜	16.0	
Z 5	½	3 ¼	5	3 ¼	17.5	
"	9-16	3 5-16	5 1-16	3 5-16	19.8	
"	⅝	3 ⅜	5 ⅛	3 ⅜	22.1	
Z 6	11-16	3 ¼	5	3 ¼	23.2	
"	¾	3 5-16	5 1-16	3 5-16	25.5	
"	13-16	3 ⅜	5 ⅛	3 ⅜	27.8	
Z 7	¼	3 1-16	4	3 1-16	8.0	
"	5-11	3 ⅛	4 1-16	3 ⅛	10.1	
"	⅜	3 3-16	4 ⅛	3 3-16	12.2	
Z 8	7-16	3 1-16	4	3 1-16	13.5	
"	½	3 ⅛	4 1-16	3 ⅛	15.5	
"	9-16	3 3-16	4 ⅛	3 3-16	17.6	
Z 9	⅝	3 1-16	4	3 1-16	18.5	
"	11-16	3 ⅛	4 1-16	3 ⅛	20.5	
"	¾	3 3-16	4 ⅛	3 3-16	22.5	
Z10	¼	2 11-16	3	2 11-16	6.6	
"	5-16	2 ¾	3 1-16	2 ¾	8.3	
Z11	⅜	2 11-16	3	2 11-16	9.5	
"	7-16	2 ¾	3 1-16	2 ¾	11.2	
Z12	½	2 11-16	3	2 11-16	12.3	
"	[illegible]	2 ¾	3 1-16	2 ¾	13.9	

Pencoyd Steel Angles.

EVEN LEGS.

e ...es.	Approximate Weight in Pounds per Foot ... s Thicknesses in Inches. ⅛ .125	3-16 .1875	¼ .25	5-16 .3125	⅜ .375	7-16 .4375	½ .50	9-16 .5625	⅝ .625	11-16 .6875	¾ .75	⅞ .875	1 1.00
×6					14.8	17.3	19.9	22.3	24.9	26.5	29.1	34.2	39.3
×5					12.2	14.3	16.4	18.5	20.7	22.8	25.0	29.2	33.4
×4				8.2	9.8	11.3	13.0	14.6	16.1	17.7	19.3		
×3½				7.1	8.6	10.0	11.4	12.8	14.2				
×3			4.9	6.0	7.1	8.3	9.4	10.5	11.6				
×2¾			4.5	5.6	6.7	7.8	8.9						
×2½		3.1	4.1	5.1	6.1	7.1							
×2¼		2.7	3.6	4.5	5.4								
×2		2.44	3.3	4.1	4.9								
×1¾		2. 4	2.9	3.6	4.4								
×1½	1.16	1.80	2.4	3.0	3.6								
×1¼	1.02	1.53	2.04										
×1	0.82	1.16	1.53										

UNEVEN LEGS.

Size in Inches.	Approximate Weight in Pounds per Foot for Various Thicknesses in Inches. ⅛ .125	3-16 .1875	¼ .25	5-16 .3125	⅜ .375	7-16 .4375	½ .50	9-16 .5625	⅝ .625	11-16 .6875	¾ .75	⅞ .875	1 1.00
×3½							17.0	18.9	20.9	22.8	24.8	28.6	32.5
½×4					12.9	15.0	17.1	19.3	21.4	23.6	25.7	30.0	34.3
×4					12.2	14.4	16.4	18.6	20.7	22.8	24.9	29.1	33.3
×3½					11.5	13.6	15.6	17.6	19.7	21.7	23.8	27.8	31.9
½×3½					11.0	12.8	14.6	16.4	18.2				
×4					11.0	12.8	14.6	16.4	18.2	20.0	21.8		
×3½				8.7	10.3	12.0	13.6	15.2	16.8	18.5	20.1		
×3				8.2	9.7	11.2	12.8	14.3	15.8	17.3	18.9		
½×3				7.7	9.2	10.6	12.1	13.6	15.0	16.5	18.0		
×3½				7.7	9.2	10.6	12.1	13.6	15.0	16.5	18.0		
×3				7.1	8.6	10.0	11.4	12.8	14 2				
×3				6.6	7.9	9.2	10.5	11.8	13.1				
×2½			4.9	6.0	7.1	8.3	9.4						
×2			4.5	5.6	6.7								
×2½			4.5	5.6	6.7	7.8	8.9						
×2			4.1	5.1	6.1	7.1	8.2						
×2		2.7	3.6	4.5	5.4	6.3	7.2						
×1½		2.24	3.03	3.8	4.6								
×1¼		1.94	2.7	3.3	4.0								

Pencoyd Tees.

EVEN TEES.

Chart Number.	Size in Inches.	Weight per Foot. Iron.	Weight per Foot. Steel.
70	4 × 4	12.40	12.65
71	3½ × 3½	10.17	10.37
72	3 × 3	8.33	8.50
82	3 × 3	6.43	6.56
83	3 × 3	7.53	7.68
84	2½ × 2½	4.88	4.93
73	2½ × 2½	6.50	6.68
74	2½ × 2½	5.73	5.85
75	2¼ × 2¼	3.90	3.98
76	2¼ × 2¼	3.93	4.01
77	2 × 2	3.47	3.54
78	1¾ × 1¾	2.37	2.41
79	1½ × 1½	2.00	2.04
80	1¼ × 1¼	1.50	1.53
81	1 × 1	1.03	1.05
85	4 × 4	10.98	11.19

UNEVEN TEES.

Chart Number.	Size in Inches.	Weig F Iron.
107	5 × 4	14.70
106	5 × 3½	16.13
93	5 × 2 9-16	11.03
92	5 × 2½	10.23
90	4½ × 3½	14.83
109	4 × 4½	13.23
91	4 × 3½	13.98
94	4 × 3	8.63
95	4 × 3	8.37
96	4 × 2	6.43
97	3 × 3½	9.37
98	3 × 2½	7.93
110	3 × 2½	5.87
111	3 × 2½	6.87
117	3 × 2½	5.00
99	3 × 1½	3.73
105	2¾ × 2	7.13
104	2¾ × 1¾	6.53
100	2½ × 1¼	3.03
108	2¼ × 0 9-16	2.20
101	2 × 1½	2.90
112	2 × 1 1-16	2.07
102	2 × 1	2.33
103	2 × 9-16	2.03
116	1¾ × 1¼	3.47
113	1¾ × 1 1-16	1.87
114	1½ × 15-16	1.87
115	1¼ × 15-16	1.13
118	3 × 2½	5.9
119	2¾ × 2½	5.6

Pencoyd Car-Builders' Channels, Iron.

Section Number.	Depth in Inches.	Minimum Flange Width in Inches.	Minimum Web Thickness in Inches.	Minimum Weight per Foot in Pounds.	Approximate Weight in Pounds pe Foot for Each Thickness of Web, in Inches. 5-16	⅜	7-16	½	9-16	⅝
55	13	3⅞	⅝	29.5		29.5	32.2	34.9	37.6	40.
54	12	3	9-32	22.4	23.6	26.1	28.6	31.1	33.6	
33½	10½	{ 3⅝	7-16	23.6			23.6	25.8		
33	10½	} 2½	5-16	17.6	17.6	19.8				

Pencoyd Car-Builders' Channels, Steel.

Section Number.	Depth in Inches.	Minimum Flange Width in Inches.	Minimum Web Thickness in Inches.	Minimum Weight per Foot in Pounds.	5-16	⅜	7-16	½	9-16	⅝
55	13	3⅞	⅜	30.1		30.1	32.9	35.6	38.4	41.
54	12	3	9-32	22.8	24.1	26.6	29.2	31.7	34.3	
33½	10½	{ 3⅝	7-16	24.1			24.1	26.3		
33	[illegible]	[illegible]	5-16	17.9	17.9	20.2				

D WEIGHTS OF ROOFING MATERIALS.

rrugated Iron (Phœnix Iron Co.).

	ACK IRON.		GALVANIZED IRON.		
ht bs. t.,	Weight in Lbs. per Sq. Ft. on Roof. Flat.	Weight in Lbs. per Sq. Ft., on Roof. Corrugated	Weight in Lbs. per Sq. Ft., Flat.	Weight in Lbs. per Sq. Ft. on Roof. Flat.	Weight in Lbs. per Sq. Ft., on Roof. Corrugated
1	3.03	3.37	3.00	3.50	3.88
7	2.29	2.54	2.37	2.76	3.07
)	1.63	1.82	1.75	2.03	2.26
2	1.31	1.45	1.31	1.53	1.71
3	1.03	1.14	1.06	1.24	1.37
2	0.84	0.93	0.94	1.09	1.21

ble is calculated for the ordinary size of sheet, which is from le, and from 6 to 8 feet long, allowing 4 inches lap in length 1 width of sheet.

ng of sheet iron adds about one-third of a pound to its weight

iron made by the Keystone Bridge Co., the corrugations are sured on the straight line; they require a length of iron of one corrugation, and the depth of corrugation is 21-32″. is allowed for lap in the width of the sheet and 6″ in the usual pitch of roof of two to one. Sheets can be corrugated not exceeding ten feet. The most advantageous width is lowing ½″ for irregularities) will make eleven corrugations ng allowance for laps, will cover 24¼″ of the surface of the

it was found that corrugated iron No. 20, spanning 6 feet, e a permanent deflection for a load of 30 lbs. per square foot, collapse with a load of 60 lbs. per square foot. The distance of purlins should therefore not exceed 6 feet, and, prefer- n this.

Terra-Cotta.

cotta roofing 3″ thick weighs 16 lbs. per square foot and 2″ r square foot.
of the same material 2″ thick weighs 11 lbs. per square foot.

Tiles.

″ × 10½″ × ⅝″ weigh from 1480 to 1850 lbs. per square of ng one-half the length of the tile.
oves and fillets weigh from 740 to 925 lbs. per square of roof.
″ × 10½″ laid 10″ to the weather, weigh 850 lbs. per square.

Tin.

es for roofing tin are 14″ × 20″ and 20″ × 28″. Without p or waste, tin roofing weighs from 50 to 62 lbs. per square. f weighs from 62 to 75 lbs. per square.
s or terne plates (steel plates coated with an alloy of tin ade only in IC and IX thicknesses (27 and 29 Birmingham e" and "charcoal" tin plates, old names used when iron and charcoal was used for the tinned plate, are still used in ugh steel plates have been substituted for iron; a coke plate meaning one made of Bessemer steel, and a charcoal plate rth steel. The thickness of the tin coating on the plates rent "*brands.*"
nformation on Tin Roofing, see circulars of Merch

TIN PLATES. (TINNED SHEET STEEL.)

Standard Stock Sizes, with Number of Sheets and Net Weight per Box.

B. W. Gauge.	Thickness.	Size.	Sheets.	Net Weight lbs.
29	IC	10×14	225	108
27	IX	10×14	225	135
26	IXX	10×14	225	160
29	IC	12×12	225	110
27	IX	12×12	225	138
26	IXX	12×12	225	165
29	IC	14×20	112	108
27	IX	14×20	112	135
26	IXX	14×20	112	160
25	IXXX	14×20	112	180
24½	IXXXX	14×20	112	200
29	IC	20×28	112	216
27	IX	20×28	112	270
26	IXX	20×28	112	320
25	IXXX	20×28	56	180
24½	IXXXX	20×28	56	200
29	IC	13×13	225	132
27	IX	13×13	225	162
26	IXX	13×13	225	192
29	IC	14×14	225	155
27	IX	14×14	225	193
26	IXX	14×14	225	230
29	IC	15×15	225	178
27	IX	15×15	225	218
26	IXX	15×15	225	260
29	IC	16×16	225	200
27	IX	16×16	225	248
26	IXX	16×16	225	290
29	IC	17×17	225	230
27	IX	17×17	225	289
26	IXX	17×17	225	340
29	IC	18×18	112	138
27	IX	18×18	112	158
26	IXX	18×18	112	178
29	IC	20×20	112	160
27	IX	20×20	112	195
26	IXX	20×20	112	222
29	IC	22×22	112	190
27	IX	22×22	112	235
26	IXX	22×22	112	275
29	IC	24×24	112	220
27	IX	24×24	112	276
26	IXX	24×24	112	330
29	IC	10×20	225	[illegible]
27	IX	10×20	225	[illegible]
26	IXX	10×20	225	[illegible]
29	IC	11×22	225	[illegible]
27	IX	11×22	225	[illegible]
26	IXX	11×22	225	[illegible]
29	IC	12×24	112	[illegible]
27	IX	12×24	112	[illegible]
26	IXX	12×24	112	[illegible]
29	IC	13×26	112	[illegible]
27	IX	13×26	112	[illegible]
26	IXX	13×26	112	[illegible]
29	IC	14×22	112	[illegible]
27	IX	14×22	112	[illegible]
26	IXX	14×22	112	[illegible]
29	IC	14×24	112	[illegible]
27	IX	14×24	112	[illegible]
26	IXX	14×24	112	[illegible]
29	IC	14×28	112	[illegible]
27	IX	14×28	112	[illegible]
26	IXX	14×28	112	[illegible]
29	IC	14×31	112	[illegible]
27	IX	14×31	112	[illegible]
26	IXX	14×31	112	[illegible]
27	IX	14×56	56	[illegible]
26	IXX	14×56	56	[illegible]
27	IX	14×60	56	[illegible]
26	IXX	14×60	56	[illegible]
29	IC	15×21	112	[illegible]
27	IX	15×21	112	[illegible]
26	IXX	15×21	112	[illegible]
29	IC	16×19	112	[illegible]
27	IX	16×19	112	[illegible]
26	IXX	16×19	112	[illegible]
29	IC	16×20	112	[illegible]
27	IX	16×20	112	[illegible]
26	IXX	16×20	112	[illegible]
29	IC	16×22	112	[illegible]
27	IX	16×22	112	[illegible]
26	IXX	16×22	112	[illegible]

B. W. Gauge.	Thickness.	Size.	Sheets.	Net Weight lbs.
28	DC	12½×17	100	94
25	DX	12½×17	100	122
24	DXX	12½×17	100	143
23	DXXX	12½×17	100	164
22	DXXXX	12½×17	100	185
28	DC	15×21	100	130
25	DX	15×21	100	180
24	DXX	15×21	100	213
23	DXXX	15×21	100	[illegible]
22	DXXXX	15×21	100	[illegible]
28	DC	17×25	50	[illegible]
25	DX	17×25	50	[illegible]
24	DXX	17×25	50	[illegible]
23	DXXX	17×25	50	[illegible]
22	DXXXX	17×25	50	[illegible]

Terne Plates, 112 sheets in a box: 10×20, IC, 80 lbs., IX 100 lbs. per box.
14×20, IC, 112 lbs., IX 140 " " "
20×28, IC, 224 lbs., IX 280 " " "

* Tin and Iron, 36 and 38 B. W. G., 10×14 and 14×20. 112 lbs. per

Slate.

Number and superficial area of slate required for one square of roof.
(1 square = 100 square feet.)

Dimensions in Inches.	Number per Square.	Superficial Area in Sq. Ft.	Dimensions in Inches.	Number per Square.	Superficial Area in Sq. Ft.
6×12	533	267	12×18	160	240
7×12	457	....	10×20	169	235
8×12	400	....	11×20	154	
9×12	355	....	12×20	141	
7×14	374	254	14×20	121	
8×14	327	....	16×20	137	
9×14	291	....	12×22	126	231
10×14	261	....	14×22	108	
8×16	277	246	12×24	114	228
9×16	246	....	14×24	98	
10×16	221	....	16×24	86	
9×18	213	240	14×26	89	225
10×18	192	....	16×26	78	

As slate is usually laid, the number of square feet of roof covered by one slate can be obtained from the following formula :

$$\frac{\text{width} \times (\text{length} - 3 \text{ inches})}{288} = \text{the number of square feet of roof covered.}$$

Weight of slate of various lengths and thicknesses required for one square of roof :

Length in Inches.	Weight in Pounds per Square for the Thickness.							
	⅛″	3-16″	¼″	⅜″	½″	⅝″	¾″	1″
12	483	724	967	1450	1936	2419	2902	3872
14	460	688	920	1379	1842	2301	2760	3683
16	445	667	890	1336	1784	2229	2670	3567
18	434	650	869	1303	1740	2174	2607	3480
20	425	637	·851	1276	1704	2129	2553	3408
22	418	626	836	1254	1675	2093	2508	3350
24	412	617	825	1238	1653	2066	2478	3306
26	407	610	815	1222	1631	2039	2445	3263

The weights given above are based on the number of slate required for one square of roof, taking the weight of a cubic foot of slate at 175 pounds.

Pine Shingles.

Number and weight of pine shingles required to cover one square of roof :

Number of Inches Exposed to Weather.	Number of Shingles per Square of Roof.	Weight in Pounds of Shingle on One-square of Roofs.	Remarks.
4	900	216	The number of shingles per square is for common gable-roofs. For hip-roofs add five per cent. to these figures.
4½	800	192	*The weights per square are based on the number per square.*
5	720	173	
5½	655	157	
6	*600*	*144*	

Skylight Glass.

The weights of various sizes and thicknesses of fluted or rough plate-g required for one square of roof.

Dimensions in Inches.	Thickness in Inches.	Area in Square Feet.	Weight in Lbs. Square of Ro
12 × 48	3-16	3.997	250
15 × 60	¼	6.246	350
20 × 100	⅜	13.880	500
94 × 156	½	101.768	700

In the above table no allowance is made for lap.

If ordinary window-glass is used, single thick glass (about 1-16″) will w about 82 lbs. per square, and double thick glass (about ⅛″) will weigh a 164 lbs. per square, *no allowance being made for lap.* A box of ordi window-glass contains as nearly 50 square feet as the size of the panes admit of. Panes of any size are made to order by the manufacturers, b great variety of sizes are usually kept in stock, ranging from 6 × 8 inche 36 × 60 inches.

APPROXIMATE WEIGHTS OF VARIOUS ROOF COVERINGS.

For preliminary estimates the weights of various roof coverings ma taken as tabulated below:

Name.	Weight in Lbs. per Square of Roof.
Cast-iron plates (⅜″ thick)	1500
Copper	80- 125
Felt and asphalt	100
Felt and gravel	800-1000
Iron, corrugated	100- 375
Iron, galvanized, flat	100- 350
Lath and plaster	900-1000
Sheathing, pine, 1″ thick yellow, northern	300
" " " " " southern	400
Spruce, 1″ thick	200
Sheathing, chestnut or maple, 1″ thick	400
" ash, hickory, or oak, 1″ thick	500
Sheet iron (1-16″ thick)	300
" " " " and laths	500
Shingles, pine	200
Slates (¼″ thick)	900
Skylights (glass 3-16″ to ½″ thick)	250- 700
Sheet lead	500- 800
Thatch	650
Tin	70- 125
Tiles, flat	1500-2000
" (grooves and fillets)	700-1000
" pan	1000
" with mortar	2000-3000
Zinc	100- 200

WEIGHT OF CAST-IRON PIPES OR COLUMNS.

In Lbs. per Lineal Foot.

Cast iron = 450 lbs. per cubic foot.

Thick. of Metal.	Weight per Foot.	Bore.	Thick. of Metal.	Weight per Foot.	Bore.	Thick. of Metal.	Weight per Foot.
Ins.	Lbs.	Ins.	Ins.	Lbs.	Ins.	Ins.	Lbs.
3/8	12.4	10	3/4	79.2	22	3/4	167.5
1/2	17.2	10½	1/2	54.0		7/8	196.5
5/8	22.2		5/8	68 2	23	3/4	174.9
3/8	14.3		3/4	82.8		7/8	205.1
1/2	19.6	11	1/2	56.5		1	235.6
5/8	25.3		5/8	71.3	24	3/4	182.2
3/8	16.1		3/4	86.5		7/8	213.7
1/2	22.1	11½	1/2	58.9		1	245.4
5/8	28.4		5/8	74.4	25	3/4	189.6
3/8	17.9		3/4	90.2		7/8	222.3
1/2	24.5	12	1/2	61.3		1	255.3
5/8	31.5		5/8	77.5	26	3/4	197.0
3/8	19.8		3/4	93.9		7/8	230.9
1/2	27.0	12½	1/2	63.8		1	265.1
5/8	34.4		5/8	80.5	27	3/4	204.3
3/8	21.6		3/4	97.6		7/8	239.4
1/2	29.4	13	1/2	66.3		1	274.9
5/8	37.6		5/8	83.6	28	3/4	211.7
3/8	23.5		3/4	101.2		7/8	248.1
1/2	31.8	14	1/2	71.2		1	284.7
5/8	40.7		5/8	89.7	29	3/4	219.1
3/8	25.3		3/4	108.6		7/8	256.6
1/2	34.4	15	5/8	95.9		1	294.5
5/8	43.7		3/4	116.0	30	7/8	265.2
3/8	27.1		7/8	136.4		1	304.3
1/2	36.8	16	5/8	102.0		1⅛	343 7
5/8	46.8		3/4	123.3	31	7/8	273.8
3/8	29.0		7/8	145.0		1	314.2
1/2	39.3	17	5/8	108.2		1⅛	354.8
5/8	49.9		3/4	130.7	32	7/8	282 4
3/8	30.8		7/8	153.6		1	324.0
1/2	41.7	18	5/8	114.3		1⅛	365.8
5/8	52.9		3/4	138.1	33	7/8	291.0
1/2	44.2		7/8	162.1		1	333.8
5/8	56.0	19	5/8	120.4		1⅛	376.9
3/4	68 1		3/4	145.4	34	7/8	299.6
1/2	46.6		7/8	170.7		1	343.7
5/8	59.1	20	5/8	126.6		1⅛	388.0
3/4	71.8		3/4	152.8	35	7/8	308.1
1/2	49.1		7/8	179.3		1	353.4
5/8	62.1	21	5/8	132 7		1⅛	399.0
3/4	75.5		3/4	160.1	36	7/8	316 6
1/2	51.5		7/8	187.9		1	363 1
5/8	65.2	22	5/8	138.8		1⅛	410.0

he weight of the two flanges may be reckoned = weight of one foot.

WEIGHTS OF CAST-IRON PIPE TO LAY 12 FI LENGTH.

Weights are Gross Weights, including Hub.

(Calculated by F. H. Lewis.)

Thickness.		Inside Diameter.							
Inches.	Equiv. Decimals.	4″	6″	8″	10″	12″	14″	16″	18″
3/8	.375	209	304	400					
13-32	.40625	228	331	435					
7-16	.4375	247	358	470	581				
15-32	.4687	266	386	505	624	692	804		
1/2	.5	286	414	541	668	744	863		
17-32	.53125	306	442	577	712	795	922	1050	117
9-16	.5625	327	470	613	756	846	983	1118	125
19-32	.59375	......	498	649	801	899	1043	1186	132
5/8	.625	......	......	686	845	951	1103	1254	140
11-16	.6875	......	......	......	935	1003	1163	1322	148
3/4	.75	......	......	......	1026	1110	1285	1460	164
13-16	.8125	......	......	......	......	1216	1408	1598	178
7/8	.875	......	......	......	......	1324	1531	1738	194
15-16	.9375	......	......	......	......	1432	1656	1879	210
1	1.	......	......	......	......	......	1783	2021	225
1 1/8	1.125	......	......	......	......	......	1909	2163	241
1 1/4	1.25	......	......	......	......	......	......	......	273
1 3/8	1.375	......	......	......	......	......	......	......	306

Thickness.		Inside Diameter.							
Inches.	Equiv. Decimals.	22″	24″	27″	30″	33″	36″	42″	48″
5/8	.625	1799							
11-16	.6875	1985	2160	2422					
3/4	.75	2171	2362	2648	2934	3221	3507		
13-16	.8125	2359	2565	2875	3186	3496	3806	4426	
7/8	.875	2547	2769	3103	3437	3771	4105	4773	544
15-16	.9375	2737	2975	3332	3690	4048	4406	5122	583
1	1.	2927	3180	3562	3942	4325	4708	5472	623
1 1/8	1.125	3310	3598	4027	4456	4886	5316	6176	703
1 1/4	1.25	3698	4016	4492	4970	5447	5924	6880	782
1 3/8	1.375	......	4439	4964	5491	6015	6540	7591	864
1 1/2	1.5	......	......	5439	6012	6584	7158	8303	944
1 5/8	1.625	......	......	......	6539	7159	7782	9022	1026
1 3/4	1.75	......	......	......	......	7737	8405	9742	1107
1 7/8	1.875	......	......	......	......	......	......	10468	1189
2	2.	......	......	......	......	......	......	11197	1272
2 1/4	2.25	......	......	......	......	......	......	......	1438
2 1/2	2.5	......	......	......	......	......	......	......	......
2 3/4	2.75	......	......	......	......	......	......	......	......

CAST-IRON PIPE FITTINGS.

Approximate Weight.

Addyston Pipe and Steel Co., Cincinnati, Ohio.

CROSSES.

ze in hes.	Weight in Lbs.
	40
	104
2	90
	150
3	114
2	110
	200
4	150
3	150
	325
6	265
4	265
3	225
	510
8	415
6	388
4	338
3	350
	700
10	650
8	615
6	540
4	525
3	495
10	750
8	635
6	570
	1025
14	1070
12	1025
10	1010
8	825
6	700
4	650
	1790
12	1370
10	1225
	1000
	1000
	1000
	2190
0	2020
	1340
0	2635
2	2250
	1995

TEES.

ze in hes.	Weight in Lbs.
	28
	76
	76
	100
	90
	87
	150
	130
	125
	120
	266
	252
	222

TEES.

Size in Inches.	Weight in Lbs.
8 × 3	220
10	390
10 × 8	330
10 × 6	312
10 × 4	292
10 × 3	290
12	565
12 × 10	510
12 × 8	492
12 × 6	484
12 × 4	460
14 × 12	650
14 × 10	650
14 × 8	575
14 × 6	545
14 × 4	525
14 × 3	490
16	790
16 × 14	850
16 × 12	825
16 × 10	890
16 × 8	755
16 × 6	630
16 × 4	655
20	1375
20 × 16	1115
20 × 12	1025
20 × 10	1090
20 × 8	900
20 × 6	875
20 × 4	845
21 × 10	1465
24	1875
24 × 12	1425
24 × 8	1375
24 × 6	1375
30	3025
30 × 24	2640
30 × 20	2200
30 × 12	2035
30 × 10	2050
30 × 6	1825
36	5140
36 × 30	4200
36 × 12	4050

45° BRANCH PIPES.

Size in Inches.	Weight in Lbs.
3	90
6 × 6 × 4	145
8	300
8 × 6	290
24	2765
24 × 24 × 20	2145
30	*4170*
36	*10300*

SLEEVES.

Size in Inches.	Weight in Lbs.
2	*10*
3	*20*
4	*44*

SLEEVES.

Size in Inches.	Weight in Lbs.
6	65
8	86
10	140
12	176
14	208
16	340
20	500
24	710
30	965
36	1500

90° ELBOWS.

Size in Inches.	Weight in Lbs.
2	14
3	34
4	48
6	110
8	145
10	225
12	370
14	450
16	525
20	900
24	1400

⅛ or 45° BENDS.

Size in Inches.	Weight in Lbs.
3	30
4	65
6	85
8	160
10	190
12	290
16	510
20	740
24	1425
30	2000

1-16 or 22½° BENDS.

Size in Inches.	Weight in Lbs.
6	150
8	155
10	165
12	260
16	500
24	1280
30	1735

REDUCERS.

Size in Inches.	Weight in Lbs.
3 × 2	35
4 × 3	42
4 × 2	40
6 × 4	95
6 × 3	80
8 × 6	*126*
8 × 4	*116*
8 × 3	*116*
10 × 8	*212*
10 × 6	150

REDUCERS.

Size in Inches.	Weight in Lbs.
10 × 4	128
12 × 10	278
12 × 8	254
12 × 6	250
12 × 4	250
14 × 12	475
14 × 10	430
14 × 8	340
14 × 6	285
16 × 12	475
16 × 10	435
20 × 16	690
20 × 14	575
20 × 12	540
20 × 8	300
24 × 20	745
30 × 24	1305
30 × 18	1385
36 × 30	1730

ANGLE REDUCERS FOR GAS.

Size in Inches.	Weight in Lbs.
6 × 4	95
6 × 3	80

S PIPES.

Size in Inches.	Weight in Lbs.
4	90
6	190

PLUGS.

Size in Inches.	Weight in Lbs.
2	2
3	5
4	8
6	12
8	26
10	46
12	66
14	70
16	100
20	150
24	185
30	370

CAPS.

Size in Inches.	Weight in Lbs.
3	15
4	25
6	60
8	75
10	100
12	120

DRIP BOXES.

Size in Inches.	Weight in Lbs.
4	235
8	355
10	760
20	1420

WEIGHTS OF CAST-IRON WATER- AND GAS-PIP

(Addyston Pipe and Steel Co., Cincinnati, Ohio.)

Size in Inches.	Standard Water-Pipe.			Size in Inches.	Standard Gas-Pipe.		
	Per Foot.	Thick-ness.	Per Length.		Per Foot.	Thick-ness.	Pe Len
2	7	5-16	63	2	6	¼	4
3	15	⅜	180	3	12½	5-16	1
3	17	½	204				
4	22	½	264	4	17	⅜	2
6	33	½	396	6	30	7-16	3
8	42	½	504	8	40	7-16	4
10	60	9-16	720	10	50	7-16	6
12	75	9-16	900	12	70	½	8
14	117	¾	1400	14	84	9-16	10
16	125	¾	1500	16	100	9-16	12
18	167	⅞	2000	18	134	11-16	16
20	200	15-16	2400	20	150	11-16	1
24	250	1	3000	24	184	¾	2
30	350	1⅛	4200	30	250	¾	3
36	475	1⅜	5700	36	350	⅞	4
42	600	1⅜	7200	42	383	⅞	4
48	775	1½	9300	48	542	1⅛	6
60	1330	2	15960	60	900	1⅜	10

THICKNESS OF CAST-IRON PIPES.

P. H. Baermann, in a paper read before the Engineers' Club of delphia in 1882, gave twenty different formulas for determining the ness of cast-iron water-pipes under pressure. The formulas are of classes:

1. Depending upon the diameter only.
2. Those depending upon the diameter and head, and which add stant.
3. Those depending upon the diameter and head, contain an addit subtractive term depending upon the diameter, and add a constant.

The more modern formulas are of the third class, and are as follow

$t = .00008hd + .01d + .36$ .... Shedd, No. 1.
$t = .00006hd + .0133d + .296$ .... Warren Foundry, No. 2.
$t = .000058hd + .0152d + .312$ .... Francis, No. 3.
$t = .000048hd + .013d + .32$ .... Dupuit, No. 4.
$t = .00004hd + .1\sqrt{d} + .15$ .... Box, No. 5.
$t = .000135hd + .4 - .0011d$ .... Whitman, No. 6.
$t = .00006(h + 230)d + .333 - .0033d$ ...... Fanning, No. 7.
$t = .00015hd + .25 - .0052d$ .... Meggs, No. 8.

In which t = thickness in inches, h = head in feet, d = diameter in in

Rankine, "Civil Engineering," p. 721, says: "Cast-iron pipes sho made of a soft and tough quality of iron. Great attention should be to moulding them correctly, so that the thickness may be exactly unifo round. Each pipe should be tested for air-bubbles and flaws by rin with a hammer, and for strength by exposing it to double the int greatest working pressure." The rule for computing the thickness of to resist a given working pressure is $t = \frac{rp}{f}$, where r is the radius in i p the pressure in pounds per square inch, and f the tenacity of the ir square inch. When $f = 18000$, and a factor of safety of 5 is used, the expressed in terms of d and h becomes

$$t = \frac{.5d\,.433h}{3600} = \frac{dh}{16628} = .00006dh.$$

"There ar[illegible] ns, however, arising from difficulties in castin be the st[illegible] by shocks, which cause the thickness to be [illegible] the above formula."

Thickness of Metal and Weight per Length for Different Sizes of Cast-Iron Pipes under Various Heads of Water.

(Warren Foundry and Machine Co.)

Size.	50 Ft. Head.		100 Ft. Head.		150 Ft. Head.		200 Ft. Head.		250 Ft. Head.		300 Ft. Head.	
	Thickness of Metal.	Weight per Length.	Thickness of Metal.	Weight per Length.	Thickness of Metal.	Weight per Length.	Thickness of Metal.	Weight per Length.	Thickness of Metal.	Weight per Length.	Thickness of Metal.	Weight per Length.
3	.344	144	.353	149	.362	153	.371	157	.380	161	.390	166
4	.361	197	.373	204	.385	211	.397	218	.409	226	.421	235
5	.378	254	.393	265	.408	275	.423	286	.438	298	.453	309
6	.393	315	.411	330	.429	345	.447	361	.465	377	.483	393
8	.422	445	.450	475	.474	502	498	529	.522	557	.546	584
10	.459	600	.489	641	.519	682	.549	723	.579	766	.609	808
12	.491	768	.527	826	.563	885	.599	944	.635	1004	.671	1064
14	.524	952	.566	1031	.608	1111	.650	1191	.692	1272	.734	1352
16	.557	1152	.604	1253	.652	1360	.700	1463	.748	1568	.796	1673
18	.589	1370	.643	1500	.697	1630	.751	1761	.805	1894	.859	2026
20	.622	1603	.682	1763	.742	1924	.802	2086	.862	2248	.922	2412
24	.687	2120	.759	2349	.831	2580	.903	2811	.975	3045	1.047	3279
30	.785	3020	.875	3376	.965	3735	1.055	4095	1.145	4458	1.235	4822
36	.882	4070	.990	4581	1.098	5096	1.206	5613	1.314	6133	1.422	6656
42	.980	5265	1.106	5958	1.232	6657	1.358	7360	1.484	8070	1.610	8804
48	1.078	6616	1.222	7521	1.366	8431	1.510	9340	1.654	10269	1.798	11195

All pipe cast vertically in dry sand; the 3 to 12 inch in lengths of 12 feet, all larger sizes in lengths of 12 feet 4 inches.

Safe Pressures and Equivalent Heads of Water for Cast-Iron Pipe of Different Sizes and Thicknesses.

(Calculated by F. H. Lewis, from Fanning's Formula.)

Thickness.	Size of Pipe.																	
	4″		6″		8″		10″		12″		14″		16″		18″		20″	
	Pressure in Pounds.	Head in Feet.	Pressure in Pounds.	Head in Feet.	Pressure in Pounds.	Head in Feet.	Pressure in Pounds.	Head in Feet.	Pressure in Pounds.	Head in Feet.	Pressure in Pounds.	Head in Feet.	Pressure in Pounds.	Head in Feet.	Pressure in Pounds.	Head in Feet.	Pressure in Pounds.	Head in Feet.
7-16	112	258	49	112	18	42												
1-2	224	516	124	286	74	171	44	101	24	55								
9-16	336	774	199	458	130	300	89	205	62	143	42	97						
5-8	....	....	274	631	186	429	132	304	99	228	74	170	56	129	41	95		
11-16	....	....	....	....	....	....	177	408	137	316	106	244	84	194	66	152	51	118
3-4	....	....	....	....	....	....	224	516	174	401	138	316	112	258	91	210	74	170
13-16	....	....	....	....	....	....	....	....	212	488	170	392	140	323	116	267	96	221
7-8	....	....	....	....	....	....	....	....	249	574	202	465	168	387	141	325	119	274
15-16	....	....	....	....	....	....	....	....	....	....	234	538	196	452	166	382	141	325
1	....	....	....	....	....	....	....	....	....	....	266	612	224	516	191	440	164	378
1 1-8	....	....	....	....	....	....	....	....	....	....	....	....	....	....	216	497	209	481
1 1-4	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	256	589

Safe Pressures, etc., for Cast-Iron Pipe.—(*Continued.*)

Thickness.	22″ Pressure in Pounds.	22″ Head in Feet.	24″ Pressure in Pounds.	24″ Head in Feet.	27″ Pressure in Pounds.	27″ Head in Feet.	30″ Pressure in Pounds.	30″ Head in Feet.	33″ Pressure in Pounds.	33″ Head in Feet.	36″ Pressure in Pounds.	36″ Head in Feet.	42″ Pressure in Pounds.	42″ Head in Feet.	48″ Pressure in Pounds.	48″ Head in Feet.
11-16	40	92	30	69	19	64										
3-4	60	138	49	113	36	83	24	55								
13-16	80	184	68	157	52	120	39	90								
7-8	101	233	86	198	69	159	54	124	42	97	32	74				
15-16	121	279	105	242	85	196	69	159	55	127	44	101				
1	142	327	124	286	102	235	84	194	69	159	57	131	38	88	24	55
1 1-8	182	419	161	371	135	311	114	263	96	221	82	189	59	136	43	99
1 1-4	224	516	199	458	169	389	144	332	124	286	107	247	81	187	62	143
1 3-8	....	....	237	546	202	465	174	401	151	348	132	304	103	237	81	187
1 1-2	....	....	....	....	236	544	204	470	178	410	157	362	124	286	99	228
1 5-8	....	....	....	....	....	....	234	538	205	472	182	419	145	334	118	272
1 3-4	....	....	....	....	....	....	....	....	233	537	207	477	167	385	136	313
1 7-8	....	....	....	....	....	....	....	....	....	....	....	....	188	433	155	357
2	....	....	....	....	....	....	....	....	....	....	....	....	210	484	174	401
2 1-8	....	....	....	....	....	....	....	....	....	....	....	....	....	....	193	445
2 1-4	....	....	....	....	....	....	....	....	....	....	....	....	....	....	212	488
2 1-2	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
2 3-4	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....

NOTE.—The absolute safe static pressure which may be put upon pipe is given by the formula $P = \frac{2T}{D} \times \frac{S}{5}$, in which formula P is the pressure per square inch; T, the thickness of the shell; S, the ultimate strength per square inch of the metal in tension; and D, the inside diameter of the pipe. In the tables S is taken as 18000 pounds per square inch, with a working strain of one fifth this amount or 3600 pounds per square inch. The formula for the absolute safe static pressure then is: $P = \frac{7200T}{D}$.

It is, however, usual to allow for "water-ram" by increasing the thickness enough to provide for 100 pounds additional static pressure, and, to insure sufficient metal for good casting and for wear and tear, a further increase equal to $.333\left(1 - \frac{D}{100}\right)$.

The expression for the thickness then becomes:

$$T = \frac{(P+100)D}{7200} + .333\left(1 - \frac{D}{100}\right),$$

and for safe working pressure

$$P = \frac{7200}{D}\left(T - .333\left(1 - \frac{D}{100}\right)\right) - 100.$$

The additional section provided as above represents an increased value under static pressure for the different sizes of pipe as follows (see table in margin). So that to test the pipes up to one fifth of the ultimate strength of the material, the pressures in the marginal table should be added to the pressure-values given in the table above.

Size of Pipe.
4″
6
8
10
12
14
16
18
20
22
24
27
30
33
36
42
48
60

SHEET-IRON HYDRAULIC PIPE.

(Pelton Water-Wheel Co.)

it per foot, with safe head for various sizes of double-riveted pipe.

Area of Pipe.	Thickness of Iron by Wire Gauge.	Safe Head in Feet the Pipe will stand.	Weight of Pipe per Lineal Ft.	Diameter of Pipe.	Area of Pipe.	Thickness of Iron by Wire Gauge.	Safe Head in Feet the Pipe will stand.	Weight of Pipe per Lineal Ft.
q.in.	B.W.G.	feet.	lbs.	in.	sq.in.	B.G.W.	feet.	lbs.
7	18	400	2	18	254	16	165	16½
12	18	350	2¼	18	254	14	252	20½
12	16	525	3	18	254	12	385	27¼
20	18	325	3½	18	254	11	424	30
20	16	500	4¼	18	254	10	505	34
20	14	675	5	20	314	16	148	18
28	18	296	4¼	20	314	14	227	22½
28	16	487	5¾	20	314	12	346	30
28	14	743	7½	20	314	11	380	32½
38	18	254	5¼	20	314	10	456	36½
38	16	419	6¾	22	380	16	135	20
38	14	640	8½	22	380	14	206	24¾
50	16	367	7½	22	380	12	316	32¾
50	14	560	9½	22	380	11	347	35¾
50	12	854	13	22	380	10	415	40
63	16	327	8½	24	452	14	188	27¼
63	14	499	10¾	24	452	12	290	35½
63	12	761	14¼	24	452	11	318	39
78	16	295	9¼	24	452	10	379	43½
78	14	450	11¾	24	452	8	466	53
78	12	687	15¾	26	530	14	175	29¼
78	11	754	17½	26	530	12	267	38½
78	10	900	19¼	26	530	11	294	42
95	16	269	9¾	26	530	10	352	47
95	14	412	13	26	530	8	432	57¼
95	12	626	17¼	28	615	14	162	31¼
95	11	687	18¾	28	615	12	247	41¼
95	10	820	21	28	615	11	273	45
113	16	246	11¼	28	615	10	327	50¼
113	14	377	14	28	615	8	400	61¼
113	12	574	18½	30	706	12	231	44
113	11	630	19¾	30	706	11	254	48
113	10	753	22¾	30	706	10	304	54
132	16	228	12	30	706	8	375	65
132	14	348	15	30	706	7		74
132	12	530	20	36	1017	11		58
132	11	583	22	36	1017	10		67
132	10	696	24½	36	1017	8		78
153	16	211	13	36	1017	7		88
153	14	324	16	40	1256	10		71
153	12	494	21½	40	1256	8		86
153	11	543	23½	40	1256	7		97
153	10	648	26	40	1256	6		108
176	16	197	13¾	40	1256	4		126
176	14	302	17	42	1385	10		74½
176	12	460	23	42	1385	8		91
176	11	507	24½	42	1385	7		102
176	10	606	28	42	1385	6		114
201	16	185	14½	42	1385	4		133
201	14	283	17¼	42	1385	¼		137
201	12	432	24¼	42	1385	3		145
201	11	474	26½	42	1385	5-16		
201	10	567	29½	42	1385	⅜		

STANDARD PIPE FLANGES.

Adopted July 18, 1894, at a conference of committees of the Society of Mechanical Engineers, and the Master Steam and Hot ters' Association, with representatives of leading manufacturers of pipe.

The list is divided into two groups; for medium and high pres first ranging up to 75 lbs. per square inch, and the second up to 20

Pipe size, inches.	Pipe Thickness, $\frac{P+100}{.4s}d+.333\left(1-\frac{d}{100}\right)$	Thickness, nearest Fraction, inches	Stress on Pipe per square inch @ 200 lbs.	Radius of Fillet, inches.	Flange Diameters, inches.		Flange Thickness, inches.		Width Flange Face, inches.		Bolt Circle Diameter, inches.		Number of Bolts.	Bolt Diameter, inches.
2	.409	7/16	460	1/8		6		5/8		2		4¾	4	5/8
2½	.429	7/16	550	1/8		7		11/16		2⅛		5¼	4	5/8
3	.448	7/16	690	1/8		7½		¾		2¼		6	4	5/8
3½	.466	½	700	1/8		8½		13/16		2¼		6½	4	5/8
4	.486	½	800	1/8		9		15/16		2⅜		7¼	4	¾
4½	.498	½	900	1/8		9¼		15/16		2⅜		7¾	8	¾
5	.525	½	1000	1/8		10		15/16		2½		8½	8	¾
6	.563	9/16	1060	1/8		11		1		2⅝		9½	8	¾
7	.60	⅝	1120	1/8		12½		1 1/16		2⅝		10¾	8	¾
8	.639	⅝	1280	1/8		13½		1⅛		2¾		11¾	8	¾
9	.678	11/16	1310	3/16		15		1⅛		2⅞		13	12	¾
10	.713	¾	1330	3/16		16		1 3/16		3		14¼	12	⅞
12	.79	13/16	1470	3/16		19		1¼		3¼		16½	12	⅞
14	.864	⅞	1600	3/16		21		1⅜		3½		18¾	12	1
15	.904	15/16	1600	3/16		22¼		1⅜		3⅝		20	16	1
16	.946	1	1600	3/16		23½		1 7/16		3¾		21¼	16	1
18	1.02	1 1/16	1690	3/16		25		1 9/16		3½		22¾	16	1⅛
20	1.09	1⅛	1780	3/16		27½		1 11/16		3⅝		25	20	1⅛
22	1.18	1 3/16	1850	¼		29½		1 13/16		3⅝		27¼	20	1¼
24	1.25	1¼	1920	¼	31½	32	1¼	1⅞	3¾	4	29¼	29½	20	1¼
26	1.30	1 5/16	1980	¼	33¾	34¼	1⅜	2	3⅞	4⅛	31¼	31¾	24	1¼
28	1.38	1⅜	2040	¼	36	36½	1 7/16	2 1/16	4	4¼	33½	34	28	1¼
30	1.48	1½	2000	¼	38	38¾	1½	2⅛	4	4⅜	35½	36	28	1⅜
36	1.71	1¾	1920	¼	44½	45¾	1¾	2⅜	4¼	4⅞	42	42¾	32	1⅜
42	1.87	2	2100	¼	51	52¾	1⅞	2⅝	4½	5⅜	48½	49½	36	1½
48	2.17	2¼	2130	¼	57½	59½	2	2¾	4¾	5¾	54¾	56	44	1½

NOTES.—Sizes up to 24 inches are designed for 200 lbs. or less.

Sizes from 24 to 48 inches are divided into two scales, one for 2 other for less.

The sizes of bolts given are for high pressure. For medium pr diameters are ⅛-inch less for pipes 2 to 20 inches diameter inclus inch less for larger sizes, except 48-inch pipe, for which the size of inches.

When two lines of figures occur under one heading, the single c to 24 inches are for both medium and high pressures. Beginni inches, the left-hand columns are for medium and the right-han for high pressures.

The sudden increase in diameters at 16 inches is due to the pos tion of wrought-iron pipe, making with a nearly constant width greater diameter desirable.

When wrought-iron pipe is used, if thinner flanges than those sufficient, it is proposed that bosses be used to bring the nuts standard lengths. This avoids the use of a reinforcement around

Figures in the third, fourth, fifth, and last columns refer only t

...es a vertical line parallel to the spindle

...es on the upper side of the flanges.

DIMENSIONS OF PIPE FLANGES AND CAST-IRON PIPES.

(J. E. Codman, Engineers' Club of Philadelphia, 1889.)

Diameter of Flange.	Diameter of Bolt Circle.	Diameter of Bolt	Number of Bolts.	Thickness of Flange.	Thickness of Pipe.		Weight per foot without Flange.	Weight of Flange and Bolts.
					Frac.	Dec.		
6¼	4¾	¾	4	⅝	⅜	.373	6.96	4.41
7½	5⅞	¾	4	⅝	13-32	.396	11.16	5.93
9	7	¾	6	11-16	7-16	.420	15.84	7.66
9¾	8	¾	6	¾	7-16	.443	21.00	9.63
10¾	9⅛	¾	8	¾	15-32	.466	26.64	11.82
13¼	11⅜	¾	8	13-16	½	.511	39.36	16.91
15¼	13¼	¾	10	⅞	9-16	.557	54.00	23.00
17¾	15¾	⅞	12	15-16	19-32	.603	70 56	30.13
20	18	⅞	14	1	21-32	.649	89.04	38.34
22	20	⅞	16	1 1-16	11-16	.695	109.44	47.70
24	22¼	⅞	16	1⅛	¾	.741	131.76	58.23
27	24½	1	18	1 3-16	25-32	.787	156.00	70 00
28¾	26½	1	20	1¼	27-32	.833	182 16	83.05
31¼	28¾	1	22	1 5-16	⅞	.879	210.24	97.42
33¼	31	1	24	1⅜	15-16	.925	240.24	113.18
35½	33¼	1	24	1 7-16	31-32	.971	272.16	130.35
38	35½	1	26	1 9-16	1	1.017	306.00	149.00
40	37½	1⅛	28	1⅝	1 1-16	1.063	341.76	169.17
42¾	40	1⅛	30	1 11-16	1⅛	1.109	379.44	190.90
45	42	1⅛	32	1¾	1 5-32	1.155	419.04	214.26
47	44	1⅛	32	1 13-16	1 3-16	1.201	460.56	239.27
49	46	1⅛	34	1⅞	1¼	1.247	504.00	266.00
51¼	48¼	1⅛	34	1 15-16	1 5-16	1.293	549.36	294 49
53½	50¼	1¼	36	2	1 11-32	1.339	596.64	324 78
55¾	52¾	1¼	38	2 1-16	1⅜	1.385	645.84	356.94
58	55	1¼	40	2⅛	1 7-16	1.431	696.96	391.00

D = Diameter of pipe. All dimensions in inches.

FORMULÆ.—Thickness of flange $= 0.033D + 0.56$.
Thickness of pipe $= 0.023D + 0.327$.
Weight of pipe per foot $= 0.24D^2 + 3D$.
Weight of flange $= .001D^3 + 0.1D^2 + D + 2$.
Diameter of flange $= 1.125D + 4.25$.
Diameter of bolt-circle $= 1.092D + 2.566$.
Diameter of bolt $= 0.011D + 0.73$.
Number of bolts $= 0.78D + 2.56$.

PIPE FLANGES FOR HIGH STEAM-PRESSURE.

(Chapman Valve Mfg. Co.)

Size of Pipe.	Diameter of Flange.	Number of Bolts.	Diameter of Bolts.	Diameter of Bolt Circle.	Length of Pipe-Thread.
Inches.	Inches.		Inches.	Inches.	Inches.
2½	7½	6	⅝	5⅞	1⅛
3	9	6	¾	6⅝	1⅜
3½	9	7	¾	7¼	1 7-16
4	10	8	¾	7⅞	1 9-16
4½	10½	8	¾	8½	1 11-16
[illegible]	11	9	¾	9¼	1 13-16
[illegible]	13	10	⅞	10⅝	1⅞
[illegible]	14	12	⅞	11⅞	1 15-16
[illegible]	15	12	⅞	13	2
[illegible]	16	13	⅞	14	2
[illegible]	17½	15	⅞	15¼	2⅛
[illegible]	20	18	⅞	17¾	2¼
[illegible]	23	18	1	20¼	2½
[illegible]	23½	18	1	21¼	2⅝

STANDARD SIZES, ETC., OF WROUGHT-IRON I
For Water, Gas, or Steam.

(Briggs Standard.)

Diameter of Tube.			Thickness of Metal.	Internal Circumference.	External Circumference.	Length of Pipe per Sq. Ft. of Inside Surface.	Length of Pipe per Sq. Ft. of Outside Surface.	Internal Area.
Nominal Inside.	Actual Inside.	Actual Outside.						
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Feet.	Feet.	Ins.
1/8	.270	.405	.068	.848	1.272	14.15	9.44	.0572
1/4	.364	.540	.088	1.144	1.696	10.50	7.075	.1041
3/8	.494	.675	.091	1.552	2.121	7.67	5.657	.1916
1/2	.623	.840	.109	1.957	2.652	6.13	4.502	.3048
3/4	.824	1.050	.113	2.589	3.299	4.635	3.637	.5333
1	1.048	1.315	.134	3.292	4.134	3.679	2.903	.8627
1 1/4	1.380	1.660	.140	4.335	5.215	2.768	2.301	1.496
1 1/2	1.610	1.900	.145	5.061	5.969	2 371	2.01	2.038
2	2.067	2.375	.154	6.494	7.461	1.848	1.611	3.355
2 1/2	2.468	2.875	.204	7.754	9.032	1.547	1.328	4.788
3	3.067	3.500	.217	9.636	10.996	1.245	1.091	7.388
3 1/2	3.548	4.000	.226	11.146	12.566	1.077	.955	9.887
4	4.026	4.500	.237	12.648	14.137	.949	.849	12.730
4 1/2	4.508	5.000	.246	14.153	15.708	.848	.765	15.939
5	5.045	5.563	.259	15.849	17.475	.757	.629	19.990
6	6.065	6.625	.280	19.054	20 813	.63	.577	28.889
7	7.023	7.625	.301	22.063	23.954	.544	.505	38.737
8	7.982	8.625	.322	25.076	27.096	.478	.444	50.039
*9	9.000	9.688	.344	28.277	30.433	.425	.394	63.633
10	10.019	10.750	.366	31.475	33.772	.381	.355	78.838

* By the action of the Manufacturers of Wrought-iron Pipe and Tubes, at a meeting held in New York, May 9, 1889, a change in size of outside diameter of 9-inch pipe was adopted, making the latter 9.625 i of 9.688 inches, as given in the table of Briggs' standard pipe diamete

For discussion of the Briggs Standard of Wrought-iron Pipe Dimen see Report of the Committee of the A. S. M. E. in "Standard Pipe an Threads," 1886. Trans., Vol. VIII, p. 29. The figures in the next to t column are derived from the formula

$$D - (0.05D + 1.9) \times \frac{1}{n},$$

in which D = outside diameter of the tubes, and n the number of thre the inch. The figures in the last column are derived from the fo $0.8\frac{1}{n} \times 2 + d$, or $1.6\frac{1}{n} + d$, in which d is the diameter at the bottom thread at the end of the pipe.

Having the taper, length of full-threaded portion, and the sizes at t and top of thread at the end of the pipe, as given in the table, taps a can be made to secure these points correctly, the length of the imp threaded portions on the pipe, and the length the tap is run into the beyond the point at which the size is as given, or, in other words, the end of the pipe, having no effect upon the standard. The angle thread is 60°, and it is slightly rounded off at top and bottom, so that, i of its depth being equal to its pitch, as is the case with a full V-thre 4/5 the pitch, or equal to $0.8\frac{1}{n}$, n being the number of threads per in

Sizes, etc., of Wrought-iron Pipe—(*Continued.*)

Sizes, etc.				Screwed Ends.			
Length of Pipe Containing One Cubic Foot.	Weight per Foot of Length.	Contents in U. S. Gallons per Foot.	Weight of Water per Foot of Length.	Number of Threads per Inch.	Length of Perfect Screw.	Diameter of Bottom of Thread at End of Pipe.	Diameter of Top of Thread at End of Pipe.
Feet.	Lbs.		Lbs.	No.	Inch.	Inches.	Inches.
2500.	.243	.0006	.005	27	.19	.334	.393
1385.	.422	.0026	.021	18	.29	.433	.522
751.5	.561	.0057	.047	18	.30	.567	.656
472.4	.845	.0102	.085	14	.39	.701	.815
270.	1.126	.0230	.190	14	.40	.911	1.025
166.9	1.670	.0408	.349	11½	.51	1.144	1.283
96.25	2 258	.0638	.527	11½	.54	1.488	1.627
70.65	2.694	.0918	.760	11½	.55	1.727	1.866
42.36	3.667	.1632	1.356	11½	.58	2.2	2.339
30.11	5.773	.2550	2.116	8	.89	2.62	2.82
19.49	7.547	.3673	3.049	8	.95	3.241	3.441
14.56	9.055	.4998	4.155	8	1.00	3.738	3.938
11.31	10.728	.6528	5.405	8	1.05	4.235	4.435
9.03	12.492	.8263	6.851	8	1.10	4.732	4.932
7.20	14.564	1.020	8.500	8	1.16	5.291	5.491
4.98	18.767	1.469	12.312	8	1.26	6.346	6.546
3.72	23.410	1.999	16.662	8	1.36	7.34	7.54
2.88	28.348	2.611	21.750	8	1.46	8.334	8.534
2.26	34.077	3.300	27.500	8	1.57	9.39	9.59
1.80	40.641	4.081	34.000	8	1.68	10.445	10.645

...er of conical tube ends, 1 in 32 to axis of tube = ¾ inch to the foot taper.
...ch and below are butt-welded, and proved to 300 pounds per square inch ...ulic pressure.
...inch and above are lap-welded, and proved to 500 pounds per square ...hydraulic pressure.

SIZES ABOVE 10 INCHES.

(Morris, Tasker & Co., Limited.)

Actual Inside Diameter.	Actual Outside Diameter.	Thickness.	Internal Circumference.	External Circumference.	Internal Area.	External Area.	Length of Pipe per sq. ft. of Inside Surface.	Length of Pipe per sq. ft. of Outside Surface.	Length of Pipe containing 1 cubic foot.	Weight per foot of Length.
in.	in.	in.	in.	in.	sq. in.	sq. in.	ft.	ft.	ft.	lbs.
11.224	12	.388	35.26	37.70	98.94	113.10	.340	.318	1.455	47.73
12.180	13	.41	38.26	40.84	116.54	132.73	.313	.293	1.235	54 66
13.136	14	.432	41.27	43.98	134.58	153.94	.290	.273	1.069	61.94
14.092	15	.454	44.27	47.12	155.97	176.72	.271	.254	.923	70.01
15.048	16	.476	47.27	50.27	177.87	201.06	.254	.238	.809	78.27
16.004	17	.498	50.28	53.41	201.16	226.98	.238	.225	.715	87.12
16.960	18	.520	53.28	56.55	225.91	254.47	.225	.212	.638	96.38
17.916	19	.542	56.28	59.69	252.10	283 53	.213	.201	.571	106.07
18.872	20	.564	59.29	62.83	279.72	314.16	.202	.191	.515	116.21
19.828	21	.586	62.29	65.97	308.77	346.36	.192	.183	.466	126.76

WROUGHT-IRON WELDED TUBES, EXTRA STRON[G]
Standard Dimensions.

Nominal Diameter.	Actual Outside Diameter.	Thickness, Extra Strong.	Thickness, Double Extra Strong.	Actual Inside Diameter, Extra Strong.	Actual Inside Diameter, Double Extra Strong.
Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
1/8	0.405	0.100	....	0.205	....
1/4	0.54	0.123	....	0.294	....
3/8	0.675	0.127	....	0.421	....
1/2	0.84	0.149	0.298	0.542	0.244
3/4	1.05	0.157	0.314	0.736	0.422
1	1.315	0.182	0.364	0.951	0.587
1 1/4	1.66	0.194	0.388	1.272	0.884
1 1/2	1.9	0.203	0.406	1.494	1.088
2	2.375	0.221	0.442	1.933	1.491
2 1/2	2.875	0.280	0.560	2.315	1.755
3	3.5	0.304	0.608	2.892	2.284
3 1/2	4.0	0.321	0.642	3.358	2.716
4	4.5	0.341	0.682	3.818	3.136

STANDARD SIZES, ETC., OF LAP-WELDED CHA[RCOAL-IRON] COAL-IRON BOILER-TUBES.

(Morris, Tasker & Co., Limited).

External Diameter.	Internal Diameter.	Standard Thickness.	Internal Circumference.	External Circumference.	Internal Area.		External Area.		Length of Tube per Sq. Ft. of Inside Surface.	Length of Tube per Sq. Ft. of Outside Surface.	Length of Tube per Sq. Ft. of Mean Surface.
Ins.	Ins.	Ins.	Ins.	Ins.	Sq. In.	Sq. Ft	Sq. In.	Sq. Ft	Ft.	Ft.	Ft.
1	.856	.072	2.689	3.142	.575	.004	.785	.0055	4.460	3.819	4.139
1 1-4	1.106	.072	3.474	3.927	.960	.0067	1.227	.0085	3.455	3.056	3.255
1 1-2	1.334	.083	4.191	4.712	1.396	.0097	1.767	.0123	2.863	2.547	2.705
1 3-4	1.560	.095	4.901	5.498	1.911	.0133	2.405	.0167	2.448	2.183	2.315
2	1.804	.098	5.667	6.283	2.556	.0177	3.142	.0218	2.118	1.909	2.013
2 1-4	2.054	.098	6.484	7.069	3.314	.0230	3.976	.0276	1.850	1.698	1.774
2 1-2	2.283	.109	7.172	7.854	4.094	.0284	4.909	.0341	1.673	1.528	1.600
2 3-4	2.533	.109	7.957	8.639	5.039	.035	5.940	.0412	1.508	1.390	1.449
3	2.783	.109	8.743	9.425	6.083	.0422	7.069	.0491	1.373	1.273	1.322
3 1-4	3.012	.119	9.462	10.210	7.125	.0495	8.296	.0576	1.268	1.175	1.221
3 1-2	3.262	.119	10.248	10.995	8.357	.058	9.621	.0668	1.171	1.091	1.131
3 3-4	3.512	.119	11.033	11.781	9.687	.0673	11.045	.0767	1.088	1.018	1.053
4	3.741	.130	11.753	12.566	10.992	.0763	12.566	.0872	1.023	.955	.989
4 1-2	4.241	.130	13.323	14.137	14.126	.0981	15.904	.1104	.901	.849	.875
5	4.720	.140	14.818	15.708	17.497	.1215	19.635	.1364	.809	.764	.786
6	5.699	.151	17.904	18.849	25.509	.1771	28.274	.1963	.670	.637	.653
7	6.657	.172	20.914	21.991	34.805	.2417	38.484	.2673	.574	.545	.560
8	7.636	.182	23.989	25.132	45.795	.318	50.265	.3490	.500	.478	.489
9	8.615	.193	27.055	28.274	58.291	.4048	63.617	.4418	.444	.424	.434
10	9.573	.214	30.074	31.416	71.975	.4998	78.540	.5454	.399	.382	.391
11	10.560	.22	33.175	34.557	87.479	.6075	95.033	.6601	.361	.347	.354
12	11.542	.229	36.26	37.699	103.749	.7205	113.097	.7854	.330	.318	.324
13	12.524	.238	39.345	40.840	123.187	.8554	132.732	.9213	.305	.293	.299
14	13.504	.248	42.414	43.982	143.189	.9943	153.938	1.069	.282	.272	.277
15	14.482	.259	45.496	47.124	164.718	1.1438	176.715	1.2272	.263	.254	.258
16	15.458	.271	48.562	50.265	187.667	1.3032	201.062	1.188	.247	.238	.242
17	16.432	.284	51.662	53.407	212.227	1.4738	226.980	1.5762	.232	.224	.228
18	17.416	.292	54.714	56.548	238.224	1.6543	254.469	1.7671	.219	.212	.215
19	18.400	.3	57.805	59.690	265.903	1.8465	283.529	1.969	.207	.200	.203
20	19.360	.32	60.821	62.832	294.373	2.0443	314.159	2.1817	.197	.190	.193
21	20.320	.34	63.837	65.973	324.311	2.2522	346.361	2.4053	.188	.181	.184

In estimating the effective steam-heating or boiler surface of tubes, the surf[ace in] contact with air or gases of combustion (whether internal or external to the tube[s]) be tak[en].

For ... by steam, superheating steam, or transferring heat fr[om] ... the mean surface of the tubes is to be taken.

find the square feet of surface, S, in a tube of a given length, L, in feet, diameter, d, in inches, multiply the length in feet by the diameter in es and by .2618. Or, $S = \frac{3.1416dL}{12} = .2618dL$. For the diameters in the e below, multiply the length in feet by the figures given opposite the neter.

ches, meter.	Square Feet per Foot Length.	Inches, Diameter.	Square Feet per Foot Length.	Inches, Diameter.	Square Feet per Foot Length.
¼	.0654	2¼	.5890	5	1.3090
½	.1309	2½	.6545	6	1.5708
¾	.1963	2¾	.7199	7	1.8326
1	.2618	3	.7854	8	2.0944
1¼	.3272	3¼	.8508	9	2.3562
1½	.3927	3½	.9163	10	2.6180
1¾	.4581	3¾	.9817	11	2.8798
2	.5236	4	1.0472	12	3.1416

RIVETED IRON PIPE.

(Abendroth & Root Mfg. Co.)

heets punched and rolled, ready for riveting, are packed in convenient ns for shipment. The following table shows the iron and rivets required punched and formed sheets.

mber Square Feet of Iron quired to make 100 Lineal et Punched and Formed eets when put together.			Approximate No. of Rivets 1 Inch apart, required for 100 Lineal Feet Punched and Formed Sheets.	Number Square Feet of Iron required to make 100 Lineal Feet Punched and Formed Sheets when put together.			Approximate No. of Rivets 1 Inch apart, required for 100 Lineal Feet Punched and Formed Sheets.
iam-eter in ches.	Width of Lap in Inches.	Square Feet.		Diam-eter in Inches.	Width of Lap in Inches.	Square Feet.	
3	1	90	1,600	14	1½	397	2,800
4	1	116	1,700	15	1½	423	2,900
5	1½	150	1,800	16	1½	452	3,000
6	1½	178	1,900	18	1½	506	3,200
7	1½	206	2,000	20	1½	562	3,500
8	1½	234	2,200	22	1½	617	3,700
9	1½	258	2,300	24	1½	670	3,900
10	1½	289	2,400	26	1½	725	4,100
11	1½	314	2,500	28	1½	779	4,400
12	1½	343	2,600	30	1½	836	4,600
13	1½	369	2,700	36	1½	998	5,200

WEIGHT OF ONE SQUARE FOOT OF SHEET-IRON FOR RIVETED PIPE.

Thickness by the Birmingham Wire-Gauge.

No. of Gauge.	Thickness in Decimals of an Inch.	Weight in lbs., Black.	Weight in lbs., Galvanized.	No of Gauge.	Thickness in Decimals of an Inch.	Weight in lbs., Black.	Weight in lbs., Galvanized.
26	.018	.72	.94	18	.049	1.97	2.19
24	.022	.88	1.13	16	.065	2.61	2.82
22	.028	1.12	1.38	14	.083	3.33	3.52
20	.035	1.40	1.69	12	.109	4.37	4.50

SPIRAL RIVETED PIPE.

(Abendroth & Root Mfg. Co.)

Thickness. B. W. G. No.	Thickness. Inches.	Diameter, Inches.	Approximate Weight in lbs. per foot in Length.	Approximate ...ing Pressure per sq. in.
26	.018	3 to 6	lbs.=	
24	.022	3 to 12	" =⅛ of diam. in ins.	
22	.028	3 to 14	" =.4 " "	
20	.035	3 to 24	" =.5 " "	2700 lbs.÷diam
18	.049	3 to 24	" =.6 " "	3600 " ÷ "
16	.065	6 to 24	" =.8 " "	4800 " ÷ "
14	.083	8 to 24	" =1.1 " "	6400 " ÷ "

The above are black pipes. Galvanized weighs from 10 to 30 p[er cent] heavier. Double Galvanized Spiral Riveted Flanged Pressure Pipe, to 150 lbs. hydraulic pressure.

Inside diameters, inches........	3	4	5	6	7	8	9	10	11	12	13	14	15
Thickness, B. W. G............	20	20	20	18	18	18	18	16	16	16	16	14	14
Nominal weight per foot, lbs...	2¼	3	4	5	6	7	8	11	12	14	15	20	22

DIMENSIONS OF SPIRAL PIPE FITTINGS.

Dimensions in Inches.

Inside Diameter.	Outside Diameter Flanges.	Number Bolt Holes.	Diameter Bolt Holes.	Diameter Circles on which Bolt Holes are Drilled.	Size Bo...
ins.					
3	6	4	½	4¾	7-16
4	7	8	½	5 15-16	7-16
5	8	8	½	6 15-16	7-16
6	8 ⅞	8	⅝	7⅞	½
7	10	8	⅝	9	½
8	11	8	⅝	10	½
9	13	8	⅝	11¼	½
10	14	8	⅝	12¼	½
11	15	12	⅝	13⅜	½
12	16	12	⅝	14¼	½
13	17	12	⅝	15¼	½
14	17 ⅞	12	⅝	16¼	½
15	19	12	⅝	17 7-16	½
16	21 3-16	12	⅝	19¼	½
18	23 ¼	16	11-16	21¼	½
20	25 ⅛	16	11-16	23⅛	½

SEAMLESS BRASS TUBE. IRON-PIPE SIZE

(Randolph & Clowes).

(For actual dimensions see tables of Wrought-iron Pipe.)

Nominal Size.	Weight per Foot, lbs.	Nominal Size.	Weight per Foot, lbs.	Nominal Size.	Weight per Foot, lbs.	Nominal Size.
⅛	.266	¾	1.228	2	4	4
¼	.461	1	1.837	2½	6.323	5
⅜	[illegible]	1¼	2.468	3	8.266	6
½	[illegible]	1½	3.045	3½	9.878	7
						8

SEAMLESS DRAWN BRASS-TUBING.

(Randolph & Clowes, Waterbury, Conn.)

iameter 3-16 to 7¾ inches. Thickness of walls 8 to 25 Stubbs' th 12 feet. The following are the standard sizes:

SEAMLESS DRAWN BRASS-TUBING.

gth eet.	Stubbs' or Old Gauge.	Outside Diameter.	Length Feet.	Stubbs' or Old Gauge.	Outside Diameter.	Length Feet.	Stubbs' or Old Gauge.
12	20	1⅜	12	14	2⅝	12	11
12	19	1½	12	14	2¾	12	11
12	19	1⅝	12	13	3	12	11
12	18	1¾	12	13	3¼	12	11
12	18	1 13-16	12	13	3½	12	11
12	17	1⅞	12	12	4	10 to 12	11
12	17	1 15-16	12	12	5	10 to 12	11
12	17	2	12	12	5¼	10 to 12	11
12	17	2⅛	12	12	5½	10 to 12	11
12	16	2¼	12	12	5¾	10 to 12	11
12	16	2⅜	12	12	6	10 to 12	11
12	15	2½	12	11			

COILED PIPES.

(National Pipe-bending Co., New Haven, Conn.)

S OF STEEL OR IRON PIPE; WELDED LENGTHS.

	Butt-welded Pipe.						Lap-welded Pipe.	
........................Inches	¼	⅜	½	¾	1	1¼	1½	2
le diameter of coil contain- t of pipe and less. . Inches	2	2½	3½	4	6	8	12	18
de diameter of coils over 25 ot over 200 feet.......Inches	6	7	7½	8½	9	11	14	18

F SEAMLESS DRAWN BRASS AND COPPER TUBING.

, outside ......Ins.	¼	⅜	½	¾	1	1¼	1⅜	1½	1¾	2	2¼	2⅜	2½
de diam- ls....Ins.	1	1½	2	3	4	6	7	8	10	12	14	16	18

lid drawn-steel tubes, imported by P. S. Justice & Co., Phila- made in sizes from ⅛ to 4⅛ inches e... l diameter, varying d with thickness of walls from 1-16 to 11-16 inches. The maxi- is 15 feet.

WEIGHT OF BRASS, COPPER, AND ZINC TU
Per Foot.

Thickness by Brown & Sharpe's Gauge.

Brass, No. 17.		Brass, No. 20.		Copper, Lightning-rod No. 23. / Zinc, No.
Inch.	Lbs.	Inch.	Lbs.	Inch.
1/4	.107	1/8	.032	1/2
5-16	.157	3-16	.039	9-16
3/8	.185	1/4	.063	5/8
7-16	.234	5-16	.106	11-16
1/2	.266	3/8	.126	3/4
9-11	.318	7-16	.158	**Zinc, No.**
5/8	.333	1/2	.189	1/2
3/4	.377	9-16	.208	5/8
7/8	.462	5/8	.220	3/4
1	.542	3/4	.252	7/8
1 1/8	.675	7/8	.284	1
1 1/4	.740	1	.378	1 1/4
1 1/2	.915	1 1/4	.500	1 1/2
1 3/4	.980	1 1/2	.580	
2	1.90			
2 1/2	1.506			
3	2.188			

LEAD PIPE IN LENGTHS OF 10 FEET.

In.	3-8 Thick.		5-16 Thick.		1/4 Thick.		3-16
	lb.	oz.	lb.	oz.	lb.	oz.	lb.
2 1/2	17	0	14	0	11	0	8
3	20	0	16	0	12	0	9
3 1/2	22	0	18	0	15	0	9
4	25	0	21	0	16	0	12
4 1/2					18	0	14
5	31	0			20	0	

LEAD WASTE-PIPE.

1 1/2 in., 2 lbs. per foot.
2 " 3 and 4 lbs. per foot.
3 " 3 1/2 and 5 lbs. per foot.
3 1/2 in., 4 lbs. per foot.
4 " 5, 6, and 8 lbs.
4 1/2 " 6 and 8 lbs.
5 in. 8, 10, and 12 lbs.

LEAD AND TIN TUBING.

1/8 inch. 1/4 inch.

SHEET LEAD.

Weight per square foot, 2 1/2, 3, 3 1/2, 4, 4 1/2, 5, 6, 8, 9, 10 lbs. and up
Other weights rolled to order.

BLOCK-TIN PIPE.

3/8 in., 4 1/2, 6 1/2, and 8 oz. per foot.
1/2 " [illegible] oz. "
[illegible] "
[illegible] "
1 in., 15, and 18 oz. per f
1 1/4 " 1 1/4 and 1 1/2 lbs. "
1 1/2 " 2 and 2 1/2 lbs. "
2 " 2 1/2 and 3 lbs. "

LEAD AND TIN-LINED LEAD PIPE.

(Tatham & Bros., New York.)

Letter.	Weight per Foot and Rod.	Thickness in 1-100th In.	Calibre.	Letter.	Weight per Foot and Rod.	Thickness in 1-100th In.
E	7 lbs. per rod		1 in.	E	1½ lbs. per foot	10
D	10 oz. per foot	6	"	D	2 " "	11
C	12 " "	8	"	C	2½ " "	14
B	1 lb. "	12	"	B	3¼ " "	17
A	1¼ " "	16	"	A	4 " "	21
AA	1½ " "	19	"	AA	4¾ " "	24
AAA	1¾ " "	27	"	AAA	6 " "	30
	13 oz. "		1¼ in.	E	2 " "	10
	1 lb. "		"	D	2½ " "	12
E	9 lbs. per rod	7	"	C	3 " "	14
D	¾ lb. per foot	9	"	B	3¾ " "	16
C	1 " "	11	"	A	4¾ " "	19
B	1¼ " "	13	"	AA	5¾ " "	25
	1½ " "		"	AAA	6¾ " "	
A	1¾ " "	16	1½ in.	E	3 " "	12
AA	2 " "	19	"	D	3½ " "	14
	2½ " "	23	"	C	4¼ " "	17
AAA	3 " "	25	"	B	5 " "	19
E	12 " per rod	8	"	A	6½ " "	23
D	1 " per foot	9	"	AA	8 " "	27
C	1½ " "	13	"	AAA	9 " "	
B	2 " "	16	1¾ in.	C	4 " "	13
A	2½ " "	20	"	B	5 " "	17
AA	2¾ " "	22	"	A	6½ " "	21
AAA	3½ " "	25	"	AA	8½ " "	27
E	1 " per foot	8	2 in.	C	4¾ " "	15
D	1¼ " "	10	"	B	6 " "	18
C	1¾ " "	12	"	A	7 " "	22
B	2¼ " "	16	"	AA	9 " "	27
A	3 " "	20		AAA	11¾ " "	
AA	3½ " "	23				
AAA	4¾ " "	30				

GHT OF LEAD PIPE WHICH SHOULD BE USED FOR A GIVEN HEAD OF WATER.

(Tatham & Bros., New York.)

or ber ed ll.	Pressure per sq. inch.	Calibre and Weight per Foot.						
		Letter.	⅜ inch.	½ inch.	⅝ inch.	¾ inch.	1 inch.	1¼ in.
ft.	15 lbs.	D	10 oz.	¾ lb.	1 lb.	1¼ lbs.	2 lbs.	2½ lbs.
ft.	25 lbs.	C	12 oz.	1 lb.	1½ lbs.	1¾ lbs.	2½ lbs.	3 lbs.
ft.	38 lbs.	B	1 lb.	1¼ lbs.	2 lbs.	2¼ lbs.	3¼ lbs.	3¾ lbs.
ft.	50 lbs.	A	1¼ lbs.	1¾ lbs.	2½ lbs.	3 lbs.	4 lbs.	4¾ lbs.
ft.	75 lbs.	AA	1½ lbs.	2 lbs.	2¾ lbs.	3½ lbs.	4¾ lbs.	6 lbs.
ft.	100 lbs.	AAA	1¾ lbs.	3 lbs.	3½ lbs.	4¾ lbs.	6 lbs.	6¾ lbs.

To find the thickness of lead pipe required when the head of water is given. (Chadwick Lead Works).

RULE.—Multiply the head in feet by size of pipe wanted, expressed decimally, and divide by 750; the quotient will give thickness required, in one-hundredths of an inch.

EXAMPLE.—*Required thickness of half-inch pipe for a head of 25 feet.*

$$25 \times 0.50 \div 750 = 0.16 \text{ inch.}$$

WEIGHT OF COPPER AND BRASS WIRE AND PLATES.

Brown & Sharpe's Gauge.

(From tables of leading manufacturers.)

No. of Gauge.	Size of Each No.	Weight of Wire per 1000 Lineal Feet.		Weight of Plates per Square Foot.		No. of Gauge.	Size of Each No.	Weight of Wire per 1,000 Lineal Feet.		Weight of Plates per Square Foot.	
		Copper.	Brass.	Copper.	Brass.			Copper.	Brass.	Copper.	Brass.
	Inch.	Lbs.	Lbs.	Lbs.	Lbs.		Inch.	Lbs.	Lbs.	Lbs.	Lbs.
0000	.46000	640.5	605.28	20.84	19.69	21	.028462	2.45	2.317	1.29	1.22
000	.40964	508.0	479.91	18.55	17.53	22	.025347	1.94	1.838	1.15	1.08
00	.36480	402.0	380.77	16.52	15.61	23	.022571	1.54	1.457	1.02	.966
0	.32476	319.5	301.82	14.72	13.90	24	.020100	1.22	1.155	.911	.860
1	.28930	253.3	239.45	13.10	12.38	25	.017900	.699	.916	.811	.766
2	.25763	200.9	189.82	11.67	11.03	26	.01494	.769	.727	.722	.682
3	.22942	159.3	150.52	10.39	9.82	27	.014195	.610	.576	.643	.608
4	.20431	126.4	119.48	9.25	8.74	28	.012641	.484	.457	.573	.541
5	.18194	100.2	94.67	8.24	7.79	29	.011257	.383	.362	.510	.482
6	.16202	79.46	75.08	7.34	6.93	30	.010025	.304	.287	.454	.429
7	.14428	63.01	59.55	6.54	6.18	31	.008928	.241	.228	.404	.382
8	.12849	49.98	47.22	5.82	5.50	32	.007950	.191	.181	.360	.340
9	.11443	39.64	37.44	5.18	4.90	33	.007080	.152	.143	.321	.303
10	.10189	31.43	29.69	4.62	4.36	34	.006304	.120	.114	.286	.270
11	.090742	24.92	23.55	4.11	3.88	35	.005614	.096	.0915	.254	.240
12	.080808	19.77	18.68	3.66	3.46	36	.005000	.0757	.0715	.226	.214
13	.071961	15.65	14.81	3.26	3.08	37	.004453	.0600	.0567	.202	.191

WEIGHT OF ROUND BOLT COPPER.

Per Foot.

…es.	Pounds.	Inches.	Pounds.	Inches.	Pounds.
	.425	1	3.02	1⅝	7.99
	.756	1⅛	3.83	1¾	9.27
	1.18	1¼	4.72	1⅞	10.64
	1.70	1⅜	5.72	2	12.10
	2.31	1½	6.81		

WEIGHT OF SHEET AND BAR BRASS.

…ness, …or …m.	Sheets per sq. ft.	Square Bars 1 ft. long.	Round Bars 1 ft. long.	Thickness, Side or Diam.	Sheets per sq. ft.	Square Bars 1 ft. long.	Round Bars 1 ft. long.
…es				Inches.			
…6	2.72	.014	.011	1 1-16	46.32	4.10	3.22
	5.45	.056	.045	1⅛	49.05	4.59	3.61
…6	8.17	.128	.100	1 3-16	51.77	5.12	4.02
	10.90	.227	.178	1¼	54.50	5.67	4.45
…6	13.62	.355	.278	1 5-16	57.22	6.26	4.91
	16.35	.510	.401	1⅜	59.95	6.86	5.39
…6	19.07	.695	.545	1 7-16	62.67	7.50	5.89
	21.80	.907	.712	1½	65.40	8.16	6.41
…6	24.52	1.15	.902	1 9-16	68.12	8.86	6.95
	27.25	1.42	1.11	1⅝	70.85	9.59	7.53
…6	29.97	1.72	1.35	1 11-16	73.57	10.34	8.12
	32.70	2.04	1.60	1¾	76.30	11.12	8.73
…6	35.42	2.40	1.88	1 13-16	79.02	11.93	9.36
	38.15	2.78	2.18	1⅞	81.75	12.76	10.01
…6	40.87	3.19	2.50	1 15-16	84.47	13.63	10.70
	43.60	3.63	2.85	2	87.20	14.52	11.40

…POSITION OF VARIOUS GRADES OF ROLLED BRASS, ETC.

Trade Name.	Copper	Zinc.	Tin.	Lead.	Nickel.
…non high brass	61.5	38.5	......	......	......
…w metal	60	40	......	......	......
…idge brass	66⅔	33⅓	......	......	......
…brass	80	20	......	......	......
… brass	60	40	......	1½	......
…rod	60	40	......	1½ to 2	......
…g brass	66⅔	33⅓	1½	......	......
…r cent German silver	61½	20½	......	......	18

…above table was furnished by the superintendent of a mill in Connec… in 1894. He says: While each mill has its own proportions for various …ures, depending upon the purposes for which the product is intended, …gures given are about the average standard. Thus, between cartridge … with 33⅓ per cent zinc and common high brass with 38½ per cent there are any number of different mixtures known generally as "high …" or specifically as "spinning brass," "drawing brass," etc., wherein …mount of zinc is dependent upon the amount of scrap used in the mix… …he degree of working to which the metal is to be subjected, etc.

AMERICAN STANDARD SIZES OF DROP-SHOT

	Diameter.	No. of Shot to the oz.		Diameter.	No. of Shot to the oz.		Diam-eter.
Fine Dust.	3-100″	10784	No. 8	Trap Shot	472	No. 2....	15-100″
Dust......	4-100	4565	" 8	9-100″	399	" 1..	16-100
No. 12.....	5-100	2326	" 7	Trap Shot	338	" B...	17-100
" 11.....	6-100	1346	" 7	10-100″	291	" BB.	18-100
" 10.....	Trap Shot	1056	" 6	11-100	218	" BBB	19-100
" 10.....	7-100″	848	" 5	12-100	168	" T..	20-100
" 9.....	Trap Shot	688	" 4	13-100	132	" TT..	21-100
" 9.....	8-100″	568	" 3	14-100	106	" F..	22-100
						" FF..	23-100

COMPRESSED BUCK-SHOT.

	Diameter.	No. of Balls to the lb.		Diameter.	No. o to t
No. 3.........	25-100″	284	No. 00.......	34-100″	
" 2.........	27-100	232	" 000......	36-100	
" 1.........	30-100	173	Balls.........	38-100	
" 0.........	32-100	140	"	44-100	

SCREW-THREADS, SELLERS OR U. S. STANDA

In 1864 a committee of the Franklin Institute recommended the a of the system of screw-threads and bolts which was devised by Mr. Sellers, of Philadelphia. This same system was subsequently ado the standard by both the Army and Navy Departments of the United and by the Master Mechanics' and Master Car Builders' Associati that it may now be regarded, and in fact is called, the United State dard.

The rule given by Mr. Sellers for proportioning the thread is as f Divide the pitch, or, what is the same thing, the side of the thre eight equal parts; take off one part from the top and fill in one part bottom of the thread; then the flat top and bottom will equal one e the pitch, the wearing surface will be three quarters of the pitch, diameter of screw at bottom of the thread will be expressed by mula

$$\text{diameter of bolt} - \frac{1.299}{\text{no. threads per inch}}.$$

For a sharp V thread with angle of 60° the formula is

$$\text{diameter of bolt} - \frac{1.733}{\text{no. of threads per inch}}.$$

The angle of the thread in the Sellers system is 60°. In the Whitw English system it is 55°, and the point and root of the thread are rou

Screw-Threads, United States Standard.

Diam.	Pitch.	Diam.	Pitch.	Diam.	Pitch.	Diam.	Pitch.	Diam.
¼	20	¾	10	1¼	7	1 15-16	5	2 13-1
5-16	18	13-16	10	1 5-16	6	2	4½	3
⅜	16	⅞	9	1⅜	6	2¼	4½	3¼
7-16	14	15-16	9	1½	6	2 5-16	4½	3 5-16
½	13	1	8	1⅝	5½	2⅜	4	3½
9-16	12	1 1-16	7	1¾	5	2½	4	3¾
⅝	[illegible]	[illegible]	7	1⅞	5	2¾	4	4

Screw-Threads, Whitworth (English) Standard.

Pitch.	Diam.	Pitch.	Diam.	Pitch.	Diam.	Pitch.	Diam.	Pitch.
20	5/8	11	1	8	1 3/4	5	3	3 1/2
18	11-16	11	1 1/8	7	1 7/8	4 1/2	3 1/4	3 1/4
16	3/4	10	1 1/4	7	2	4 1/2	3 1/2	3 1/4
14	13-16	10	1 3/8	6	2 1/4	4	3 3/4	3
12	7/8	9	1 1/2	6	2 1/2	4	4	3
12	15-16	9	1 5/8	5	2 3/4	3 1/2		

.S. OR SELLERS SYSTEM OF SCREW-THREADS.

OLTS AND THREADS.					HEX. NUTS AND HEADS.					
Threads per Inch.	Diam. of Root of Thread.	Width of Flat.	Area of Bolt Body in Sq. Inches.	Area at Root of Thread in Sq. Inches.	Short Diam., Rough.	Short Diam., Finish.	Long Diam., Rough.	Thickness, Rough.	Thickness, Finish.	Long Diam. Sq. Nuts Rough.
	Ins.	Ins.			Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
20	.185	.0062	.049	.027	1/2	7-16	37-64	1/4	3-16	7-10
18	.240	.0074	.077	.045	19-32	17-32	11-16	5-16	1/4	10-12
16	.294	.0078	.110	[illegible]	11-16	5/8	51-64	3/8	5-16	63-64
14	.344	.0089	.150	.093	25-32	23-32	9-10	7-16	3/8	1 7-64
13	.400	.0096	.196	.126	7/8	13-16	1	1/2	7-16	1 15-64
12	.454	.0104	.249	.162	31-32	29-32	1 1/8	9-16	1/2	1 23-64
11	.507	.0113	.307	.202	1 1-16	1	1 7-32	5/8	9-16	1 1/2
10	.620	.0125	.442	.302	1 1/4	1 3-16	1 7-16	3/4	11-16	1 49-64
9	.731	.0138	.601	.420	1 7-16	1 3/8	1 21-32	7/8	13-16	2 1-32
8	.837	.0156	.785	.550	1 5/8	1 9-16	1 7/8	1	15-16	2 19-64
7	.940	.0178	.994	.694	1 13-16	1 3/4	2 3-32	1 1/8	1 1-16	2 9-16
7	1.065	.0178	1.227	.893	2	1 15-16	2 5-16	1 1/4	1 3-16	2 53-64
6	1.160	.0208	1.485	1.057	2 3-16	2 1/8	2 17-32	1 3/8	1 5-16	3 3-32
6	1.284	.0208	[illegible]	1.295	2 3/8	2 5-16	2 3/4	1 1/2	1 7-16	3 23-64
5 1/2	1.389	.0227	2.074	1.515	2 9-16	2 1/2	2 31-32	1 5/8	1 9-16	3 5/8
5	1.491	.0250	2.405	1.746	2 3/4	2 11-16	3 3-16	1 3/4	1 11-16	3 57-64
5	1.616	.0250	2.761	2.051	2 15-16	2 7/8	3 13-32	1 7/8	1 13-16	4 5-32
4 1/2	1.712	.0277	3.142	2.302	3 1/8	3 1-16	3 5/8	2	1 15-16	4 27-64
4 1/2	1.962	.0277	3.976	3.023	3 1/2	3 7-16	4 1-16	2 1/4	2 3-16	4 61-64
4	2.176	.0312	4.909	3.719	3 7/8	3 13-16	4 1/2	2 1/2	2 7-16	5 31-64
4	2.426	.0312	5.940	4.620	4 1/4	4 3-16	4 29-32	2 3/4	2 11-16	6
3 1/2	2.629	.0357	7.069	5.428	4 5/8	4 9-16	5 3/8	3	2 15-16	6 17-32
3 1/2	2.879	.0357	8.296	6.510	5	4 15-16	5 13-16	3 1/4	3 3-16	7 1-16
3 1/4	3.100	.0384	9.621	7.548	5 3/8	5 5-16	6 7-64	3 1/2	3 7-16	7 39-64
3	3.317	.0413	11.045	8.641	5 3/4	5 11-16	6 21-32	3 3/4	3 11-16	8 1/8
3	3.567	.0413	12.566	9.993	6 1/8	6 1-16	7 3-32	4	3 15-16	8 41-64
2 7/8	3.798	.0435	14.186	11.329	6 1/2	6 7-16	7 9-16	4 1/4	4 3-16	9 3-16
2 3/4	4.028	.0454	15.904	12.743	6 7/8	6 13-16	7 31-32	4 1/2	4 7-16	9 3/4
2 5/8	4.256	.0476	17.721	14.226	7 1/4	7 3-16	8 13-32	4 3/4	4 11-16	10 1/4
2 1/2	4.480	.0500	19.635	15.763	7 5/8	7 9-16	8 27-32	5	4 15-16	10 49-64
2 1/2	4.730	.0500	21.648	17.572	8	7 15-16	9 9-32	5 1/4	5 3-16	11 23-64
2 3/8	4.953	.0526	23.758	19.267	8 3/8	8 5-16	9 23-32	5 1/2	5 7-16	11 7/8
2 3/8	5.203	.0526	25.967	21.262	8 3/4	8 11-16	10 5-32	5 3/4	5 11-16	12 3/8
2 1/4	5.423	.0555	28.274	23.098	9 1/8	9 1-16	10 19-32	6	5 15-16	12 15-16

T GAUGES FOR IRON FOR SCREW THREADS.

pting the Sellers, or Franklin Institute, or United States Standard, *riously called, a difficulty arose from* the fact that it is the habit *nufacturers to make iron over-size,* and as there are no over-size

STANDARD SET-SCREWS AND CAP-SCREWS.

American, Hartford, and Worcester Machine-Screw Companies.
(Compiled by W. S. Dix.)

	(A)	(B)	(C)	(D)	(E)	(F)	(G
Diameter of Screw....	⅛	3-16	¼	5-16	⅜	7-16	
Threads per Inch.....	40	24	20	18	16	14	
Size of Tap Drill*.....	No. 43	No. 30	No. 5	17-64	21-64	⅜	[illegible]

	(H)	(I)	(J)	(K)	(L)	(M)	(N
Diameter of Screw....	9-16	⅝	¾	⅞	1	1⅛	[illegible]
Threads per Inch....	12	11	10	9	8	7	[illegible]
Size of Tap Drill*....	31-64	17-32	21-32	49-64	⅞	63-64	[illegible]

Set Screws.			Hex. Head Cap-screws.			Sq. Head Cap-screws		
Short Diam of Head	Long Diam. of Head	Lengths (under Head).	Short Diam. of Head.	Long Diam. of Head.	Lengths (under Head).	Short Diam. of Head.	Long Diam. of Head.	Lengths (under Head)
(C) ¼	.35	¾ to 3	7-16	.51	¾ to 3	⅜	.53	¾ to 3
(D) 5-16	.44	¾ to 3¼	½	.58	¾ to 3¼	7-16	.62	¾ to 3
(E) ⅜	.53	¾ to 3½	9-16	.65	¾ to 3½	½	.71	¾ to 3
(F) 7-16	.62	¾ to 3¾	⅝	.72	¾ to 3¾	9-16	.80	¾ to 3
(G) ½	.71	¾ to 4	¾	.87	¾ to 4	⅝	.89	¾ to 4
(H) 9-16	.80	¾ to 4¼	13-16	.94	¾ to 4¼	11-16	.98	¾ to 4
(I) ⅝	.89	¾ to 4½	⅞	1.01	1 to 4½	¾	1.06	1 to 4
(J) ¾	1.06	1 to 4¾	1	1.15	1¼ to 4¾	⅞	1.24	1¼ to 4
(K) ⅞	1.24	1¼ to 5	1⅛	1.30	1½ to 5	1⅛	1.60	1½ to 5
(L) 1	1.42	1½ to 5	1¼	1.45	1¾ to 5	1¼	1.77	1¾ to 5
(M) 1⅛	1.60	1¾ to 5	1⅜	1.59	2 to 5	1⅜	1.95	2 to 5
(N) 1¼	1.77	2 to 5	1½	1.73	2 to 5	1½	2.13	2¼ to 5

Round and Filister Head Cap-screws.		Flat Head Cap-screws.		Button-head Cap-screws.	
Diam. of Head.	Lengths (under Head).	Diam. of Head.	Lengths (including Head).	Diam. of Head.	Lengths (under Head)
(A) 3-16	¾ to 2½	¼	¾ to 1¾	7-32 (.225)	¾ to 1¾
(B) ¼	¾ to 2¾	⅜	¾ to 2	5-16	¾ to 2
(C) ⅜	¾ to 3	15-32	¾ to 2¼	7-16	¾ to 2
(D) 7-16	¾ to 3¼	⅝	¾ to 2¾	9-16	¾ to 2
(E) 9-16	¾ to 3½	¾	¾ to 3	⅝	¾ to 3
(F) ⅝	¾ to 3¾	13-16	1 to 3	¾	¾ to 3
(G) ¾	¾ to 4	⅞	1¼ to 3	13-16	1 to 3
(H) 13-16	1 to 4¼	1	1½ to 3	15-16	1¼ to 3
(I) ⅞	1¼ to 4½	1⅛	1¾ to 3	1	1½ to 3
(J) 1	1½ to 4¾	1⅜	2 to 3	1¼	1¾ to 3
(K) 1⅛	1¾ to 5				
(L) 1¼	2 to 5				

* For cast iron. For numbers of twist-drills see p. 29.

Threads are U. S. Standard. Cap-screws are threaded ¾ length up to and including 1" diam. × 4" long, and ½ length above. Lengths increase by [illegible] each regular size between the limits given. Lengths of heads, except [illegible] and button, equal diam. of screws.

The angle of the cone of the flat-head screw is 76°, the sides making ang[illegible] with the top.

piece is turned to the size given in the ninth column, these sizes being the same as those of the regular U. S. Standard bolt, at the bottom of the thread, plus the amount allowed for fit and wear of tap; or, in other words, $d' =$ U. S. Standard $d + (D' - D)$. Gauges like the one in the cut, Fig. 72, are furnished for this sizing. In finishing the threads of the tap a tool

FIG. 72.

is used which has a removable cutter finished accurately to gauge by grinding, this tool being correct U. S. Standard as to angle, and flat at the point. It is fed in and the threads chased until the flat point just touches the portion of the tap which has been turned to size d'. Care having been taken with the form of the tool, with its grinding on the top face (a fixture being provided for this to insure its being ground properly), and also with the setting of the tool properly in the lathe, the result is that the threads of the tap are correctly sized without further attention.

It is evident that one of the points of advantage of the Sellers system is sacrificed, i.e., instead of the taps being flatted at the top of the threads they are sharp, and are consequently not so durable as they otherwise would be; but practically this disadvantage is not found to be serious, and is far overbalanced by the greater ease of getting iron within the prescribed limits; while any rough bolt when reduced in size at the top of the threads, by filing or otherwise, will fit a hole tapped with the U. S. Standard hand taps, thus affording proof that the two kinds of bolts or screws made for the two different kinds of work are practically interchangeable. By this system ¾″ iron can be .005″ smaller or .0108″ larger than the nominal diameter, or, in other words, it may have a total variation of .0158″, while 1¼″ iron can be .0105″ smaller or .0309″ larger than nominal—a total variation of .0414″—and within these limits it is found practicable to procure the iron.

STANDARD SIZES OF SCREW-THREADS FOR BOLTS AND TAPS.

(CHAS. A. BAUER.)

1	2	3	4	5	6	7	8	9	10
A	n	D	d	h	f	$D' - D$	D'	d'	H
		Inches.	Inches	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
¼	20	.2608	.1855	.0379	.0062	.006	.2668	.1915	.2024
5-16	18	.3245	.2403	.0421	.0070	.006	.3305	.2463	.2589
⅜	16	.3885	.2938	.0474	.0078	.006	.3945	.2998	.3139
7-16	14	.4530	.3447	.0541	.0089	.006	.4590	.3507	.3670
½	13	.5166	.4000	.0582	.0096	.006	.5225	.4060	.4236
9-16	12	.5805	.4543	.0631	.0104	.007	.5875	.4613	.4802
⅝	11	.6447	.5069	.0689	.0114	.007	.6517	.5139	.5346
¾	10	.7717	.6201	.0758	.0125	.007	.7787	.6271	.6499
⅞	9	.8991	.7307	.0842	.0139	.007	.9061	.7377	.7630
1	8	1.0271	.8376	.0947	.0156	.007	1.0341	.8446	.8731
1⅛	7	1.1559	.9394	.1083	.0179	.007	1.1629	.9464	.9789
1¼	7	1.2809	1.0644	.1083	.0179	.007	1.2879	1.0714	1.1039

A = nominal diameter of bolt.
D = actual diameter of bolt.
d = diameter of bolt at bottom of thread.
n = *number of threads per inch.*
f = *flat of bottom of thread.*
h = *depth of thread.*
D' *and* d' = *diameters of tap.*
H = *hole in nut before tapping.*

$$D = A + \frac{.2165}{n}.$$

$$d = A - \frac{1.29904}{n}.$$

$$h = \frac{.7577}{n} = \frac{D - d}{2}.$$

$$f = \frac{.125}{n}.$$

$$H = D' - \frac{1.288}{n} = D' - .85(2h.)$$

STANDARD SET-SCREWS AND CAP-SCREWS.

American, Hartford, and Worcester Machine-Screw Companies.
(Compiled by W. S. Dix.)

	(A)	(B)	(C)	(D)	(E)	(F)	(G)
Diameter of Screw....	⅛	3-16	¼	5-16	⅜	7-16	½
Threads per Inch.....	40	24	20	18	16	14	13
Size of Tap Drill*.....	No. 43	No. 30	No. 5	17-64	21-64	⅜	27-64

	(H)	(I)	(J)	(K)	(L)	(M)	(N)
Diameter of Screw....	9-16	⅝	¾	⅞	1	1⅛	1¼
Threads per Inch....	12	11	10	9	8	7	7
Size of Tap Drill*....	31-64	17-32	21-32	49-64	⅞	63-64	1⅛

Set Screws.			Hex. Head Cap-screws.			Sq. Head Cap-screws.		
Short Diam of Head	Long Diam. of Head	Lengths (under Head).	Short Diam. of Head.	Long Diam. of Head.	Lengths (under Head).	Short Diam. of Head.	Long Diam. of Head.	Lengths (under Head)
(C) ¼	.35	¾ to 3	7-16	.51	¾ to 3	⅜	.53	¾ to 3
(D) 5-16	.44	¾ to 3¼	½	.58	¾ to 3¼	7-16	.62	¾ to 3¼
(E) ⅜	.53	¾ to 3½	9-16	.65	¾ to 3½	½	.71	¾ to 3½
(F) 7-16	.62	¾ to 3¾	⅝	.72	¾ to 3¾	9-16	.80	¾ to 3¾
(G) ½	.71	¾ to 4	¾	.87	¾ to 4	⅝	.89	¾ to 4
(H) 9-16	.80	¾ to 4¼	13-16	.94	¾ to 4¼	11-16	.98	¾ to 4¼
(I) ⅝	.89	¾ to 4½	⅞	1.01	1 to 4½	¾	1.06	1 to 4½
(J) ¾	1.06	1 to 4¾	1	1.15	1¼ to 4¾	⅞	1.24	1¼ to 4¾
(K) ⅞	1.24	1¼ to 5	1⅛	1.30	1½ to 5	1⅛	1.60	1½ to 5
(L) 1	1.42	1½ to 5	1¼	1.45	1¾ to 5	1¼	1.77	1¾ to 5
(M) 1⅛	1.60	1¾ to 5	1⅜	1.59	2 to 5	1⅜	1.95	2 to 5
(N) 1¼	1.77	2 to 5	1½	1.73	2 to 5	1½	2.13	2¼ to 5

Round and Filister Head Cap-screws.			Flat Head Cap-screws.		Button-head Cap-screws.	
	Diam. of Head.	Lengths (under Head).	Diam. of Head.	Lengths (including Head).	Diam. of Head.	Lengths (under Head).
(A)	3-16	¾ to 2½	¼	¾ to 1¾	7-32 (.225)	¾ to 1¾
(B)	¼	¾ to 2¾	⅜	¾ to 2	5-16	¾ to 2
(C)	⅜	¾ to 3	15-32	¾ to 2¼	7-16	¾ to 2¼
(D)	7-16	¾ to 3¼	⅝	¾ to 2¾	9-16	¾ to 2½
(E)	9-16	¾ to 3½	¾	¾ to 3	⅝	¾ to 2¾
(F)	⅝	¾ to 3¾	13-16	1 to 3	¾	¾ to 3
(G)	¾	¾ to 4	⅞	1¼ to 3	13-16	1 to 3
(H)	13-16	1 to 4¼	1	1½ to 3	15-16	1¼ to 3
(I)	⅞	1¼ to 4½	1⅛	1¾ to 3	1	1½ to 3
(J)	1	1½ to 4¾	1⅜	2 to 3	1¼	1¾ to 3
(K)	1⅛	1¾ to 5				
(L)	1¼	2 to 5				

* For cast iron. For numbers of twist-drills see p. 29.

Threads are U. S. Standard. Cap screws are threaded ¾ length up to and including 1″ diam. × 4″ long, and ½ length above. Lengths increase by ¼″ each regular size between the limits given. Lengths of heads, except flat and button, equal diam. of screws.

The angle of the cone of the flat-head screw is 76°, the sides making angles *of 52° with the top.*

STANDARD MACHINE SCREWS.

(Am. Screw Co.'s Catalogue, 1883, 1892.)

Threads per Inch.	Diam. of Body.	Diam. of Flat Head.	Diam. of Round Head.	Diam. of Filister Head.	Lengths.	
					From	To
56	.0842	.1631	.1544	.1332	3-16	½
48	.0973	.1894	.1786	.1545	3-16	⅝
32, 36, 40	.1105	.2158	.2028	.1747	3-16	¾
32, 36, 40	.1236	.2421	.2270	.1985	3-16	⅞
30, 32	.1368	.2684	.2512	.2175	3-16	1
30, 32	.1500	.2947	.2754	.2392	¼	1⅛
30, 32	.1631	.3210	.2936	.2610	¼	1¼
24, 30, 32	.1763	.3474	.3238	.2805	¼	1⅜
24, 30, 32	.1894	.3737	.3480	.3035	¼	1½
20, 24	.2158	.4263	.3922	.3445	⅜	1¾
20, 24	.2421	.4790	.4364	.3885	⅜	2
16, 18, 20	.2684	.5316	.4866	.4300	⅜	2¼
16, 18	.2947	.5842	.5248	.4710	½	2½
16, 18	.3210	.6368	.5690	.5200	½	2¾
16, 18	.3474	.6894	.6106	.5557	½	3
14, 16	.3737	.7420	.6522	.6005	½	3
14, 16	.4000	.7420	.6938	.6425	½	3
14, 16	.4263	.7946	.7354	.6920	¾	3
14, 16	.4520	.8473	.7770	.7240	⅞	3
					1	3

gths vary by 16ths from 3-16 to ½, by 8ths from ½ to 1½, by 4ths from
3.

SIZES AND WEIGHTS OF SQUARE AND HEXAGONAL NUTS.

ted States Standard Sizes. Chamfered and trimmed. Punched to suit U. S. Standard Taps.

Width.	Thickness.	Diam. of Hole.	Long Diam. Sq. Nuts.	Long Diam. Hex. Nuts.	Square.		Hexagon.	
					No. in 100 lbs.	Wt. each in lbs.	No. in 100 lbs.	Wt. each in lbs.
½	¼	13-64	11-16	9-16	7270	.0138	7615	.0131
19-32	5-16	¼	13-16	11-16	4700	.0231	5200	.0192
11-16	⅜	19-64	1	13-16	2350	.0426	3000	.0333
25-32	7-16	11-32	1⅛	⅞	1630	.0613	2000	.050
⅞	½	25-64	1¼	1	1120	.0893	1430	.070
31-32	9-16	29-64	1⅜	1⅛	890	.1124	1100	.091
1 1-16	⅝	33-64	1½	1¼	640	.156	740	.135
1¼	¾	39-64	1¾	1 7-16	380	.263	450	.222
1 7-16	⅞	47-64	2 1-16	1 11-16	280	.357	309	.324
1⅝	1	53-64	2 5-16	1⅞	170	.588	216	.463
1 13-16	1⅛	59-64	2 9-16	2 1-16	130	.769	148	.676
2	1¼	1 1-16	2 13-16	2 5-16	96	1.04	111	.901
2 3-16	1⅜	1 5-32	3⅛	2½	70	1.43	85	1.18
2⅜	1½	1 9-32	3⅜	2¾	58	1.72	68	1.47
2 9-16	1⅝	1 13-32	3⅝	2 15-16	44	2.27	56	1.79
2¾	1¾	1½	3⅞	3 3-16	34	2.94	40	2.50
2 15-16	1⅞	1⅝	4⅛	3⅜	30	3.33	37	2.70
3⅛	2	1 23-32	4 7-16	3⅝	23	4.35	29	3.45
3½	2¼	1 15-16	4 15-16	4 1-16	19	5.26	21	4.76
3⅞	2½	2 3-16	5½	4½	12	8.33	15	6.67
4¼	2¾	2 7-16	6	4 15-16	9	11.11	11	9.09
4⅝	3	2⅝	6½	5 5-16	7⅛	13.64	8½	11.76

SIZES OF WASHERS.

Diameter in inches.	Size of Hole, in inches.	Thickness, Birmingham Wire-gauge.	Bolt in inches.	No. in 100 l
⅝	5-16	No. 16	¼	29,300
¾	⅜	" 16	5-16	18,000
1	7-16	" 14	⅜	7,600
1½	9-16	" 11	½	3,300
1½	⅝	" 11	9-16	2,180
1½	11-16	" 11	⅝	2,350
1¾	13-16	" 11	¾	1,680
2	31-32	" 10	⅞	1,140
2½	1⅛	" 8	1	580
2¾	1¼	" 8	1⅛	470
3	1⅜	" 7	1¼	360
3	1½	" 6	1⅜	360

TRACK SPIKES.

Rails used.	Spikes.	Number in Keg, 200 lbs.	Kegs per Mil Ties 24 in. between Centr
45 to 85	5½ × 9-16	380	30
40 " 52	5 × 9-16	400	27
35 " 40	5 × ½	490	22
24 " 35	4½ × ½	550	20
24 " 30	4½ × 7-16	725	15
18 " 24	4 × 7-16	820	13
16 " 20	3½ × ⅜	1250	9
14 " 16	3 × ⅜	1350	8
8 " 12	2½ × ⅜	1550	7
8 " 10	2½ × 5-16	2200	5

STREET RAILWAY SPIKES.

Spikes.	Number in Keg, 200 lbs.	Kegs per Mile, Ties 24 between Centres.
5½ × 9-16	400	30
5 × ½	575	19
4½ × 7-16	800	13

BOAT SPIKES.

Number in Keg of 200 lbs.

Length.	¼	5-16	⅜	½
4 inch.	2375	....	....	....
5 "	2050	1230	940	....
6 "	1825	1175	800	450
7 "	....	990	650	375
8 "	....	880	600	335
9 "	....	....	525	300
10 "	....	....	475	275

WROUGHT SPIKES.

Number of Nails in Keg of 150 Po......

Size.	¼ in.	5-16 in.	⅜ in.	7-16 in.	½ in.
ches......	2250	....	....	....	....
"	1890	1208	....	....	....
"	1650	1135	....	....	....
"	1464	1064	....	....	....
"	1380	930	742	....	....
"	1292	868	570	....	....
"	1161	662	482	445	306
"	....	635	455	384	256
"	....	573	424	300	240
"	....	....	391	270	222
"	....	....	....	249	203
"	....	....	....	236	180

WIRE SPIKES.

Size.	Approx. Size of Wire Nails	Ap. No. in 1 lb.	Size.	Approx. Size of Wire Nails.	Ap. No. in 1 lb.
pike......	3 in. No. 7	50	60d Spike......	6 in. No. 1	10
"	3½ " " 6	35	6½ in."	6½ " " 1	9
"	4 " " 5	26	7 " "	7 " " 0	7
"	4½ " " 4	20	8 " "	8 " " 00	5
"	5 " " 3	15	9 " "	9 " " 00	4½
"	5½ " " 2	12			

ENGTH AND NUMBER OF CUT NAILS TO THE POUND.

Size.	Length.	Common.	Clinch.	Fence.	Finishing.	Fine.	Barrel.	Casing.	Brads.	Tobacco.	Cut Spikes.
....	¾ in.	......	...	....	......	......	800	......	......	.....	....
....	⅞	......	....	....	......	......	500	......	.....	.....	....
....	1	800	...	...	1100	1000	376	......	......	.....	....
....	1¼	480	....	....	720	760	224	......	......	.....	....
....	1½	288	....	....	523	368	180	398	......	......	....
....	1¾	200	...	....	410	......	......	......	......	130	....
....	2	168	95	84	268	......	......	224	126	96	...
....	2¼	124	74	64	188	......	......	......	98	82	....
....	2½	88	62	48	146	......	......	128	75	68	....
....	2¾	70	53	36	130	......	......	110	65	.....	....
....	3	58	46	30	102	......	......	91	55	.....	28
....	3¼	44	42	24	76	......	......	71	40	.....	....
....	3½	34	38	20	62	......	......	54	27	.....	22
....	4	23	33	16	54	......	......	40	......	.....	14½
....	4½	18	20	....	......	......	......	33	......	.....	12½
....	5	14	....	....	......	......	......	27	......	.....	9½
....	5½	10	....	....	......	......	......	......	......	.....	8
....	6	8	....	....	......	......	......	......	......	.....	6

ES, LENGTH, AND NUMBER TO THE POUND OF STANDARD STEEL WIRE NAILS.

(John A. Roebling's Sons Co.)

Sizes.	Length, inches.	Common Nails and Brads.	Barbed Common.	Clinch.	Fence.	Smooth and Barbed Finishing.	Fine.	Barrel.	Casing, and Smooth and Barbed Box.	Flooring Brads.	Barbed Oval Head Car Nail. Light.	Barbed Oval Head Car Nail. Heavy.	Slating.	Barbed Roofing.	Shingle.	Tobacco.	Lining.	Wire Spikes.	Length, inches.	Sizes.
	¾							1500						714			2100		¾	
	⅞							1000						469			1780		⅞	
2d	1	1200	876	710		1558	1550	875	1350				411	411			1500		1	2d
3d fine	1⅛						1140	775											1⅛	3d fine
3d common	1¼	720	568	429		980		560	913				251	251					1¼	3d c'm
	1⅜							390											1⅜	
4d	1½	432	357	274		760	760	350	584		274	165	209	165		274			1½	4d
5d	1¾	300	245	235	142	575			410		142	118	142	142	270	235			1¾	5d
6d	2	252	204	157	124	350			310	157	124	103		103	204	157			2	6d
7d	2¼	186	139	139	92	275			238	139	92	76			182	139			2¼	7d
8d	2½	132	99	99	82	190			170	99	82	69			125	99			2½	8d
9d	2¾	105	90	90	62	173			150	90	62	54			114	90			2¾	9d
10d	3	87	69	83	50	137			121	67	57	50			83	69		50	3	10d
12d	3¼	66	53	64	38	98			97	53	47	38							3¼	12d
16d	3½	51	43	59	30	81			72	43	43	35						35	3½	16d
20d	4	35	31	43	23	71			54		31	26						26	4	20d
[illegible]	4½	27	24						46		28	24						20	4½	30d

APPROXIMATE NUMBER OF WIRE NAILS PER POUND.

Wire Gauge. B. W. G.	Length, inches. ¼	⅜	½	⅝	¾	1	1¼	1½	1¾	2	2½	3	3½	4	4½	5	6	7	8	9	10	11	12
00	....	....	....	....	....	....	33	27	23	20	16	14	12	10	9	8	7	6	5	4½	4	3¾	3¼
0	....	....	....	....	....	....	34	29	25	21	17	15	13	11	10	9	8	7	5½	5	4⅓	4	3½
1	....	....	....	....	....	57	45	38	32	28	23	19	16	14	13	11	10	8	7	6	5½	5	4½
2	....	....	....	....	....	65	52	44	37	32	26	22	19	16	14	13	11	9	8	7	6½	6	5½
3	....	....	....	....	100	76	60	50	43	38	30	25	22	19	17	15	13	11	10	8	7½	....	....
4	....	....	....	....	120	90	72	60	51	45	36	30	26	23	20	18	15	13	11	10	9	....	....
5	....	....	211	169	141	106	85	71	60	53	42	35	30	26	24	21	18	15	....	....	....	....	....
6	....	....	247	197	164	123	99	82	71	62	50	41	35	31	28	25	21	18	....	....	....	....	....
7	....	....	299	239	200	149	120	100	85	75	60	50	43	37	33	30	25	....	....	....	....	....	....
8	....	....	345	275	229	172	137	115	98	86	69	57	49	48	39	35	29	....	....	....	....	....	....
9	....	....	414	331	276	207	165	138	118	103	82	69	59	52	46	41	....	....	....	....	....	....	....
10	....	663	496	397	333	248	198	165	142	124	99	83	71	62	55	50	....	....	....	....	....	....	....
11	....	837	628	502	418	314	251	209	179	157	125	105	90	79	70	....	....	....	....	....	....	....	....
12	....	1096	822	658	548	411	329	274	235	204	164	137	117	103	....	....	....	....	....	....	....	....	....
13	....	1429	1072	857	714	536	429	357	306	268	214	178	153	....	....	....	....	....	....	....	....	....	....
14	2840	1893	1420	1136	947	710	568	473	406	350	284	236	....	....	....	....	....	....	....	....	....	....	....
15	3504	2336	1752	1402	1168	876	701	584	500	438	350												
16	4571	3048	2280	1828	1523	1143	913	761	653	571	....												
17	6233	4156	3116	2495	2077	1558	1246	1038	890	779	....												
18	8276	5517	4138	3310	2758	2069	1655	1379	1182	....	....												
19	10668	7112	5334	4267	3556	2667	2133	1778	....	....	....												
20	15000	10000	7500	6000	5000	3750	3000	....	....	....	....												
[illegible]	17777	11850	8888	7111	5926	4444	....	....	....	....	....												
[illegible]	22856	15237	11428	9143	7618	....	....	....	....	....	....												

These approximate numbers are an average only, and the figures given may be varied either way by changes in the dimensions of the heads or points. Brads and no-head nails will run more to the pound than the table shows, and large or thick-headed nails will run less.

SIZE, WEIGHT, LENGTH, AND STRENGTH OF IRO WIRE.

(Trenton Iron Co.)

No. by Wire Gauge.	Diam. in Decimals of One Inch.	Area of Section in Decimals of One Inch.	Feet to the Pound.	Weight of One Mile in pounds.	Tensile Strength (Ap proximate) of Charco Iron Wire in Pounds.	
					Bright.	Anneale
00000	.450	.15904	1.863	2833.248	12598	9449
0000	.400	.12566	2.358	2238.878	9955	7466
000	.360	.10179	2.911	1813.574	8124	6091
00	.330	.08553	3.465	1523.861	6880	5160
0	.305	.07306	4.057	1301.678	5926	4445
1	.285	.06379	4.645	1136.678	5226	3990
2	.265	.05515	5.374	982.555	4570	3425
3	.245	.04714	6.286	839.942	3948	2960
4	.225	.03976	7.454	708.365	3374	2530
5	.205	.03301	8.976	588.139	2839	2130
6	.190	.02835	10.453	505.084	2476	1860
7	.175	.02405	12.322	428.472	2136	1600
8	.160	.02011	14.736	358.3008	1813	1360
9	.145	.01651	17.950	294.1488	1507	1130
10	.130	.01327	22.333	236.4384	1233	925
11	.1175	.01084	27.340	193.1424	1010	758
12	.105	.00866	34.219	154.2816	810	607
13	.0925	.00672	44.092	119.7504	631	473
14	.080	.00503	58.916	89.6016	474	356
15	.070	.00385	76.984	68.5872	372	280
16	.061	.00292	101.488	52.0080	292	220
17	.0525	.00216	137.174	38.4912	222	165
18	.045	.00159	186.335	28.3378	169	127
19	.040	.0012566	235.084	22.3872	137	103
20	.035	.0009621	308.079	17.1389	107	80
21	.031	.0007547	392.772	13.4429		
22	.028	.0006157	481.234	10.9718		
23	.025	.0004909	603.863	8.7437		
24	.0225	.0003976	745.710	7.0805		
25	.020	.0003142	943.396	5.5968		
26	.018	.0002545	1164.689	4.5334		
27	.017	.0002270	1305.670	4.0439		
28	.016	.0002011	1476.869	3.5819		
29	.015	.0001767	1676.989	3.1485		
30	.014	.0001539	1925.321	2.7424		
31	.013	.0001327	2232.653	2.3649		
32	.012	.0001131	2620.607	2.0148		
33	.011	.0000950	3119.092	1.6928		
34	.010	.00007854	3773.584	1.3992		
35	.0095	.00007088	4182.508	1.2624		
36	.009	.00006362	4657.728	1.1336		
37	.0085	.00005675	5222.035	1.0111		
38	.008	.00005027	5896.147	.89549		
39	.0075	.00004418	6724.291	.78672		
40	.007	.00003848	7698.253	.68587		

The above figures on tensile strength are based upon tests made with good charcoal-iron wire from Trenton blooms.
The tensile strength of wire made of—

Good refined iron is about 10% less,
Swedish charcoal iron is about 10 " "
Mild Bessemer steel is about 10 more,
Ordinary crucible steel is about 25 " "
Special crucible steel is from 30 to 120 "
than that of charcoal-iron wire.

ALVANIZED IRON WIRE FOR TELEGRAPH AND TELEPHONE LINES.

(Trenton Iron Co.)

EIGHT PER MILE-OHM.—This term is to be understood as distinguishing *resistance of material* only, and means the weight of such material re-ed per mile to give the resistance of one ohm. To ascertain the mileage tance of any wire, divide the "weight per mile-ohm" by the weight of wire per mile. Thus in a grade of Extra Best Best, of which the weight mile-ohm is 5000, the mileage resistance of No. 6 (weight per mile 525 would be about 9½ ohms; and No. 14 steel wire, 6500 lbs. weight per -ohm (95 lbs. weight per mile), would show about 69 ohms.

zes of Wire used in Telegraph and Telephone Lines.

. 4. Has not been much used until recently; is now used on important where the multiplex systems are applied.
. 5. Little used in the United States.
. 6. Used for important circuits between cities.
. 8. Medium size for circuits of 400 miles or less.
. 9. For similar locations to No. 8, but on somewhat shorter circuits; l lately was the size most largely used in this country.
s. 10, 11. For shorter circuits, railway telegraphs, private lines, police fire-alarm lines, etc.
. 12. For telephone lines, police and fire-alarm lines, etc.
s. 13, 14. For telephone lines and short private lines: steel wire is used t generally in these sizes.

e coating of telegraph wire with zinc as a protection against oxidation w generally admitted to be the most efficacious method.

e grades of line wire are generally known to the trade as "Extra Best t" (E. B. B.), "Best Best" (B. B.), and "Steel."

Extra Best Best" is made of the very best iron, as nearly pure as any mercial iron, soft, tough, uniform, and of very high conductivity, its ght per mile-ohm being about 5000 lbs.

he "Best Best" is of iron, showing in mechanical tests almost as good lts as the E. B. B., but not quite as soft, and being somewhat lower in ductivity; weight per mile-ohm about 5700 lbs.

he Trenton "Steel" wire is well suited for telephone or short telegraph s, and the weight per mile-ohm is about 6500 lbs.

he following are (approximately) the weights per mile of various sizes of vanized telegraph wire, drawn by Trenton Iron Co.'s gauge:

No.	4,	5,	6,	7,	8,	9,	10,	11,	12,	13,	14.
Lbs.	720,	610,	525,	450,	375,	310,	250,	200,	160,	125,	95.

TESTS OF TELEGRAPH WIRE.

he following data are taken from a table given by Mr. Prescott relating ests of E. B. B. galvanized wire furnished the Western Union Telegraph :

ize of ire.	Diam. Parts of One Inch.	Weight.		Length. Feet per pound.	Resistance. Temp. 75.8° Fahr.		Ratio of Breaking Weight to Weight per mile.
		Grains, per foot.	Pounds per mile.		Feet per ohm.	Ohms per mile.	
4	.238	1043.2	886.6	6.00	958	5.51	
5	.220	891.3	673.0	7.85	727	7.26	
6	.203	758.9	572.2	9.20	618	8.54	3.05
7	.180	596.7	449.9	11.70	578	10.86	3.40
8	.165	501.4	378.1	14.00	409	12.92	3.07
9	.148	403.4	304.2	17.4	328	16.10	3.38
10	.134	330.7	249.4	21.2	269	19.60	3.37
11	.120	265.2	200.0	26.4	216	24.42	2.97
12	.109	218.8	165.0	32.0	179	29.60	3.43
14	.083	126.9	95.7	55.2	104	51.00	3.05

JOINTS IN TELEGRAPH WIRES.—The fewer the joints in a line the better. All joints should be carefully made and well soldered over, for a bad joint may cause as much *resistance to the electric* current as several miles of wire.

TABLE OF DIMENSIONS, WEIGHT, AND RESISTANCE OF COPPER WIRE.
(Birmingham Gauge.)

Gauge Number.	Diameter, Inch.	Sectional Area in Circular Mils. = diam².	Weight. Lbs. per Foot.	Weight. Lbs. per Ohm.	Length. Feet per Lb.	Length. Feet per Ohm.	Resistance. Ohms per Lb.	Resistance. Ohms per Foot.	Gauge Number.
0000	.454	206116	.623925	12486.73	1.0027	19966.55	.00008027	.000050084	0000
000	.425	180625	.54676	9569.13	1.8285	17496.15	.00010450	.000057152	000
00	.38	144400	.437107	6256.59	2.2878	13088.	.00015983	.000071490	00
0	.34	115600	.349928	3918.60	2.8577	11198.17	.00025528	.0000893	0
1	.3	90000	.272435	2375.17	3.6706	8718.3	.00042102	.000114701	1
2	.284	80656	.24415	1952.03	4.0958	7813.15	.00051229	.00012799	2
3	.259	67081	.202964	1318.28	4.927	6495.15	.00075857	.00015396	3
4	.238	56644	.171465	940.862	5.832	5487.11	.00106285	.000182665	4
5	.22	48400	.14651	688.494	6.8255	4688.51	.00145245	.000212795	5
6	.203	41209	.124742	497.631	8.0165	3989.26	.00200029	.00025067	6
7	.18	32400	.098076	307.82	10.1962	3138.59	.00324866	.000318615	7
8	.165	27225	.08241	217.338	12.1345	2637.29	.00460113	.000379178	8
9	.148	21904	.066305	140.754	15.0818	2122.82	.0071046	.000471073	9
10	.134	17956	.054354	94.5433	18.3979	1739.4	.0105772	.00057491	10
11	.12	14400	.04359	60.805	22.941	1394.53	.012446	.00071688	11
12	.109	11881	.035964	43.3914	27.8056	1150.91	.0241596	.000868875	12
13	.095	9025	.027319	23.8837	36.6046	874.252	.0418696	.00114384	13
14	.083	6889	.020853	14.9849	47.9547	667.338	.0718596	.0014985	14
15	.072	5184	.015692	7.88012	63.7267	502.21	.126902	.00199134	15
16	.065	4225	.012789	5.23433	78.1922	409.276	.19105	.00244334	16
17	.058	3364	.01018	3.31726	88.232	325.87	.30144	.0030687	17
18	.049	2401	.007268	1.69043	137.586	232.585	.591566	.0042995	18
19	.042	1764	.005340	.912492	187.266	170.879	1.0959	.0058521	19
20	.035	1225	.003708	.44013	269.687	118.066	2.27266	.008427	20
21	.032	1024	.003099	.307405	322.585	99.195	3.25304	.010087	21
22	.028	784	.002373	.18022	421.407	75.946	5.54876	.0131672	22
23	.025	625	.001892	.114549	528.54	60.5438	8.7299	.016517	23
24	.022	484	.001465	.0686867	682.594	49.885	14.5589	.0213287	24
25	.020	400	.001211	.046924	825.764	38.748	21.3111	.0258078	25
26	.018	324	.0009807	.030786	1019.68	31.386	32.4885	.0318615	26
27	.016	256	.0007749	.0192165	1290.489	24.7987	52.0389	.0403246	27
28	.014	196	.0005933	.0112647	1685.488	18.9866	88.773	.052668	28
29	.013	169	.0005116	.00837544	1954.652	16.371	119.397	.061083	29
30	.012	144	.0004359	.00608035	2294.104	13.949	164.46	.071688	30
31	.01	100	.0003027	.00293266	3303.60	9.687	341.034	.103231	31

Circular Mils.	Amperes. $\sqrt[4]{\left(\frac{C.M.}{104}\right)^3}$	Diameter in Mils. Mil = .001 in.	Weight. Sp. gr. 8.889. Lbs. per Foot.	Weight. Sp. gr. 8.889. Lbs. per Ohm.	Length. Feet per Lb.	Length. Feet per Ohm.	Resistance. Legal Ohms at 75° Fahr. Ohms per Lb.	Resistance. Legal Ohms at 75° Fahr. Ohms per Ft.	E. S. G. Gauge Number.
3000	12.5	54.78	.009084	2.597	110.087	285.9	.3850405	.003497600	3
5000	18.3	70.72	.015139	7.214	66.054	476.5	.1386225	.002098640	5
8000	26.0	89.45	.024220	18.464	41.288	762.3	.0651602	.001311780	8
12000	35.2	109.55	.036328	41.538	27.527	1143.4	.0240743	.000874578	12
15000	41.6	122.48	.045410	64.902	22.022	1429.2	.0154178	.000699663	15
20000	51.6	141.43	.060548	115.372	16.516	1905.7	.0086664	.000524745	20
25000	61.0	158.12	.075682	180.278	13.213	2382.0	.0055470	.000419807	25
30000	70.0	173.21	.090817	259.722	11.011	2859.9	.0038522	.000349840	30
35000	78.6	187.09	.105955	353.340	9.4381	3334.9	.0028301	.000299863	35
40000	86.8	200.00	.121082	461.440	8.2589	3811.0	.0021671	.000262400	40
45000	94.9	212.14	.136227	584.098	7.3407	4287.7	.0017120	.000233227	45
50000	102.7	223.61	.151357	721.026	6.6069	4763.8	.0013868	.000209914	50
55000	110.3	234.53	.166501	872.547	6.0060	5240.5	.0011467	.000190821	55
60000	117.7	244.95	.181625	1038.258	5.5059	5716.5	.00096315	.000174931	60
65000	125.0	254.96	.196772	1218.586	5.0820	6192.9	.00082057	.000161465	65
70000	132.1	264.58	.211901	1413.264	4.7192	6669.4	.00070758	.000149937	70
75000	139.1	273.87	.227043	1622.457	4.4044	7146.0	.00061635	.000139938	75
80000	146.0	282.85	.242176	1845.952	4.1292	7622.3	.00054172	.000131193	80
85000	152.8	291.55	.257303	2083.759	3.8865	8098.4	.00047990	.000123480	85
90000	159.5	300.00	.272434	2336.405	3.6706	8574.7	.00042807	.000116622	90
95000	166.1	308.23	.287587	2603.046	3.4773	9051.6	.00038415	.000110477	95
100000	172.6	316.23	.302709	2884.082	3.3035	9527.6	.00034673	.000104960	100
110000	185.4	331.67	.332991	3489.958	3.0031	10480.6	.00028656	.000095410	110
120000	198.0	346.42	.363267	4153.433	2.7528	11433.6	.00024070	.000084460	120
130000	210.2	360.56	.393527	4874.226	2.5411	12386.0	.00020514	.000080730	130
140000	222.2	374.17	.423797	5652.899	2.3596	13338.7	.00017690	.000074970	140
150000	234.0	387.30	.454061	6484.573	2.2023	14291.3	.00015409	.000069997	150
160000	245.6	400.00	.484328	7383.042	2.0647	15243.9	.00013544	.000065600	160
170000	257.0	412.32	.514622	8335.525	1.9432	16197.4	.00011995	.000061735	170
180000	268.3	424.27	.544884	9344.686	1.8353	17149.9	.00010701	.000058309	180
190000	279.4	435.89	.575140	10411.241	1.7387	18102.1	.00009604	.000055242	190
200000	290.4	447.22	.605427	11536.681	1.6517	19055.4	.00008667	.000052478	200
220000	312.0	469.05	.665975	13959.567	1.5016	20961.1	.00007163	.000047707	220
240000	333.0	489.90	.726498	16612.114	1.3765	22866.0	.00006019	.000043733	240
260000	353.5	509.91	.787058	19496.997	1.2706	24772.1	.00005129	.000040368	260
280000	373.7	529.16	.847605	22612.233	1.1798	26677.8	.00004422	.000037484	280
300000	393.6	547.73	.908140	25957.464	1.1012	28583.1	.00003852	.000034986	300
320000	413.1	565.69	.968672	29533.696	1.0323	30488.3	.00003386	.000032799	320
340000	432.3	583.10	1.029214	33340.181	.9716	32393.8	.00002999	.000030870	340
360000	451.3	600.00	1.089738	37376.652	.9177	34298.7	.00002675	.000029155	360

1 Mil Foot = 9.718 B. A. Units at 0° C. (Dr. Matthiessen.)

TABLE OF DIMENSIONS, WEIGHT, AND RESISTANCE OF COPPER WIRE.
(Birmingham Gauge.)

Gauge Number.	Diameter, Inch.	Sectional Area in Circular Mils. = diam².	Weight. Lbs. per Foot.	Weight. Lbs. per Ohm.	Length. Feet per Lb.	Length. Feet per Ohm.	Resistance. Ohms per Lb.	Resistance. Ohms per Foot.	Gauge Number.
	.454	206116	.623925	12486.73	1.0027	19966.55	.00008027	.000050084	0000
	.425	180625	.54676	9569.13	1.8285	17496.15	.00010450	.000057152	000
	.38	144400	.437107	6256.59	2.2878	13088.	.00015983	.000071490	00
	.34	115600	.349928	3918.60	2.8577	11198.17	.00025528	.0000893	0
1	.3	90000	.272435	2375.17	3.6706	8718.3	.00042102	.000114701	1
2	.284	80656	.24415	1952.03	4.0958	7813.15	.00051229	.00012799	2
3	.259	67081	.202964	1318.28	4.927	6495.15	.00075857	.00015396	3
4	.238	56644	.171465	940.862	5.832	5487.11	.00106285	.000182665	4
5	.22	48400	.14651	688.494	6.8255	4688.51	.00145245	.000212795	5
6	.203	41209	.124742	497.631	8.0165	3989.26	.00200029	.00025067	6
7	.18	32400	.098076	307.82	10.1962	3138.59	.00324866	.000318615	7
8	.165	27225	.08241	217.338	12.1345	2637.29	.00460113	.000379178	8
9	.148	21904	.066305	140.754	15.0818	2122.82	.0071046	.000471073	9
10	.134	17956	.054354	94.5433	18.3979	1739.4	.0105772	.00057491	10
11	.12	14400	.04359	60.805	22.941	1394.53	.012446	.00071688	11
12	.109	11881	.035964	43.3914	27.8056	1150.91	.0241596	.000868875	12
13	.095	9025	.027319	23.8837	36.6046	874.252	.0418696	.00114384	13
14	.083	6889	.020853	14.9849	47.9547	667.338	.0718596	.0014985	14
15	.072	5184	.015692	7.88012	63.7267	502.21	.126902	.00199134	15
16	.065	4225	.012789	5.23433	78.1922	409 276	.19105	.00244334	16
17	.058	3364	.01018	3.31726	88.232	325.87	.30144	.0030687	17
18	.049	2401	.007268	1.69043	137.586	232.585	.591566	.0042995	18
19	.042	1764	.005340	.912492	187.266	170.879	1.0959	.0058521	19
20	.035	1225	.003708	.44013	269.687	118.066	2.27266	.008427	20
21	.032	1024	.003099	.307405	322.585	99.195	3.25304	.010087	21
22	.028	784	.002373	.18022	421.407	75.946	5.54876	.0131672	22
23	.025	625	.001892	.114549	528.54	60.5438	8.7299	.016517	23
24	.022	484	.001465	.0686867	682.594	49.885	14.5589	.0213287	24
25	.020	400	.001211	.046924	825.764	38.748	21.3111	.0258078	25
26	.018	324	.0009807	.030786	1019.68	31.386	32.4885	.0318615	26
27	.016	256	.0007749	.0192165	1290.489	24.7987	52.0389	.0403246	27
28	.014	196	.0005933	.0112647	1685.488	18.9866	88.773	.052668	28
29	.013	169	.0005116	.00837544	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

HARD-DRAWN COPPER TELEGRAPH WIRE.

(J. A. Roebling's Sons Co.)

nished in half-mile coils, either bare or insulated.

. & S. ge.	Resistance in Ohms per Mile.	Breaking Strength.	Weight per Mile.	Approximate Size of E. B. B. Iron Wire equal to Copper. (Iron-wire Gauge)
	4.30	625	209	2
	5.40	525	166	3
	6.90	420	131	4
	8.70	330	104	6
	10.90	270	83	6½
	13.70	213	66	8
	17.40	170	52	9
	22.10	130	41	10

ndling this wire the greatest care should be observed to avoid kinks, scratches, or cuts. Joints should be made only with McIntire Con- s.

ccount of its conductivity being about five times that of Ex. B. B. ire, and its breaking strength over three times its weight per mile, may be used of which the section is smaller and the weight less than ivalent iron wire, allowing a greater number of wires to be strung on es.

les this advantage, the reduction of section materially decreases the static capacity, while its non-magnetic character lessens the self-in- of the line, both of which features tend to increase the possible of signalling in telegraphing, and to give greater clearness of enunci- ver telephone lines, especially those of great length.

INSULATED COPPER WIRES.

Weight per 1000 feet.

Weather-proof Line Wire.	Underwriters' Line Wire.	B. & S. Gauge.	Weather-proof Line Wire.	Underwriters' Line Wire.	B. & S. Gauge.	Weather-proof Line Wire.	Underwriters' Line Wire.
671.	701.	5	115.	121.	13	26.	26.5
537.	565.	6	93.	99.	14	20.5	22.
426.	447.	7	77.	80.	15	17.	20.
342.	364.	8	64.	67.	16	14.	15.
274.	294.	9	53.	54.	17	12.	13.
220.	241.	10	44.	45.	18	10.75	11.
178.	185.	11	37.	37.	19	9.	10.
141.	147.	12	30.	31.	20	7.5	8.

D-ENCASED ANTI-INDUCTION TELEPHONE AND TELEGRAPH CABLES. (Roebling's.)

N CABLES, LEAD ENCASED.		FOR METALLIC CIRCUIT.		FOR TELEGRAPH CIRCUITS.	
f .	Size Wire B. & S. Gauge.	No. of Pairs.	Size Wire B. & S. Gauge.	No. of Wires.	Size Wire B. & S. Gauge.
	18	5	18	3	14
	18	15	18	4	14
	18	25	18	7	14
	18	50	18	10	14
	18	75	18	20	14
				50	14
				100	14

FLEXIBLE CABLES.

Area Circ. Mils.	No. of Wires.	Size Wire B. & S. Gauge.	Approximate Size of Equivalent Solid Wire.	Area Circ. Mils.	No. of Wires.	Size Wire B. & S. Gauge.	Diameter of Equivalent Solid Wire, Mils.
15699.6	49	25	8 B. & S.	272410.6	133	17	522.
24963.0	49	23	6 "	433154.4	133	15	658.
39693.9	49	21	4 "	688727.2	133	13	830.
63116.9	49	19	2 "	868476.7	133	12	932.
				1095155.3	133	11	1046.
				210964.6	103	17	459.
				420127.2	129	15	649.
				657656.8	127	13	811.
				835827.2	128	12	914.
				1062198.9	129	11	1035.

WEATHERPROOF AERIAL CABLES.

No. of Conductors.	Weight per Conductor per 1000 feet.	No. of Conductors.	Weight per Conductor per 1000 feet.	No. of Conductors.	Weight per Conductor per 1000 feet.
1	10.75 lbs.	8	9.25 lbs.	15	9.25 lbs.
2	18.00 "	9	9.25 "	16	9.25 "
3	13.00 "	10	9.25 "	17	9.25 "
4	10.75 "	11	9.25 "	18	9.25 "
5	10.00 "	12	9.25 "	19	9.25 "
6	9.50 "	13	9.25 "	20	9.25 "
7	9.25 "	14	9.25 "		

LEAD-ENCASED ELECTRIC-LIGHT CABLES.

Single Wires.

(J. A. Roebling's Sons Co.)

Size, B. & S. Gauge.	Diameter of Solid Copper Wire. Mils.	Area. Circular Mils.	Nearest Approximate Birmingham Wire-gauge No.	Approximate Weight per Foot of Cable. Oz.	Approximate Diameter of Cable. Mils.
20	31.96	1021.	21	1.63	170
19	35.39	1252.	20	1.70	175
18	40.30	1624.	19	1.75	180
17	45.25	2048.	18½	1.84	185
16	50.82	2583.	18	3.00	245
15	57.07	3257.	17	3.20	250
14	64.08	4107.	16	3.38	255
13	71.96	5178.	15	3.56	265
12	80.80	6530.	14	5.00	310
11	90.74	8234.	13½	5.23	320
10	101.89	10381.	12½	5.68	335
9	114.23	13094.	11½	5.95	345
8	128.49	16509.	10½	6.35	360
7	144.28	20816.	9	6.90	375

As tested by the Bell Telephone Co. of Philadelphia, the insulation ma be stated at 2000 megohms per mile, with an electrostatic capacity of microfarad.

GALVANIZED STEEL-WIRE STRAND.

For Smokestack Guys, Signal Strand, etc.

(J. A. Roebling's Sons Co.)

s strand is composed of 7 wires, twisted together into a single strand.

Diameter.	Weight per 100 Feet.	Estimated Breaking Strength.	7 Wires.	Diameter.	Weight per 100 Feet.	Estimated Breaking Strength.
in.	lbs.	lbs.		in.	lbs.	lbs.
½	52	8,320	No. 15	¼	10	1,600
15-32	42	6,720	16	7-32	8	1,280
7-16	36	5,720	17	3-16	6	960
⅜	29	4,640	18	11-64	4 3-10	688
5-16	21	3,360	19	9-64	3 3-10	528
9-32	16	2,560	20	⅛	2 4-10	384
17-64	12	1,920	21	3-32	2	320

special purposes these strands can be made of 50 to 100 per cent r tensile strength. When used to run over sheaves or pulleys the use -iron stock is advisable.

EXIBLE STEEL-WIRE CABLES FOR VESSELS.

(Trenton Iron Co., 1886.)

numerous disadvantages, the system of working ships' anchors with ables is still in vogue. A heavy chain cable contributes to the hold-wer of the anchor, and the facility of increasing that resistance by out the cable is prized as an advantage. The requisite holding- is obtained, however, by the combined action of a comparatively nchor and a correspondingly great mass of chain of little service in tion to its weight or to the weight of the anchor. If the weight and the anchor were increased so as to give the greatest holding-power d, and it were attached by means of a light wire cable, the combined of the cable and anchor would be much less than the total weight of in and anchor, and the facility of handling would be much greater. shipbuilders have taken the initiative in this direction, and many of gest and most serviceable vessels afloat are fitted with steel-wire They have given complete satisfaction.

Trenton Iron Co.'s cables are made of crucible cast-steel wire, and teed to fulfil Lloyd's requirements. They are composed of 72 wires ided into six strands of twelve wires each. In order to obtain great lity, hempen centres are introduced in the strands as well as in the eted cable.

FLEXIBLE STEEL-WIRE HAWSERS.

e hawsers are extensively used. They are made with six strands of wires each, hemp centres being inserted in the individual strands as in the completed rope. The material employed is crucible cast steel, ized, and guaranteed to fulfil Lloyd's requirements. They are only rd the weight of hempen hawsers; and are sufficiently pliable to work any bitts to which hempen rope of equivalent strength can be applied.

h tarred Russian hemp hawser weighs about 39 lbs. per fathom.

ch white manila *hawser weighs about 20 lbs.* per fathom.

ch stud chain weighs about 68 lbs. per fathom.

galvanized steel hawser weighs about 12 lbs. per fathom.

f the above named has about the same tensile strength.

SPECIFICATIONS FOR GALVANIZED IRON WIRE.

Issued by the British Postal Telegraph Authorities.

Weight per Mile.			Diameter.			Tests for Strength and Ductility.						Resistance per Mile of the Standard Size at 60° Fahr.
Required Standard.	Allowed.		Required Standard.	Allowed.		Breaking Weight.	No. of Twists in 6 in.	For Breaking Weight not less than—	No. of Twists in 6 in.	For Breaking Weight not less than—	No. of Twists in 6 in.	
	Minimum.	Maximum.		Minimum.	Maximum.	Minimum.	Minimum.		Minimum.		Minimum.	Maximum.
lbs.	lbs.	lbs.	mils.	mils.	mils.	lbs.		lbs.		lbs.		ohms.
800	767	833	242	237	247	2480	15	2550	14	2620	13	6.75
600	571	629	209	204	214	1860	17	1910	16	1960	15	9.00
450	424	477	181	176	186	1390	19	1425	18	1460	17	12.00
400	377	424	171	166	176	1240	21	1270	20	1300	19	13.50
200	190	213	121	118	125	620	30	638	28	655	26	27.00

STRENGTH OF PIANO-WIRE.

The average strength of English piano-wire is given as follows by We-ster, Horsfals & Lean:

Numbers in Music-wire Gauge.	Equivalents in Fractions of Inches in Diameters.	Ultimate Tensile Strength in Pounds.	Numbers in Music-wire Gauge.	Equivalents in Fractions of inches in Diameters.	Ultimate Tensile Strength in Pounds.
12	.029	225	18	.041	395
13	.031	250	19	.043	425
14	.033	285	20	.045	500
15	.035	305	21	.047	540
16	.037	340	22	.052	650
17	.039	360			

These strengths range from 300,000 to 340,000 lbs. per sq. in. The compo-sition of this wire is as follows: Carbon, 0.570; silicon, 0.090; sulphur, 0.0 phosphorus, 0.018; manganese, 0.425.

"PLOUGH"-STEEL WIRE.

The term "plough," given in England to steel wire of high quality, w derived from the fact that such wire is used for the construction of rop used for ploughing purposes. It is to be hoped that the term will not used in this country, as it tends to confusion of terms. Plough-steel known here in some steel-works as the quality of plate steel used for mould-boards of ploughs, for which a very ordinary grade is good enough.

Experiments by Dr. Percy on the English plough-steel (so-called) gave following results: Specific gravity, 7.814; carbon, 0.828 per cent; man-nese, 0.587 per cent; silicon, 0.143 per cent; sulphur, 0.009 per cent; pho-phorus, nil; copper, 0.030 per cent. No traces of chromium, titanium, tungsten were found. The breaking strains of the wire were as follows:

Diameter, inch	.093	.132	.159	.191
Poun[illegible] inch	314,960	257,600	224,000	201,600

[illegible] uly from 0.75 to 1.1 per cent.

ES OF DIFFERENT METALS AND ALLOYS.

(J. Bucknall Smith's Treatise on Wire.)

Wire is commonly composed of an alloy of 1 3/4 to 2 parts of 1 part of zinc. The tensile strength ranges from 20 to 40 tons per ch, increasing with the percentage of zinc in the alloy.

n or Nickel Silver, an alloy of copper, zinc, and nickel, is brass whitened by the addition of nickel. It has been drawn into e as .002″ diam.

um wire may be drawn into the finest sizes. On account of its its use is practically confined to special scientific instruments and appliances in which resistances to high temperature, oxygen, and essential. It expands less than other metals when heated, which permits its being sealed in glass without fear of cracking. It is used in incandescent electric lamps.

hor-bronze Wire contains from 2 to 6 per cent of tin and to 1/8 per cent of phosphorus. The presence of phosphorus is al to electric conductivity.

a-metal" wire is made from an alloy of copper, iron, and zinc. h ranges from 45 to 62 tons per square inch. It is used for some ire rope, also for wire gauze. It is not subject to deposits of ver- has great toughness, even when its tensile strength is over 60 quare inch.

inum Wire. — Specific gravity .268. Tensile strength only ons per square inch. It has been drawn as fine as 11,400 yards to or .042 grains per yard.

inum Bronze, 90 copper, 10 aluminum, has high strength and is inoxidizable, sonorous. Its electric conductivity is 12.6 per cent pure copper.

n Bronze, patented in 1882 by L. Weiler of Paris, is made as luosilicate of potash, pounded glass, chloride of sodium and cal- bonate of soda and lime, are heated in a plumbago crucible, and reaction takes place the contents are thrown into the molten be treated. Silicon-bronze wire has a conductivity of from 40 to t of that of copper wire and four times more than that of iron, tensile strength is nearly that of steel, or 28 to 55 tons per square ection. The conductivity decreases as the tensile strength in- Wire whose conductivity equals 95 per cent of that of pure copper nsile strength of 28 tons per square inch, but when its conductivity ent of pure copper, its strength is 50 tons per square inch. It is ely used for telegraph wires. It has great resistance to oxidation.

ary Drawn and Annealed Copper Wire has a strength to 20 tons per square inch.

CIFICATIONS FOR HARD-DRAWN COPPER WIRE.

tish Post Office authorities require that hard-drawn copper wire o them shall be of the lengths, sizes, weights, strengths, and con- s as set forth in the annexed table.

per Statute Mile.		Approximate Equivalent Diameter.			Minimum Breaking Weight.	Minimum No. of Twists in 3 Inches.	Maximum Resistance per Mile of Wire (when hard) at 60° Fahr.	Minimum Weight of each Piece (or Coil) of Wire.
Minimum.	Maximum.	Standard.	Minimum.	Maximum.				
lbs.	lbs.	mils.	mils.	mils.	lbs.		ohms.	lbs.
97½	102½	79	78	80	330	30	9.10	50
149¾	153¾	97	95½	98	490	25	6.05	50
196	203	112	110½	113¼	650	20	4.53	10
390	410	158	155½	160¼	1300	10	2.27	[illegible]

WIRE ROPES.

List adopted by manufacturers in 1892. See pamphlets of Tre[illegible] Co., John A. Roebling's Sons Co., and other makers.

Pliable Hoisting Rope.

With 6 strands of 19 wires each.

Trade Number.	Diameter.	Circumference in inches.	Weight per foot in pounds. Rope with Hemp Centre.	Breaking Strain, tons of 2000 lbs.	Proper Working Load in tons of 2000 lbs.	Circumference of new Manilla Rope of equal Strength.
IRON.						
1	2¼	6¾	8.00	74	15	14
2	2	6	6.30	65	13	13
3	1¾	5½	5.25	54	11	12
4	1⅝	5	4.10	44	9	11
5	1½	4¾	3.65	39	8	10
5½	1⅜	4⅜	3.00	33	6½	9½
6	1¼	4	2.50	27	5½	8½
7	1⅛	3½	2.00	20	4	7½
8	1	3⅛	1.58	16	3	6½
9	⅞	2¾	1.20	11.50	2½	5½
10	¾	2¼	0.88	8.64	1¾	4¾
10¼	⅝	2	0.60	5.13	1¼	3¾
10½	9-16	1⅝	0.44	4.27	¾	3½
10¾	½	1½	0.35	3.48	½	3
10a	7-16	1⅜	0.29	3.00	⅜	2¾
10⅞	⅜	1¼	0.26	2.50	¼	2½
CAST STEEL.						
1	2¼	6¾	8.00	155	31	....
2	2	6	6.30	125	25	....
3	1¾	5½	5.25	106	21	....
4	1⅝	5	4.10	86	17	15
5	1½	4¾	3.65	77	15	14
5½	1⅜	4⅜	3.00	63	12	13
6	1¼	4	2.50	52	10	12
7	1⅛	3½	2.00	42	8	11
8	1	3⅛	1.58	33	6	9½
9	⅞	2¾	1.20	25	5	8½
10	¾	2¼	0.88	18	3½	7
10¼	⅝	2	0.60	12	2½	5¾
10½	9-16	1⅝	0.44	9	1½	5
10¾	½	1½	0.35	7	1	4½
10a	7-16	1⅜	0.29	5½	¾	3¾
10⅞	⅜	1¼	0.26	4½	½	3½

Cable-Traction Ropes.

According to English practice, cable-traction ropes, of about 3[illegible] circumference, are commonly constructed with six strands of seve[illegible] teen wires, the lays in the strands varying from, say, 3 in. to 3½ in., [illegible] lays in the ropes from, say, 7½ in. to 9 in. In the United States, b[illegible] strands of nineteen wires are generally preferred as being more f[illegible] but, on the other hand, the smaller external wires wear out more [illegible] The Market street Street Railway Company, San Francisco, has use[illegible] 1¼ in. in diameter, composed of six strands of nineteen steel wires, w[illegible] [illegible] lbs. per foot, the longest continuous length being 24,125 ft. The C[illegible] [illegible] [illegible]pany has employed cables of identical constructi[illegible] [illegible],700 ft. On the New York and Brooklyn Bridg[illegible] [illegible] 1,500 ft. long, containing 114 wires, have bee[illegible]

Transmission and Standing Rope.

With 6 strands of 7 wires each.

IRON.

Diameter.	Circumference.	Weight per foot in pounds of Rope with Hemp Centre.	Breaking Strain in tons of 2000 lbs	Proper Working Load in tons of 2000 lbs.	Circumference of new Manila Rope of equal Strength.	Min Size of Drum or Sheave in feet.
1½	4⅝	3.37	36	9	10	13
1⅜	4¼	2.77	30	7½	9	12
1¼	3¾	2.28	25	6¼	8½	10¾
1⅛	3⅜	1 82	20	5	7½	9½
1	3	1.50	16	4	6½	8½
⅞	2⅝	1.12	12.3	3	5¾	7½
¾	2⅜	0.88	8.8	2¼	4¾	6¾
11-16	2⅛	0.70	7.6	2	4½	6
⅝	1⅞	0.57	5.8	1½	4	5¼
9-16	1⅝	0.41	4.1	1	3¼	4½
½	1⅜	0.31	2.83	¾	2¾	4
7-16	1¼	0 23	2.13	½	2½	3¼
⅜	1⅛	0.19	1.65	....	2¼	2¾
5-16	1	0.16	1.38	....	2	2½
9-32	⅞	0.125	1.03	....	1¾	2¼

CAST STEEL.

Diameter.	Circumference.	Weight per foot in pounds of Rope with Hemp Centre.	Breaking Strain in tons of 2000 lbs	Proper Working Load in tons of 2000 lbs.	Circumference of new Manila Rope of equal Strength.	Min Size of Drum or Sheave in feet.
1½	4⅝	3.37	62	13	13	8½
1⅜	4¼	2.77	52	10	12	8
1¼	3¾	2.28	44	9	11	7¼
1⅛	3⅜	1.82	36	7½	10	6¼
1	3	1.50	30	6	9	5¾
⅞	2⅝	1.12	22	4½	8	5
¾	2⅜	0.88	17	3½	7	4½
11-16	2⅛	0.70	14	3	6	4
⅝	1⅞	0.57	11	2¼	5½	3½
9-16	1⅝	0.41	8	1¾	4¾	3
½	1⅜	0.31	6	1½	4	2½
7-16	1¼	0.23	4½	1¼	3½	2¼
⅜	1⅛	0.19	4	1	3¼	2
5-16	1	0.16	3	¾	2¾	1¾
9-32	⅞	0.12	2	½	2¼	1½

Plough-Steel Rope.

[illegible] ropes of very high tensile strength, which are ordinarily called [illegible]ch-steel Ropes," are made of a high grade of crucible steel, which, [illegible] in the form of wire, will bear a strain of from 100 to 150 tons per [illegible] inch.

[illegible] it is necessary to use very long or very heavy ropes, a reduction of [illegible] weight of ropes becomes a matter of serious consideration.

[illegible] advisable to reduce all bends to a minimum, and to use somewhat [illegible] drums or sheaves than are suitable for an ordinary crucible rope hav[illegible] strength of 60 to 80 tons per square inch. Before using Plo[illegible] *it is best to have advice on the subject of adaptability.*

Plough-Steel Rope.

With 6 strands of 19 wires each.

Trade Number.	Diameter in inches.	Weight per foot in pounds.	Breaking Strain in tons of 2000 lbs.	Proper Working Load.	Min. Dru She f
1	2¼	8.00	240	46	
2	2	6.30	189	37	
3	1¾	5.25	157	31	
4	1⅝	4.10	123	25	
5	1½	3.65	110	22	
5½	1⅜	3.00	90	18	
6	1¼	2.50	75	15	
7	1⅛	2.00	60	12	
8	1	1.58	47	9	
9	⅞	1.20	37	7	
10	¾	0.88	27	5	
10¼	⅝	0.60	18	3½	
10½	9-16	0.44	13	2	
10¾	½	0.35	10	1½	
With 7 Wires to the Strand.					
15	1	1.50	45	9	
16	⅞	1.12	33	6½	
17	¾	0.88	25	4½	
18	11-16	0.70	21	4	
19	⅝	0.57	16	3¾	
20	9-16	0.41	12	2	
21	½	0.31	9	1½	
22	7-16	0.23	5	¾	
23	⅜	0.19	4	¼	

Galvanized Iron Wire Rope.

For Ships' Rigging and Guys for Derricks.

CHARCOAL ROPE.

Circumference in inches.	Weight per Fathom in pounds.	Cir. of new Manila Rope of equal Strength.	Breaking Strain in tons of 2000 pounds	Circumference in inches	Weight per Fathom in pounds.	Cir. of new Manila Rope of equal Strength.
5½	26½	11	43	2½	5½	5
5¼	24½	10½	40	2¼	4½	4¾
5	22	10	35	2	3½	4½
4¾	21	9½	33	1¾	2½	3¾
4½	19	9	30	1½	2	3
4¼	16½	8½	26	1¼	1¾	2½
4	14¼	8	23	1⅛	1¼	2½
3¾	12¾	7½	20	1	⅞	2¼
3½	10¾	6½	16	⅞	¾	1¾
3¼	9½	6	14	¾	½	1½
3	8	5¾	12	⅝	⅜	1¼
2¾		5¼	10	½	¼	[illegible]

Galvanized Cast-steel Yacht Rigging.

…rcum-…rence …nches.	Weight per Fathom in pounds.	Cir. of new Manilla Rope of equal Strength.	Breaking Strain in tons of 2000 pounds	Circumference in inches	Weight per Fathom in pounds.	Cir. of new Manilla Rope of equal Strength.	Breaking Strain in tons of 2000 pounds
…	14¼	13	66	2	3½	6½	14
…½	10¾	11	43	1¾	2½	5¼	10
…	8	9½	32	1½	2	4¾	8
…¾	6¾	8½	27	1⅜	1⅞	4¼	6½
…½	5½	8	22	1¼	1¾	3¾	5½
…¼	4½	7	18	1	⅞	3	3½

Steel Hawsers.

For Mooring, Sea, and Lake Towing.

…mfer-…ce.	Breaking Strength.	Size of Manilla Hawser of equal Strength.	Circumference.	Breaking Strength.	Size of Manilla Hawser of equal Strength.
…hes.	Tons.	Inches.	Inches.	Tons.	Inches.
…½	15	6½	3½	29	9
…¾	18	7	4	35	10
3	22	8½			

Steel Flat Ropes.

(J. A. Roebling's Sons Co.)

…el-wire Flat Ropes are composed of a number of strands, alternately …ted to the right and left, laid alongside of each other, and sewed together … soft iron wires. These ropes are used at times in place of round ropes …e shafts of mines. They wind upon themselves on a narrow winding-…n, which takes up less room than one necessary for a round rope. The …iron sewing-wires wear out sooner than the steel strands, and then it …mes necessary to sew the rope with new iron wires.

…th and …ckness …nches.	Weight per foot in pounds.	Strength in pounds.	Width and Thickness in inches.	Weight per foot in pounds.	Strength in pounds.
…×2	1.19	35,700	½×3	2.38	71,400
…×2½	1.86	55.800	½×3½	2.97	89,000
…×3	2.00	60,000	½×4	3.30	99,000
…×3½	2.50	75.000	½×4½	4.00	120,000
…×4	2.86	85,800	½×5	4.27	128,000
…×4½	3.12	93,600	½×5½	4.82	144,600
…×5	3.40	100,000	½×6	5.10	153,000
…×5½	3.90	110,000	½×7	5.90	177,000

… safe working load allow from one fifth to one seventh of the breaking …s.

"Lang Lay" Rope.

…ire rope, as ordinarily made, the component strands are laid up into …n a direction opposite to that in which the wires are laid into strands; …, if the wires in the strands are laid from right to left, the strands are …to rope from left to right. In the "Lang Lay," sometimes known as …versal Lay," *the wires are laid into strands* and the strands into rope *…same direction; that is, if the wire is laid in the strands from right to …strands are also laid into rope from right to left.* Its use has been …sirable under certain conditions and for certain purposes, mostly …ge plants, inclined planes, and street railway cables, although it …en used for vertical hoists in mines, etc. Its advantages are tha…

GALVANIZED STEEL CABLES.

For Suspension Bridges. (Roebling's.)

Diameter in inches.	Ultimate Strength in tons of 2000 pounds.	Weight per foot.	Diameter in inches.	Ultimate Strength in tons of 2000 pounds.	Weight per foot.	Diameter in inches.	Ultimate Strength in tons of 2000 pounds.
2⅝	220	13	2¼	155	8.64	1¾	95
2½	200	11.3	2	110	6.5	1⅝	75
2⅜	180	10	1⅞	100	5.8	1½	65

COMPARATIVE STRENGTHS OF FLEXIBLE (VANIZED STEEL-WIRE HAWSERS,

With Chain Cable, Tarred Russian Hemp, and Manila Ropes. (Trenton Iron Co.)

Patent Flexible Steel-wire Hawsers and Cables.				Chain Cable.				Tarred Russian Hemp Rope.			V M R
Size, Circumference.	Weight per Fathom.	Guaranteed Breaking Strain.	Diameter of Barrel or Sheave round which it may be worked.	Size.	Weight per Fathom.	Proof Strain.	Breaking Strain.	Size.	Weight per Fathom.	Breaking Strain.	Size.
1	¾	1¾	6					2¾		1½	2½
1¼	1	2½	7½	½	14	4½	6	3½	3	2¼	3
1½	1¾	4	9					4	3½	3½	3¼
1¾	2	5½	10½	9-16	17	5½	7¼	5	6	5	4
2	2¾	7	12					5¾	8	7	5
2¼	3¾	9	13½	10-16	21	7	9½	6½	10	9	5¾
2½	4½	12	15					7½	13	11½	6¼
2¾	5½	15	16½	11-16	25	8½	12¾	8½	16	14	7
3	7	18	18	12-16	30	10⅛	15⅛	9	19	16½	7½
3¼	8	22	19½	13-16	35	11⅞	17 8-10	10	23	20	8¼
3½	9	26	21	15-16	48	15 8-10	23 7-10	11	28	24½	9
4	12	33	24	1	54	18	27	12	33	29	10
4½	15	39	27	1⅛	68	22¾	34⅛	13	39	34	11
5	23½	64	30	1 17-32	112	37⅞	55½	15	56	50	12¾
5½	28	74	33	1⅝	143	47½	66½	17	67	60	13½
6	33	88	36	1¾	166	55⅛	77¼	19	84	72	15
6½	37	102	39	1 15-16	204	67½	94⅛	21	106	89	
7	41	116	42	2 1-16	231	76⅛	107 1-10	23	123	106	
7½	47	130	45	2 3-16	256	86⅞	120⅛	24	134	115	
8	53	150	48	2 5-16	280	96¼	134¾	25	146	125	

MAGNESIA BRICKS.

"Foreign Abstracts" of the Institution of Civil Engineers, 1893, gives a paper by C. Bischof on the production of magnesia bricks. The material most in favor at present is the magnesite of Styria, which, although less pure considered as a source of magnesia than the Greek, has the property of fritting at a high temperature without melting. The composition of the two substances, in the natural and burnt states, is as follows:

Magnesite.	Styrian.	Greek.
Carbonate of magnesia	90.0 to 96.0%	94.46%
" " lime	0.5 to 2.0	4.49
" " iron	3.0 to 6.0	FeO 0.08
Silica	1.0	0.52
Manganous oxide	0.5	Water 0.54
Burnt Magnesite.		
Magnesia	77.6	82.46—95.36
Lime	7.3	0.83—10.92
Alumina and ferric oxide	13.0	0.56— 3.54
Silica	1.2	0.73— 7.98

At a red heat magnesium carbonate is decomposed into carbonic acid and caustic magnesia, which resembles lime in becoming hydrated and recarbonated when exposed to the air, and possesses a certain plasticity, so that it can be moulded when subjected to a heavy pressure. By long-continued or stronger heating the material becomes dead-burnt, giving a form of magnesia of high density, sp. gr. 3.8, as compared with 3.0 in the plastic form, which is unalterable in the air but devoid of plasticity. A mixture of two volumes of dead-burnt with one of plastic magnesia can be moulded into bricks which contract but little in firing. Other binding materials that have been used are: clay up to 10 or 15 per cent; gas-tar, perfectly freed from water, soda, silica, vinegar as a solution of magnesium acetate which is readily decomposed by heat, and carbolates of alkalies or lime. Among magnesium compounds a weak solution of magnesium chloride may also be used. For setting the bricks lightly burnt, caustic magnesia, with a small proportion of silica to render it less refractory, is recommended. The strength of the bricks may be increased by adding iron, either as oxide or silicate. If a porous product is required, sawdust or starch may be added to the mixture. When dead-burnt magnesia is used alone, soda is said to be the best binding material.

See also papers by A. E. Hunt, Trans. A. I. M. E., xvi, 720, and by T. Egleston, Trans. A. I. M. E., xiv, 458.

Asbestos.—J. T. Donald, *Eng. and M. Jour.*, June 27, 1891.

ANALYSIS.

	Italian.	Canadian. Broughton.	Canadian. Templeton.
Silica	40.30%	40.57%	40.52%
Magnesia	43.37	41.50	42.05
Ferrous oxide	.87	2.81	1.97
Alumina	2.27	.90	2.10
Water	13.72	13.55	13.46
	100.53	99.33	100.10

Chemical analysis throws light upon an important point in connection with asbestos, i.e., the cause of the harshness of the fibre of some varieties. Asbestos is principally a hydrous silicate of magnesia, i.e., silicate of magnesia combined with water. When harsh fibre is analyzed it is found to contain less water than the soft fibre. In fibre of very fine quality from Black Lake analysis showed *14.38% of water, while* a harsh-fibred sample gave only *11.70%. If soft fibre be heated to a temperature* that will drive off *a portion of the combined water, there* results a substance so brittle that it *may be crumbled between thumb and finger.* There is evidently some con*nection between the consistency of the fibre* and the amount of water in its *composition.*

STRENGTH OF MATERIALS.

Stress and Strain.—There is much confusion among writers on strength of materials as to the definition of these terms. An external force applied to a body, so as to pull it apart, is resisted by an internal force, or resistance, and the action of these forces causes a displacement of the molecules, or deformation. By some writers the external force is called a stress, and the internal force a strain; others call the external force a strain, and the internal force a stress: this confusion of terms is not of importance, as the words stress and strain are quite commonly used synonymously, but the use of the word strain to mean molecular displacement, deformation, or distortion, as is the custom of some, is a corruption of the language. See *Engineering News*, June 23, 1892. Definitions by leading authorities are given below.

Stress.—A stress is a force which acts in the interior of a body, and resists the external forces which tend to change its shape. A deformation is the amount of change of shape of a body caused by the stress. The word strain is often used as synonymous with stress and sometimes it is also used to designate the deformation. (Merriman.)

The force by which the molecules of a body resist a strain at any point is called the stress at that point.

The summation of the displacements of the molecules of a body for a given point is called the distortion or strain at the point considered. (Burr.)

Stresses are the forces which are applied to bodies to bring into action their elastic and cohesive properties. These forces cause alterations of the forms of the bodies upon which they act. *Strain* is a name given to the kind of alteration produced by the stresses. The distinction between stress and strain is not always observed, one being used for the other. (Wood.)

Stresses are of different kinds, viz.: *tensile, compressive, transverse, torsional*, and *shearing* stresses.

A *tensile stress*, or pull, is a force tending to elongate a piece. A *compressive stress*, or push, is a force tending to shorten it. A *transverse stress* tends to bend it. A *torsional stress* tends to twist it. A *shearing stress* tends to force one part of it to slide over the adjacent part.

Tensile, compressive, and shearing stresses are called simple stresses. Transverse stress is compounded of tensile and compressive stresses, and torsional of tensile and shearing stresses.

To these five varieties of stresses might be added *tearing* stress, which is either tensile or shearing, but in which the resistance of different portions of the material are brought into play in detail, or one after the other, instead of simultaneously, as in the simple stresses.

Effects of Stresses.—The following general laws for cases of simple tension or compression have been established by experiment. (Merriman):

1. When a small stress is applied to a body, a small deformation is produced, and on the removal of the stress the body springs back to its original form. For small stresses, then, materials may be regarded as perfectly elastic.

2. Under small stresses the deformations are approximately proportional to the forces or stresses which produce them, and also approximately proportional to the length of the bar or body.

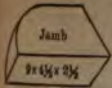
3. When the stress is great enough a deformation is produced which is partly permanent, that is, the body does not spring back entirely to its original form on removal of the stress. This permanent part is termed a set. In such cases the deformations are not proportional to the stress.

4. When the stress is greater still the deformation rapidly increases and the body finally ruptures.

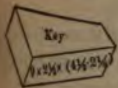
5. A sudden stress, or shock, is more injurious than a steady stress or than a stress gradually applied.

Elastic Limit.—The elastic limit is defined as that point at which the deformations cease to be proportional to the stresses, or, the point at which the rate of stretch (or other deformation) begins to increase. It is also defined as the point at which the first permanent set becomes visible. The last definition is not considered as good as the first, as it is found that with some materials a set occurs with any load, no matter how small, and that with others a set which might be called permanent vanishes with lapse of time, and as it is impossible to get the point of first set without remov-

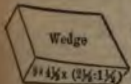
SIZES OF FIRE-BRICK.



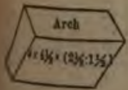
9-inch straight.............. 9 × 4½ × 2½ inches.
Soap.......................... 9 × 2½ × 2½ "
Checker...................... 9 × 3 × 3 "
2-inch........................ 9 × 4½ × 2 "
Split............................ 9 × 4½ × 1¼ "
Jamb.......................... 9 × 4½ × 2½ "
No. 1 key..................... 9 × 2½ thick × 4½ to 4 inches wide.
113 bricks to circle 12 feet inside diam.



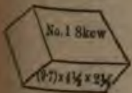
No. 2 key .... 9 × 2½ thick × 4½ to 3½ inches wide.
63 bricks to circle 6 ft. inside diam.
No. 3 key...................... 9 × 2½ thick × 4½ to 3 inches wide.
38 bricks to circle 3 ft. inside diam.



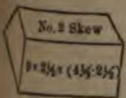
No. 4 key....................... 9 × 2½ thick × 4½ to 2¼ inches wide.
25 bricks to circle 1½ ft. inside diam.
No. 1 wedge (or bullhead). 9 × 4½ wide × 2½ to 2 in. thick, tapering lengthwise.
98 bricks to circle 5 ft. inside diam.



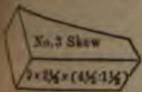
No. 2 wedge................... 9 × 4½ × 2½ to 1½ in. thick.
60 bricks to circle 2½ ft. inside diam.
No. 1 arch..................... 9 × 4½ × 2½ to 2 in. thick, tapering breadthwise.
72 bricks to circle 4 ft. inside diam.
No. 2 arch..................... 9 × 4½ × 2½ to 1½.
42 bricks to circle 2 ft. inside diam.



No. 1 skew..................... 9 to 7 × 4½ to 2½.
Bevel on one end.
No. 2 skew..................... 9 × 2½ × 4½ to 2½.
Equal bevel on both edges.
No. 3 skew..................... 9 × 2½ × 4½ to 1½.
Taper on one edge.

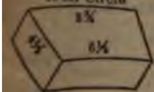


24 inch circle .... 8¼ to 5½ × 4½ × 2½.
Edges curved, 9 bricks line a 24-inch circle.
36-inch circle .... 8¾ to 6½ × 4½ × 2½.
13 bricks line a 36-inch circle.
48-inch circle................. 8¾ to 7¼ × 4½ × 2½.
17 bricks line a 48-inch circle.



13½-inch straight.......... 13½ × 2½ × 6.
13½-inch key No. 1 .... 13½ × 2½ × 6 to 5 inch.
90 bricks turn a 12-ft. circle.
13½-inch key No. 2........ 13½ × 2½ × 6 to 4⅜ inch.
52 bricks turn a 6-ft. circle.
Bridge wall, No. 1.......... 13 × 6½ × 6.
Bridge wall, No. 2.......... 13 × 6½ × 3.

36 in. Circle



Mill tile .... 18, 20, or 24 × 6 × 3.
Stock-hole tiles................ 18, 20, or 24 × 9 × 4.
18-inch block .... 18 × 9 × 6.
Flat back .... 9 × 6 × 2½.
Flat back arch.................. 9 × 6 × 3½ to 2½.
22-inch radius, 56 bricks to circle.
Locomotive tile .... 32 × 10 × 3.
34 × 10 × 3.
34 × 8 × 3.
36 × 8 × 3.
40 × 10 × 3.

Tiles, slabs, and blocks, various sizes 12 to 30 inches long, 8 to 30 inches wide, 2 to 6 inches thick.

Cupola brick, 4 and 6 inches high, 4 and 6 inches radial width, to line shells to 66 in diameter.

A 9-inch straight brick weighs 7 lbs. and contains 100 cubic inches. (=120 per cubic foot. Specific gravity 1.93.)

One cubic *foot of wall requires 17 9-inch bricks*, one cubic yard requires

Where keys, wedges, and other " *shapes* " are used, add 10 p... *mating the number required.*

stresses, the coefficient of elasticity is constant, but beyond the elastic it decreases rapidly.

In cast iron there is generally no apparent limit of elasticity, the defo tions increasing at a faster rate than the stresses, and a permanent set produced by small loads. The coefficient of elasticity therefore is not stant during any portion of a test, but grows smaller as the load incre The same is true in the case of timber. In wrought iron and steel, how there is a well-defined elastic limit, and the coefficient of elasticity w that limit is nearly constant.

Resilience, or Work of Resistance of a Material.—W the elastic limit, the resistance increasing uniformly from zero stress t stress at the elastic limit, the work done by a load applied gradually is to one half the product of the final stress by the extension or other defo tion. Beyond the elastic limit, the extensions increasing more rapidly the loads, and the strain diagram approximating a parabolic form the is approximately equal to two thirds the product of the maximum stre the extension.

The amount of work required to break a bar, measured usually in pounds, is called its resilience; the work required to strain it to the e limit is called its elastic resilience.

Under a load applied suddenly the momentary elastic distortion is to twice that caused by the same load applied gradually.

When a solid material is exposed to percussive stress, as when a w falls upon a beam transversely, the work of resistance is measured b product of the weight into the total fall.

Elevation of Ultimate Resistance and Elastic Limit was first observed by Prof. R. H. Thurston, and Commander L. A. Be lee, U. S. N., independently, in 1873, that if wrought iron be subjected stress beyond its elastic limit, but not beyond its ultimate resistance then allowed to "rest" for a definite interval of time, a considerab crease of elastic limit and ultimate resistance may be experienced. In words, the application of stress and subsequent "rest" increases the r ance of wrought iron.

This "rest" may be an entire release from stress or a simple holdin test-piece at a given intensity of stress.

Commander Beardslee prepared twelve specimens and subjected th an intensity of stress equal to the ultimate resistance of the material out breaking the specimens. These were then allowed to rest, entirel from stress, from 24 to 30 hours, after which period they were again st until broken. The gain in ultimate resistance by the rest was found t from 4.4 to 17 per cent.

This elevation of elastic and ultimate resistance appears to be pecul iron and steel: it has not been found in other metals.

Relation of the Elastic Limit to Endurance under peated Stresses (condensed from *Engineering*, August 7, 1 When engineers first began to test materials, it was soon recognize if a specimen was loaded beyond a certain point it did not recover its nal dimensions on removing the load, but took a permanent set; this was called the elastic limit. Since below this point a bar appeared to r completely its original form and dimensions on removing the load, peared obvious that it had not been injured by the load, and hence the ing load might be deduced from the elastic limit by using a small fac safety.

Experience showed, however, that in many cases a bar would not safely a stress anywhere near the elastic limit of the material as deter by these experiments, and the whole theory of any connection betwe elastic limit of a bar and its working load became almost discredite engineers employed the ultimate strength only in deducing the safe w load to which their structures might be subjected. Still, as experience mulated it was observed that a higher factor of safety was required fo load than for a dead one.

In 1871 Wöhler published the results of a number of experiments o of iron and steel subjected to live loads. In these experiments the st were put on and removed from the specimens without impact, but nevertheless, found that the breaking stress of the materials was in case much below the statical breaking load. Thus, a bar of Krupp' steel having a tenacity of 49 tons per square inch broke with a stress *tons per square inch* when the load was completely removed and re *without impa*[illegible] times. These experiments were made on rent brands of iron and steel, and the results were concor- that a bar would break with an alternating stress of only, statical breaking strength of the material, if the repetitions sufficiently numerous. At the same time, however, it ap- general trend of the experiments that a bar would stand an er of alternations of stress, provided the stress was kept

nger defines the elastic limit as the point at which stress sibly proportional to strain, the latter being measured with atus reading to $\frac{1}{5000}$th of a millimetre, or about $\frac{1}{100000}$ in. ays below the yield-point, and may on occasion be zero. On ove the yield-point, this point rises with the stress, and the or weeks, months, and possibly for years if the bar is left at ad. On the other hand, when a bar is loaded beyond its true t below its yield-point, this limit rises, but reaches a maxi- ld-point, is approached, and then falls rapidly, reaching even aving the bar at rest under a stress exceeding that of its ing-down point the elastic limit begins to rise again, and sufficient time, rise to a point much exceeding its previous

of the elastic limit of changing with the history of a bar has scredit it than anything else, nevertheless it now seems as if very property, were once more to take its former place in f engineers, and this time with fixity of tenure. It had long t the limit of elasticity might be raised, as we have said, to t within the breaking load of a bar. Thus, in some experi- sor Styffe, the elastic limit of a puddled-steel bar was raised bjecting the bar to a load exceeding its primitive elastic

limits of elasticity, one for tension and one for compression. aded a number of bars in tension until stress ceased to be tional to strain. The load was then removed and the bar ession until the elastic limit in this direction had been ex- rocess raises the elastic limit in compression, as would be g the bar in compression a second time. In place of this, now again tested in tension, when it was found that the of the limit in compression had lowered that in tension be- s value. By repeating the process of alternately testing in npression, the two limits took up points at equal distances f no load, both in tension and compression. These limits lls natural elastic limits of the bar, which for wrought iron stress of about 8½ tons per square inch, but this is practically d to which a bar of the same material can be strained alter- n and compression, without breaking when the loading is ently often, as determined by Wöhler's method
rom the rolls the elastic limit of the bar in tension is above tic limit of the bar as defined by Bauschinger, having been d by the deformations to which it has been subjected in the ufacture. Hence, when subjected to alternating stresses, sion is immediately lowered, while that in compression is y both correspond to equal loads. Hence, in Wöhler's ex- which the bars broke at loads nominally below the elastic aterial, there is every reason for concluding that the loads ater than true elastic limits of the material. This is con- on the connecting-rods of engines, which of course work ng stresses of equal intensity. Careful experiments on old the elastic limit in compression is the same as that in ten- th are far below the tension elastic limit of the material as he rolls.
opinion that straining a metal beyond its elastic limit injures untrue. It is not the mere straining of a metal beyond one t injures it, but the straining, many times repeated, beyond mits. Sir Benjamin Baker has shown that in bending a shell r the metal is of necessity strained beyond its elastic limit, of as much as 7 tons to 15 tons per square inch may obtain es from *the rolls, and unless the plate is annealed, these exist after it has been built into* the boiler. In such a case, *exposed to the additional stress* due to the pressure

the boiler, the overstrained portions of the plate will relieve themselv stretching and taking a permanent set, so that probably after a year's w ing very little difference could be detected in the stresses in a plate bui to the boiler as it came from the bending rolls, and in one which had annealed, before riveting into place, and the first, in spite of its having strained beyond its elastic limits, and not subsequently annealed, woul as strong as the other.

Resistance of Metals to Repeated Shocks.

More than twelve years were spent by Wöhler at the instance of the sian Government in experimenting upon the resistance of iron and ste repeated stresses. The results of his experiments are expressed in wh known as Wöhler's law, which is given in the following words in Du translation of Weyrauch:

"Rupture may be caused not only by a steady load which exceed carrying strength, but also by repeated applications of stresses, no which are equal to the carrying strength. The differences of these str are measures of the disturbance of continuity, in so far as by their inc the minimum stress which is still necessary for rupture diminishes."

A practical illustration of the meaning of the first portion of this law be given thus: If 50,000 pounds once applied will just break a bar of iro steel, a stress very much less than 50,000 pounds will break it if rep sufficiently often.

This is fully confirmed by the experiments of Fairbairn and Spangen as well as those of Wöhler; and, as is remarked by Weyrauch, it ma considered as a long-known result of common experience. It partiall counts for what Mr. Holley has called the "intrinsically ridiculous fact safety of six."

Another "long-known result of experience" is the fact that rupture be caused by a succession of *shocks* or *impacts*, none of which alone w be sufficient to cause it. Iron axles, the piston-rods of steam hammers, other pieces of metal subject to continuously repeated shocks, invari break after a certain length of service. They have a "life" which is ited.

Several years ago Fairbairn wrote: "We know that in some cases wro iron subjected to continuous vibration assumes a crystalline structure, that the cohesive powers are much deteriorated, but we are ignorant of causes of this change." We are still ignorant, not only of the causes of change, but of the conditions under which it takes place. Who kn whether wrought iron subjected to very slight continuous vibration wil dure forever? or whether to insure final rupture each of the continuous s shocks must amount at least to a certain percentage of single heavy s (both measured in foot-pounds), which would cause rupture with one app tion? Wöhler found in testing iron by repeated stresses (not impacts) in one case 400,000 applications of a stress of 500 centners to the square caused rupture, while a similar bar remained sound after 48,000,000 app tions of a stress of 300 centners to the square inch (1 centner = 110.2 lbs

Who knows whether or not a similar law holds true in regard to repe shocks? Suppose that a bar of iron would break under a single impac 1000 foot-pounds, how many times would it be likely to bear the repeti of 100 foot pounds, or would it be safe to allow it to remain for fifty y subjected to a continual succession of blows of even 10 foot-pounds eac

Mr. William Metcalf published in the *Metallurgical Review*, Dec. 1877, results of some tests of the life of steel of different percentages of car under impact. Some small steel pitmans were made, the specification which required that the unloaded machine should run 4½ hours at the of 1200 revolutions per minute before breaking.

The steel was all of uniform quality, except as to carbon. Here are results: The

.30 C.	ran 1 h. 21 m.	Heated and bent before breaking.
.49 C.	" 1 h. 28 m.,	" " " " "
.43 C.	" 4 h. 57 m.	Broke without heating.
.65 C.	" 3 h. 50 m.	Broke at weld where imperfect.
.80 C.	" 5 h. 40 m.	
.84 C.	" 18 h.	
.87 C.	Broke in weld near the end.	
.96 C.	Ran 4.55 m., and the machine broke down.	

s by Mr. Metcalf confirmed his conclusion.

rbon steel was better adapted to resist repeated shocks and vi-
1 low-carbon steel.
ilts, however, would scarcely be sufficient to induce any en-
.84 carbon steel in a car-axle or a bridge-rod. Further experi-
eded to confirm or overthrow them.
ption of proposed apparatus for such an investigation in the
er in Trans. A. I. M. E., vol. viii., p. 76, from which the above
ken.)

s Produced by Suddenly Applied Forces and Shocks.

Mansfield Merriman, *R. R. & Eng. Jour.*, Dec. 1889.)

e weight which is dropped from a height h upon the end of a
y be the maximum elongation which is produced. The work
the falling weight, then, is

$$W = P(h + y),$$

st equal the internal work of the resisting molecular stresses.
the bar, which is at first 0, increases up to a certain limit Q,
ter than P; and if the elastic limit be not exceeded the elonga-
s uniformly with the stress, so that the internal work is equal
stress $1/2Q$ multiplied by the total elongation y, or

$$W = 1/2\, Qy.$$

lecting the work that may be dissipated in heat,

$$1/2Qy = Ph + Py.$$

elongation due to the static load P, within the elastic limit
ence

$$Q = P\left(1 + \sqrt{1 + 2\frac{h}{e}}\right), \quad \ldots \ldots \ldots \quad (1)$$

the momentary maximum stress. Substituting this value of Q,

$$y = e\left(1 + \sqrt{1 + 2\frac{h}{e}}\right), \quad \ldots \ldots \ldots \quad (2)$$

value of the momentary maximum elongation.
sults when the force P, before its action on the bar, is moving
, as is the case when a weight P falls from a height h. The
las show that this height h may be small if e is a small quan-
very great stresses and deformations be produced. For in-
$= 4e$, then $Q = 4P$ and $y = 4e$; also let $h = 12e$, then $Q = 6P$
Or take a wrought-iron bar 1 in. square and 5 ft. long; under a
of 5000 lbs. this will be compressed about 0.0012 in., supposing
al flexure occurs; but if a weight of 5000 lbs. drops upon its end
all height of 0.0048 in. there will be produced the stress of 20,000

applied force is one which acts with the uniform intensity P
of the bar, but which has no velocity before acting upon it.
onds to the case of $h = 0$ in the above formulas, and gives $Q =$
e for the maximum stress and maximum deformation. Prob-
on of a rapidly-moving train upon a bridge produces stresses
cter.

ing the Tensile Strength of Iron Bars by Twist-
.—Ernest L. Ransome of San Francisco has obtained an English
16221 of 1888, for an "improvement in strengthening and testing
tal and steel rods or bars, consisting in twisting the same in a
. . . Any defect in the lamination of the metal which would
concealed is revealed by twisting, and imperfections are shown
e treatment may be applied to bolts, suspension-rods or bars
tensile strength of any description."
tests of this process were reported by Lieutenant F. P. Gilmore,
paper read before the Technical Society of the Pacific Coast,
the Transactions of the Society for the month of December,

also made in 1889 in the University of California. The exper-
e trials with thirty-nine bars, twenty-nine of which

riously twisted, from three-eighths of one turn to six turns per foot. Th test-pieces were cut from one and the same bar, and accurately measur and numbered. From each lot two pieces without twist were tested for ten sile strength and ductility. One group of each set was twisted until th pieces broke, as a guide for the amount of twist to be given those to b tested for tensile strain.

The following is the result of one set of Lieut. Gilmore's tests, on iro bars 8 in. long, .719 in. diameter.

No. of Bars.	Conditions.	Twists in Turns.	Twists per ft.	Tensile Strength.	Tensile per sq. in.	Gain pe cent
2	Not twisted.	0	0	22,000	54,180	
2	Twisted cold.	½	¾	23,900	59,020	9
2	" "	1	1½	25,800	63,500	17
2	" "	2	3	26,300	64,750	19
1	" "	2½	3¾	26,400	65,000	20

TENSILE STRENGTH.

The following data are usually obtained in testing by tension in a testin machine a sample of a material of construction:

The load and the amount of extension at the elastic limit.

The maximum load applied before rupture.

The elongation of the piece, measured between gauge-marks placed stated distance apart before the test; and the reduction of area at th point of fracture.

The load at the elastic limit and the maximum load are recorded in poun per square inch of the original area. The elongation is recorded as a pe centage of the stated length between the gauge-marks, and the reducti area as a percentage of the original area. The coefficient of elasticity is ca culated from the ratio the extension within the elastic limit per inch length bears to the load per square inch producing that extension.

On account of the difficulty of making accurate measurements of the fra tured area of a test-piece, and of the fact that elongation is more valuab than reduction of area as a measure of ductility and of resilience or wo of resistance before rupture, modern experimenters are abandoning th custom of reporting reduction of area. The "strength per square inch fractured section" formerly frequently used in reporting tests is now almo entirely abandoned. The data now calculated from the results of a tens test for commercial purposes are: 1. Tensile strength in pounds per squa inch of original area. 2. Elongation per cent of a stated length betwe gauge-marks, usually 8 inches. 3. Elastic limit in pounds per square inc of original area.

The short or grooved test specimen gives with most metals, especiall with wrought iron and steel, an apparent tensile strength much high than the real strength. This form of test-piece is now almost entirely aba doned.

The following results of the tests of six specimens from the same 1¼" st bar illustrate the apparent elevation of elastic limit and the changes other properties due to change in length of stems which were turned do in each specimen to .798" diameter. (Jas. E. Howard, Eng. Congress 18 Section G.)

Description of Stem.	Elastic Limit, Lbs. per Sq. In.	Tensile Strength, Lbs. per Sq. In.	Contraction of Area, per cent
1.00" long.............	64,900	94,400	49.0
.50 "	65,320	97,800	43.4
.25 "	68,000	102,420	39.6
Semicircular groove, .4" radius..........	75,000	116,380	31.6
Semicircular groove, ⅛" radius...........	86,000, about	134,960	23.0
V-shaped groove.......	90,000, about	117,000	Indeterminate

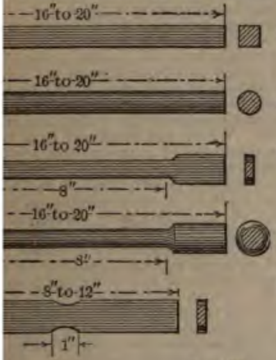
nade by the author in 1879 of straight and grooved test-pieces steel cut from the same gave the following results:

aight pieces, 56,605 to 59,012 lbs. T. S. Aver. 57,566 lbs.
oved " 64,341 to 67,400 " " " 65,452 "
f the short or grooved specimen, 21 per cent, or 12,114 lbs.

nent of Elongation.—In order to be able to compare ngation, it is necessary not only to have a uniform length of en gauge-marks (say 8 inches), but to adopt a uniform method the elongation to compensate for the difference between the gation when the piece breaks near one of the gauge-marks, reaks midway between them. The following method is rec- rans. A. S. M. E., vol. xi., p. 622):

specimen divisions of 1/2 inch each. After fracture measure nt of fracture the length of 8 of the marked spaces on each tion (or 7 + on one side and 8 + on the other if the fracture is the marks). The sum of these measurements, less 8 inches, is n of 8 inches of the original length. If the fracture is so of the specimen that 7 + spaces are not left on the shorter take the measurement of as many spaces (with the fractional the fracture) as are left, and for the spaces lacking add the of as many corresponding spaces of the longer portion as are make the 7 + spaces.

f Specimens for Tensile Tests.—The shapes shown e recommended by the author in 1882 when he was connected



No. 1. Square or flat bar, as rolled.

No. 2. Round bar, as rolled.

No. 3. Standard shape for flats or squares. Fillets ½ inch radius.

No. 4. Standard shape for rounds. Fillets ½ in. radius.

No. 5. Government shape for marine boiler-plates of iron. Not recommended for other tests, as results are generally in error.

Fig. 75.

sburgh Testing Laboratory. They are now in most general ier forms, with 5 inches or less in length between shoulders, entirely abandoned.

ions Required in making Tensile Tests.—The ine itself should be tested, to determine whether its weighing accurate, and whether it is so made and adjusted that in the perly made specimen the line of strain of the testing-machine in line with the axis of the specimen.

nen should be so shaped that it will not give an incorrect record

e of uniform minimum section for not less than five inches of

st be had to the time occupied in making tests of certain mate- ght iron and soft steel can be made to show a higher than their ent strength by keeping *them under strain* for a great length

oft alloys, copper, tin, zinc, and the like, which flow under con- *heir highest apparent strength* is obtained by testing them *ording tests of such materials* the length of time occupied in *be stated.*

For very accurate measurements of elongation, corresponding t ments of load during the tests, the electric contact micrometer, de in Trans. A. S. M. E., vol. vi., p. 479, will be found convenient. Whe ings of elongation are then taken during the test, a strain diagram plotted from the reading, which is useful in comparing the qualities ferent specimens. Such strain diagrams are made automatically by t Olsen testing-machine, described in *Jour. Frank. Inst.* 1891.

The coefficient of elasticity should be deduced from measurem served between fixed increments of load per unit section, say betwe and 12,000 pounds per square inch or between 1000 and 11,000 pounds of between 0 and 10,000 pounds.

COMPRESSIVE STRENGTH.

What is meant by the term "compressive strength" has not ye settled by the authorities, and there exists more confusion in regard term than in regard to any other used by writers on strength of ma The reason of this may be easily explained. The effect of a comp stress upon a material varies with the nature of the material, and w shape and size of the specimen tested. While the effect of a tensile s to produce rupture or separation of particles in the direction of the strain, the effect of a compressive stress on a piece of material may be to cause it to fly into splinters, to separate into two or more wedge- pieces and fly apart, to bulge, buckle, or bend, or to flatten out and ult sist rupture or separation of particles. A piece of speculum meta compressive stress will exhibit no change of appearance until ruptur place, and then it will fly to pieces as suddenly as if blown apart powder. A piece of cast iron or of stone will generally split into shaped fragments. A piece of wrought iron will buckle or bend. A p wood or zinc may bulge, but its action will depend upon its shape a A piece of lead will flatten out and resist compression till the last that is, the more it is compressed the greater becomes its resistance.

Air and other gaseous bodies are compressible to any extent as they retain the gaseous condition. Water not confined in a vessel pressed by its own weight to the thickness of a mere film, while wh fined in a vessel it is almost incompressible.

It is probable, although it has not been determined experimentall solid bodies when confined are at least as incompressible as water. they are not confined, the effect of a compressive stress is not shorten them, but also to increase their lateral dimensions or bulge Lateral strains are therefore induced by compressive stresses.

The weight per square inch of original section required to produ given amount or percentage of shortening of any material is not a co quantity, but varies with both the length and the sectional area, w shape of this sectional area, and with the relation of the area to the The "compressive strength" of a material, if this term be supposed t the weight in pounds per square inch necessary to cause rupture, ma with every size and shape of specimen experimented upon. Still mor cult would it be to state what is the "compressive strength" of a m which does not rupture at all, but flattens out. Suppose we are te cylinder of a soft metal like lead, two inches in length and one inch in eter, a certain weight will shorten it one per cent, another weight t cent, another fifty per cent, but no weight that we can place upon rupture it, for it will flatten out to a thin sheet. What, then, is its co sive strength? Again, a similar cylinder of soft wrought iron woul ably compress a few per cent, bulging evenly all around; it would the mence to bend, but at first the bend would be imperceptible to the e too small to be measured. Soon this bend would be great enough noticed, and finally the piece might be bent nearly double, or otherw torted. What is the "compressive strength" of this piece of iron the weight per square inch which compresses the piece one per cent per cent, that which causes the first bending (impossible to be disco or that which causes a perceptible bend?

As showing the confusion concerning the definitions of comp strength, the following statements from different authorities on the st of wrought iron are of interest.

Wood's Resistance of Materials states, "comparatively few exper have been made to determine how much wrought iron will sustain ut of … dgkinson gives 65,000, Rondulet 70,800, Weisbac ame diameter, does not vary much, provided the length of the ess than one and does not exceed four or five diameters, and ght which will just crush a short prism whose base equals one and whose height is not less than 1 to 1½ and does not exceed ers, is called the crushing strength of the material. It would perimenters would all agree upon some such definition of the ing strength," and insist that all experiments which are made ose of testing the relative values of different materials in com- made on specimens of exactly the same shape and size. An e and shape should be assumed and agreed upon for this pur- ze mentioned by Stoney is definite as regards area of section, are inch, but is indefinite as regards length, viz., from one to s. In some metals a specimen five diameters long would bend, ich lower apparent strength than a specimen having a length of r. The words "will just crush" are also indefinite for ductile which the resistance increases without limit if the piece tested d. In such cases the weight which causes a certain percentage on, as five, ten, or fifty per cent, should be assumed as the ngth.

experiments on crushing strength three things are desirable: itrary standard shape and size of test specimen for comparison ials. Secondly, a standard limit of compression for ductile ich shall be considered equivalent to fracture in brittle mate- y, an accurate knowledge of the relation of the crushing specimen of standard shape and size to the crushing strength of all other shapes and sizes. The latter can only be very extensive and accurate series of experiments upon all erials, and on specimens of a great number of different shapes

proposes, as a standard shape and size, for a compressive test all metals, a cylinder one inch in length, and one half square nal area, or 0.798 inch diameter; and for the limit of compres- nt to fracture, ten per cent of the original length. The term e strength," or "compressive strength of standard specimen," nean the weight per square inch required to fracture by com- ss a cylinder one inch long and 0.798 inch diameter, or to gth to 0.9 inch if fracture does not take place before that reduc-

The Committee on Standard Tests of the American Society of Mechan Engineers say (vol. xi., p. 624):

"Although compression tests have heretofore been made on dimin sample pieces, it is highly desirable that tests be also made on long p from 10 to 20 diameters in length, corresponding more nearly with ac practice, in order that elastic strain and change of shape may be determ by using proper measuring apparatus.

The elastic limit, modulus or coefficient of elasticity, maximum and mate resistances, should be determined, as well as the increase of sectio various points, viz., at bearing surfaces and at crippling point.

The use of long compression-test pieces is recommended, because th vestigation of short cubes or cylinders has led to no direct applicatio the constants obtained by their use in computation of actual structu which have always been and are now designed according to empirical mulæ obtained from a few tests of long columns."

COLUMNS, PILLARS, OR STRUTS.

Hodgkinson's Formula for Columns.

P = crushing weight in pounds; d = exterior diameter in inches; d_1 = terior diameter in inches; L = length in feet.

Kind of Column.	Both ends rounded, the length of the column exceeding 15 times its diameter.	Both ends flat, length of the col exceeding 30 t its diameter.
Solid cylindrical columns of cast iron.....	$P = 33,380\frac{d^{3.76}}{L^{1.7}}$	$P = 98,920\frac{d^{3.55}}{L^{1.7}}$
Hollow cylindrical columns of cast iron.....	$P = 29,120\frac{d^{3.76} - d_1^{3.76}}{L^{1.7}}$	$P = 99,320\frac{d^{3.55} - d}{L^{1.7}}$
Solid cylindrical columns of wrought iron.	$P = 95,850\frac{d^{3.76}}{L^2}$	$P = 299,600\frac{d^{3.55}}{L^2}$
Solid square pillar of Dantzic oak (dry)....	....	$P = 24,540\frac{d^4}{L^2}$
Solid square pillar of red deal (dry)........	....	$P = 17,510\frac{d^4}{L^2}$

The above formulæ apply only in cases in which the length is so great the column breaks by bending and not by simple crushing. If the col be shorter than that given in the table, and more than four or five time diameter, the strength is found by the following formula:

$$W = \frac{PCK}{P + \frac{3}{4}CK},$$

in which P = the value given by the preceding formulæ, K = the transv section of the column in square inches, C = the ultimate compressive r tance of the material, and W = the crushing strength of the column.

Hodgkinson's experiments were made upon comparatively short colu the greatest length of cast-iron columns being $60\frac{1}{2}$ inches, of wrought $90\frac{3}{4}$ inches.

The following are some of his conclusions:

1. In all long pillars of the same dimensions, when the force is appli the direction of the axis, the strength of one which has flat ends is a three times as great as one with rounded ends.

2. The strength of a pillar with one end rounded and the other flat is arithmetical mean between the two given in the preceding case of the s dimensions.

3. The strength of a pillar having both ends firmly fixed is the sam one of half the length with both ends rounded.

4. The strength of a pillar is not increased more than one seventh by

mulæ deduced from Hodgkinson's experiments are more n Hodgkinson's own. They are:

h both ends fixed or flat, $P = \frac{fS}{1 + a\frac{l^2}{r^2}}$;

h one end flat, the other end round, $P = \frac{fS}{1 + 1.8a\frac{l^2}{r^2}}$;

h both ends round, or hinged, $P = \frac{fS}{1 + 4a\frac{l^2}{r^2}}$;

ross-section in inches;
resistance of column, in pounds;
strength of the material in lbs. per square inch;
ius of gyration, in inches, $r^2 = \frac{\text{Moment of inertia}}{\text{area of section}}$;
column in inches;
ent depending upon the material;
ly taken as constants: they are really empirical variables, e dimensions and character of the column as well as upon r.)

t-iron columns, values commonly taken are: $f = 36{,}000$ to $\frac{1}{40{,}000}$.

n columns, $f = 80{,}000$, $a = \frac{1}{6400}$.

ron columns, fixed ends, $p = \frac{80{,}000}{1 + \frac{1}{800}\frac{l^2}{d^2}}$, $l =$ length and e same unit, and $p =$ strength in lbs. per square inch.

ker gives,

mild steel, $f = 67{,}000$ lbs., $a = \frac{1}{22{,}400}$.

strong steel, $f = 114{,}000$ lbs., $a = \frac{1}{14{,}400}$

rs these only loose approximations for the ultimate resis-

ankine gives $f = 7200$ lbs., $a = 1/3000$.

INERTIA AND RADIUS OF GYRATION.

t of Inertia of a section is the sum of the products of rea of the section into the square of its distance from an tation, as the neutral axis.

f gyration of the section equals the square root of the oment of inertia divided by the area of the section. If tion, $I =$ moment of inertia and $A =$ area,

$$R = \sqrt{\frac{I}{A}}. \qquad \frac{I}{A} = R^2.$$

inertia of various sections are as follows:
outside diameter; $d_1 =$ inside diameter; $b =$ breadth; inside breadth and diameter;

$= 1/12bh^3$;	Hollow rectangle $I = 1/12(bh^3 - b_1h_1^3)$;
$= 1/12b^4$;	Hollow square $I = 1/12(b^4 - b_1^4)$;
$= 1/64\pi d^4$;	Hollow cylinder $I = 1/64\pi(d^4 - d_1^4)$.

nertia and Radius of Gyration for Various their Use in the Formulas for Strength of olumns.—The strength of sections to resist strains, as columns, depends not only on the area but also on the n, *and the property of the section* which forms the basis *sed in the formulas for strength of girders and* [illegible] *of the form, is its moment of inertia about* *ent of resistance of any section to transve* is its moment of inertia divided by the distance from the neutra the fibres farthest removed from that axis; or

$$\text{Moment of resistance} = \frac{\text{Moment of inertia}}{\text{Distance of extreme fibre from axis}}.$$

Moment of Inertia of Compound Shapes. (Penco Works.)—The moment of inertia of any section about any axis is eq I about a parallel axis passing through its centre of gravity + (th the section × the square of the distance between the axes).

By this rule, the moments of inertia or radii of gyration of any s tions being known, corresponding values may be obtained for any c tion of these sections.

Radius of Gyration of Compound Shapes.—In the pair of any shape without a web the value of R can always be fou out considering the moment of inertia.

The radius of gyration for any section around an axis parallel to axis passing through its centre of gravity is found as follows:

Let r = radius of gyration around axis through centre of gravi radius of gyration around another axis parallel to above; d = dist tween axes:

$$R = \sqrt{d^2 + r^2}.$$

When r is small, R may be taken as equal to d without material e

Graphical Method for Finding Radius of Gyration F. La Rue, *Eng. News*, Feb. 2, 1893, gives a short graphical me finding the radius of gyration of hollow, cylindrical, and rectang umns, as follows:

For cylindrical columns:

Lay off to a scale of 4 (or 40) a right-angled triangle, in which equals the outer diameter, and the altitude equals the inner diamet column, or *vice versa*. The hypothenuse, measured to a scale of 10), will be the radius of gyration sought.

This depends upon the formula

$$G = \sqrt{\frac{\text{Mom. of Inertia}}{\text{Area}}} = \frac{\sqrt{D^2 + d^2}}{4},$$

in which A = area and D = diameter of outer circle, a = area and meter of inner circle, and G = radius of gyration. $\sqrt{D^2 + d^2}$ is the sion for the hypothenuse of a right-angled triangle, in which D and base and altitude.

The sectional area of a hollow round column is $.7854(D^2 - d^2)$. structing a right-angled triangle in which D equals the hypothenus equals the altitude, the base will equal $\sqrt{D^2 - d^2}$. Calling the valu expression for the base B, the area will equal $.7854B^2$.

Value of G for square columns:

Lay off as before, but using a scale of 10, a right-angled triangle the base equals D or the side of the outer square, and the altitude the side of the inner square. With a scale of 3 measure the hypo which will be, approximately, the radius of gyration.

This process for square columns gives an excess of slightly more By deducting 4% from the result, a close approximation will be obta

A very close result is also obtained by measuring the hypothen the same scale by which the base and altitude were laid off, and mul by the decimal 0.29; more exactly, the decimal is 0.28867.

The formula is

$$G = \sqrt{\frac{\text{Mom. of inertia}}{\text{Area}}} = \frac{1}{\sqrt{12}}\sqrt{D^2 + d^2}, = 0.28867\sqrt{D^2 +$$

This may also be applied to any rectangular column by using th diameters of an unsupported column, and the greater diameters if umn is supported in the direction of its least dimensions.

ELEMENTS OF USUAL SECTIONS.

Moments refer to horizontal axis through centre of gravity. This intended for convenient application where extreme accuracy is no *tant. Some of the terms are only approximate; those marked * are* *Values for radius of gyration in flanged beams apply to standard m* *sections only.* A = area of section; b = breadth; h = depth; D =

ion.	Moment of Inertia.	Moment of Resistance.	Square of Least Radius of Gyration.	Least Radius of Gyration.
Rect-e.	$\frac{bh^3}{12}$ *	$\frac{bh^2}{6}$ *	$\frac{(\text{Least side})^2}{12}$ *	$\frac{\text{Least side}}{3.46}$ *
Rect-e.	$\frac{bh^3-b_1h_1^3}{12}$ *	$\frac{bh^3-b_1h_1^3}{6h}$ *	$\frac{h^2+h_1^2}{12}$ *	$\frac{h+h^1}{4.89}$
Circle.	$\frac{AD^2}{16}$ *	$\frac{AD}{8}$ *	$\frac{D^2}{16}$ *	$\frac{D}{4}$ *
Circle. rea of section; ea of l section.	$\frac{AD^2-ad^2}{16}$	$\frac{AD^2-ad^2}{8D}$	$\frac{D^2+d^2}{16}$ *	$\frac{D+d}{5.64}$
Triangle.	$\frac{bh^3}{36}$	$\frac{bh^2}{24}$	The least of of the two: $\frac{h^2}{18}$ or $\frac{b^2}{24}$	The least of the two: $\frac{h}{4.24}$ or $\frac{b}{4.9}$
Angle.	$\frac{Ah^2}{10.2}$	$\frac{Ah}{7.2}$	$\frac{b^2}{25}$	$\frac{b}{5}$
n Angle.	$\frac{Ah^2}{9.5}$	$\frac{Ah}{6.5}$	$\frac{(hb)^2}{13(h^2+b^2)}$	$\frac{hb}{2.6(h+b)}$
Cross.	$\frac{Ah^2}{19}$	$\frac{Ah}{9.5}$	$\frac{h^2}{22.5}$	$\frac{h}{4.74}$
Tee.	$\frac{Ah^2}{11.1}$	$\frac{Ah}{8}$	$\frac{b^2}{22.5}$	$\frac{b}{4.74}$
n.	$\frac{Ah^2}{6.66}$	$\frac{Ah}{3.2}$	$\frac{b^2}{21}$	$\frac{b}{4.58}$
el.	$\frac{Ah^2}{7.34}$	$\frac{Ah}{3.67}$	$\frac{b^2}{12.5}$	$\frac{b}{3.54}$
Beam.	$\frac{Ah^2}{6.9}$	$\frac{Ah}{4}$	$\frac{b^2}{36.5}$	$\frac{b}{6}$

from centre of gravity, solid triangle, $\frac{h}{3}$; even angle, $\frac{h}{3.3}$;
even tee, $\frac{h}{3.3}$; *deck beam*, $\frac{h}{2.3}$; all other shapes given in

Solid Cast-iron Columns.

Hurst gives the following table, based on Hodgkinson's formula (
2240 lbs.)

The figures are the safe load or $\frac{1}{6}$ of the breaking weight in tons, f
columns, ends flat and fixed.

Diam. in Inches.	Length of Column in Feet.							
	6.	8.	10.	12.	14.	16.	18.	20.
1½	.92	.59	.34	.25	.19	.15	.13	.11
1¾	1.43	.97	.60	.44	.34	.27	.22	.19
2	2.31	1.41	.97	.71	.55	.44	.35	.30
2¼	3.52	2.16	1.48	1.08	.83	.67	.54	.46
2½	5.15	3.16	2.15	1.58	1.22	.97	.80	.66
2¾	7.25	4.45	3.05	2.23	1.72	1.37	1.12	.94
3	9.93	6.09	4.17	3.06	2.35	1.87	1.53	1.29
3½	17.29	10.60	7.25	5.32	4.10	3.26	2.67	2.25
4	27.96	17.15	11.73	8.61	6.62	5.28	4.32	3.61
4½	42.73	26.20	17.93	13.15	10.12	8.07	6.60	5.30
5	62.44	38.29	26.20	19.22	14.79	11.79	9.65	8.06
5½	88.00	53.97	36.93	27.09	20.84	16.61	13.60	11.37
6	120.4	73.82	50.51	37.05	28.51	22.72	18.60	15.55
6½	160.6	98.47	67.38	49.43	38.03	30.31	24.01	20.74
7	209.7	128.6	87.98	64.53	49.66	39.57	32.30	27.08
7½	268.8	164.8	112.8	82.73	63.66	50.73	41.53	34.72
8	339.1	207.9	142.3	104.4	80.31	64.00	52.39	43.80
8½	421.8	258.6	177.0	129.8	99.90	79.61	65.16	54.48
9	518.2	317.7	217.4	159.5	122.7	97.80	80.05	66.92
9½	629.5	386.0	264.2	193.8	149.1	118.8	97.25	81.70
10	757.2	464.3	317.7	233.1	179.3	142.9	117.0	97.79
10½	902.6	553.5	378.7	277.8	213.8	170.3	139.4	116.6
11	1067.1	654.4	447.8	328.5	252.7	201.4	164.9	137.8
11½	1252.3	767.9	525.5	385.4	296.6	236.4	193.5	161.7
12	1459.6	895.1	612.5	449.3	345.7	275.5	225.5	188.5

The correction for short columns should be applied where the le
less than 30 diameters.

$$\text{Strength in tons of short columns} = \frac{SC}{10S + \frac{3}{4}C},$$

S being the strength for long columns given in the above table, and
times the sectional area of the metal in inches.

Hollow Columns.—The strength nearly equals the differe
tween that of two solid columns the diameters of which are equal
external and internal diameters of the hollow one.

Ultimate Strength of Hollow, Cylindrical Wrough Cast-iron Columns, when fixed at the ends.

(Pottsville Iron and Steel Co.)

$$\text{Computed by Gordon's formula, } p = \frac{f}{1 + C\left(\frac{l}{d}\right)^2}.$$

p = Ultimate strength in lbs. per square inch;
l = Length of column, } both in same units;
d = Diameter of column, }
f = { 40,000 lbs. for wrought-iron; } { 80,000 lbs. for cast-iron; }
C = 1/3000 for wrought-iron, and 1/800 for cast-iron.

For cast-iron, $p = \dfrac{80000}{1 + \frac{1}{800}\left(\frac{l}{d}\right)^2}$.

For wrought-iron, $p = \dfrac{40000}{1 + \frac{1}{3000}\left(\frac{l}{d}\right)^2}$.

HOLLOW CYLINDRICAL COLUMNS.

Maximum Load per sq. in.		Safe Load per square inch.	
Cast Iron.	Wrought Iron.	Cast Iron, Factor of 6.	Wrought Iron, Factor of 4.
74075	39164	12346	9791
71110	38710	11851	9677
67796	38168	11299	9542
64256	37546	10709	9386
60606	36854	10101	9213
56938	36100	9489	9025
53332	35294	8889	8823
49845	34442	8307	8610
46510	33556	7751	8389
43360	32642	7226	8161
40404	31712	6734	7928
37646	30768	6274	7692
35088	29820	5848	7455
32718	28874	5453	7218
30584	27932	5097	6983
28520	27002	4753	6750
26666	26086	4444	6522
24962	25188	4160	6297
23396	24310	3899	6077
21946	23454	3658	5863
20618	22620	3436	5655
19392	21818	3262	5454
18282	21036	3047	5259
17222	20284	2870	5071
16260	19556	2710	4889
15368	18856	2561	4714
14544	18180	2424	4545

imate Strength of Wrought-iron Columns.

p = ultimate strength per square inch;
l = length of column in inches;
r = least radius of gyration in inches.

are end-bearings, $p = \dfrac{40000}{1 + \frac{1}{40000}\left(\frac{l}{r}\right)^2}$.

pin and one square bearing, $p = \dfrac{40000}{1 + \frac{1}{30000}\left(\frac{l}{r}\right)^2}$.

o pin-bearings, $p = \dfrac{40000}{1 + \frac{1}{20000}\left(\frac{l}{r}\right)^2}$.

e working load on *these columns* use a factor of 4 when used in *, or when subjected to dead load only*; but when used in b*
should be 5.

WROUGHT-IRON COLUMNS.

$\frac{l}{r}$	Ultimate Strength in lbs. per square inch.			$\frac{l}{r}$	Safe Strength in lbs. square inch—Factor	
	Square Ends.	Pin and Square End.	Pin Ends.		Square Ends.	Pin and Square End.
10	39944	39866	39800	10	7989	7973
15	39776	39702	39554	15	7955	7940
20	39604	39472	39214	20	7921	7894
25	39384	39182	38788	25	7877	7836
30	39118	38834	38278	30	7821	7767
35	38810	38430	37690	35	7762	7686
40	38460	37974	37036	40	7692	7595
45	38072	37470	36322	45	7614	7494
50	37646	36928	35525	50	7529	7386
55	37186	36336	34744	55	7437	7267
60	36697	35714	33898	60	7339	7143
65	36182	34478	33024	65	7236	6896
70	35634	34384	32128	70	7127	6877
75	35076	33682	31218	75	7015	6736
80	34482	32966	30288	80	6896	6593
85	33883	32236	29384	85	6777	6447
90	33264	31496	28470	90	6653	6299
95	32636	30750	27562	95	6527	6150
100	32000	30000	26666	100	6400	6000
105	31357	29250	25786	105	6271	5850

Maximum Permissible Stresses in columns used in bui
(Building Ordinances of City of Chicago, 1893.)

Maximum permissible loads:

For cast-iron round columns:

$$S = \frac{10000a}{1 + \frac{l^2}{600d^2}}.$$

l = length of column in inches;
d = diameter of column in inches;
a = area of column in square inches.

For cast-iron rectangular columns;

$$S = \frac{10000a}{1 + \frac{l^2}{800d^2}}.$$

l and a as before;
d = least horizontal dimension of colum

For riveted or other forms of wrought-iron columns:

$$S = \frac{12000a}{1 + \frac{l^2}{36000r^2}}.$$

l = and a as before;
r = least radius of gyration in inches.

For riveted or other steel columns, if more than $60r$ in length:

$$S = 17,000 - \frac{60l}{r}.$$ l and r as before.

If less than $60r$ in length::

$S = 13,500a$. a as before.

For wooden posts:

$$S = \frac{ac}{1 + \frac{l^2}{250d^2}}.$$

a = area of post in square inches;
d = least side of rectangular post in in
l = length of post in inches;

$$c = \begin{cases} 600 \text{ for white or Norway pine;} \\ 800 \text{ for oak;} \\ 900 \text{ for long-leaf yellow pine.} \end{cases}$$

LOAD OF HOLLOW CYLINDRICAL CAST-IRON COLUMNS. (New Jersey Steel Iron Co.)

(One fifth the breaking weight.)

llowing tables give the safe load in tons of 2,000 lbs., for columns apitals and bases *accurately turned* to a true plane, and having a fair bearing on these surfaces. In the case of columns having nds, but set only with the degree of care usual in ordinary building, *half* of these loads should be taken; and for columns not turned at ving rounded ends, *one third* of these amounts should be taken for load. Columns having one end accurately turned to a true plane, other rounded, may be loaded to two thirds the amount given in the

oad, in Tons of 2000 lbs. for Cast-iron Columns with Turned Capitals and Bases.

side meter, ches.		Length in ft.	Outside Diameter, 3 inches.			Length in ft.	Outside Diameter, 4 inches.				Length in ft.	Outside Diameter, 4 inches.			
ness in ches.			Thickness in inches.				Thickness in inches.					Thickness in inches.			
¾	1		½	¾	1		½	¾	1	1¼		½	¾	1	1¼
.9	17.2	17	3.0	3.6	3.9	7	24.9	32.9	38.3	41.7	17	7.0	8.9	10.1	10.7
.0	14.0	18	2.8	3.3	3.5	8	21.7	28.4	33.0	35.8	18	6.4	8.1	9.1	9.7
.7	11.4	19	2.5	3.0	3.2	9	19.0	24.8	28.7	31.0	19	5.8	7.4	8.3	8.8
.9	9.6	20	2.3	2.7	2.9	10	17.4	22.0	24.9	26.3	20	5.3	6.8	7.6	8.1
.8	8.1	21	2.1	2.5	2.7	11	14.8	18.7	21.1	22.4	21	4.9	6.2	7.0	7.5
.6	7.0	22	1.9	2.3	2.5	12	12.7	16.2	18.2	19.3	22	4.6	5.8	6.5	6.9
.7	6.1	23	1.8	2.1	2.3	13	11.1	14.1	15.9	16.8	23	4.2	5.3	6.0	6.4
.0	5.4	24	1.7	2.0	2.1	14	9.8	12.4	14.0	14.9	24	3.9	5.0	5.6	5.9
.5	4.8	25	1.6	1.9	2.0	15	8.7	11.1	12.5	13.2	25	3.7	4.6	5.2	5.5
.0	4.3					16	7.8	9.9	11.2	11.8					

de Diameter, 5 inches.			Outside Diameter, 6 inches.				Outside Diameter, 7 inches.			
ness in inches.			Thickness in inches.				Thickness in inches.			
¾	1	1¼	¾	1	1¼	1½	¾	1	1¼	1½
.8	65.0	73.3	77.3	95.5	110.3	122.1	102.4	128.7	150.7	169.4
.6	57.3	64.4	69.7	85.7	98.7	108.8	93.6	117.0	136.9	153.5
.3	50.7	56.8	62.8	77.1	88.5	97.3	85.6	106.7	124.6	139.3
.7	45.1	50.4	56.9	69.6	79.6	87.4	78.4	97.5	113.5	126.6
.8	40.3	44.9	51.6	63.0	71.9	78.7	71.8	89.2	103.6	115.3
.5	36.2	40.3	46.9	57.2	65.2	71.2	66.0	81.7	94.8	105.3
.6	32.2	35.2	42.9	52.1	59.3	64.6	60.7	75.1	87.0	96.5
.3	28.3	31.0	39.3	47.6	54.1	58.9	56.0	69.2	80.0	88.6
.6	25.2	27.6	36.8	43.9	49.0	52.6	51.8	63.9	73.8	81.6
.4	22.6	24.7	33.0	39.4	44.0	47.2	48.1	59.2	68.2	75.4
.5	20.4	22.3	29.8	35.5	39.7	42.5	44.6	54.9	63.2	69.8
.9	18.5	20.2	27.0	32.2	36.0	38.6	42.0	50.9	57.8	63.0
.5	16.9	18.4	24.6	29.4	32.8	35.2	38.3	46.4	52.7	57.4
.3	15.4	16.9	22.6	26.9	30.1	32.3	35.1	42.5	48.3	52.6
.2	14.2	15.5	20.8	24.8	27.7	29.7	32.3	39.1	44.5	48.4
.3	13.1	14.4	19.2	22.9	25.6	27.4	29.8	36.2	41.1	44.7
.5	12.2	13.3	17.8	21.2	23.7	25.4	27.7	33.5	38.1	41.5
.7	11.3	12.4	16.6	19.7	22.1	23.7	25.7	31.2	35.4	38.6
.1	10.6	11.5	15.4	18.4	20.6	22.1	24.0	29.1	33.1	36.0

Safe Load, in Tons of 2000 lbs. for Cast-Iron C[olumns]
with Turned Capitals and Bases.

Length in ft.	Outside Diameter, 8 inches. Thickness in inches.				Outside Diameter. 9 inches. Thickness in inches.				Outside Dia[meter,] 10 inche[s.] Thickness in [inches.]	
	¾	1	1¼	1½	¾	1	1¼	1½	¾	1
7	128.3	162.6	193.0	219.5	154.8	197.7	236.6	271.4	181.6	233.4
8	118.7	150.1	177.7	201.6	144.7	184.5	220.2	252.0	171.1	219.5
9	109.8	138.5	163.6	185.2	135.0	171.8	204.7	233.9	160.9	206.2
10	101.5	127.8	150.7	170.2	126.0	160.0	190.3	217.0	151.2	193.4
11	94.0	118.0	139.0	156.7	117.5	149.0	177.0	201.4	142.0	181.4
12	87.0	109.2	128.2	144.3	109.6	138.8	164.5	187.0	133.4	170.1
13	80.7	101.1	118.5	133.2	102.4	129.4	153.2	173.9	125.3	159.6
14	75.0	93.8	109.8	123.2	95.7	120.8	142.8	161.9	117.8	149.8
15	69.8	87.1	101.9	114.2	89.5	112.9	133.3	150.9	110.8	140.7
16	65.0	81.1	94.7	106.1	83.9	105.7	124.6	140.9	104.3	132.4
17	60.7	75.7	88.3	98.7	78.7	99.0	116.7	131.8	98.3	124.6
18	56.8	70.7	82.4	92.1	73.9	92.9	109.4	123.5	92.7	117.4
19	53.2	66.2	77.1	86.1	69.6	87.4	102.7	115.9	87.5	110.8
20	51.1	62.7	72.1	79.5	65.5	82.3	96.7	108.9	82.7	104.6
21	47.0	57.7	66.4	73.2	61.8	75.5	91.0	102.6	78.3	99.0
22	43.5	53.3	61.3	67.6	58.4	73.2	85.9	96.7	74.2	93.7
23	40.3	49.4	56.8	62.7	55.9	69.3	80.4	89.5	70.4	88.9
24	37.5	46.0	52.9	58.3	52.0	64.4	74.8	83.3	66.9	84.3
25	35.0	42.9	49.3	54.4	48.5	60.1	69.8	77.7	64.9	81.0

Length in ft.	Outside Diameter, 11 inches. Thickness in inches.				Outside Diameter, 12 inches. Thickness in inches.				Outside Dia[meter,] 13 inche[s.] Thickness in [inches.]	
	1	1¼	1½	2	1	1¼	1½	2	1	1¼
7	269.4	325.9	377.6	469.5	305.3	370.8	431.7	540.9	341.5	414.4
8	255.1	308.1	356.8	442.2	290.9	352.8	410.2	512.8	327.0	396.3
9	241.2	290.8	336.3	415.6	276.6	335.0	389.1	485.0	312.4	378.4
10	227.8	274.2	316.7	390.3	262.7	317.7	368.6	458.3	298.0	360.6
11	214.9	258.4	298.1	366.3	249.2	301.0	348.8	432.9	284.0	343.4
12	202.7	243.5	280.5	343.9	236.3	285.1	330.0	408.6	270.5	326.7
13	191.2	229.4	264.0	322.8	223.9	270.0	312.2	385.7	257.5	310.8
14	180.5	216.2	248.5	303.3	212.3	255.6	295.3	364.1	245.0	295.5
15	170.3	203.9	234.1	285.1	201.2	242.1	279.4	343.9	233.2	281.1
16	160.9	192.4	220.7	268.3	190.8	229.4	264.5	325.0	222.0	267.3
17	152.1	181.7	208.2	252.7	181.1	217.5	250.6	307.4	211.3	254.4
18	143.9	171.7	196.7	238.3	171.9	206.3	237.5	290.9	201.3	242.1
19	136.2	162.5	185.9	225.0	163.3	195.8	225.3	275.6	191.8	230.6
20	129.1	153.9	176.0	212.6	155.2	186.0	213.9	261.3	182.8	219.7
21	122.4	145.9	166.7	201.2	147.7	176.9	203.2	247.9	174.4	209.5
22	116.3	138.4	158.1	190.6	140.6	168.3	193.3	235.5	166.5	199.9
23	110.5	131.5	150.1	180.7	134.0	160.3	184.0	224.0	159.0	190.8
24	105.2	125.1	142.7	171.6	127.8	152.8	175.3	213.2	152.0	182.3
25	100.2	119.1	135.7	163.1	122.0	145.8	167.1	203.1	145.4	174.3

afe Load of Cast-iron Columns—(*Continued*).

Diameter, nches.			Outside Diameter, 15 inches.				Outside Diameter, 16 inches.			
s in inches.			Thickness in inches.				Thickness in inches.			
	1½	2	1	1¼	1½	2	1	1¼	1½	2
.1	539.9	684.6	413.7	506.1	594.0	756.7	449.8	551.1	648.0	828.6
8	518.0	655.9	399 3	487.9	572.2	727.7	435.3	532.8	626.3	799.8
.4	496 3	627.0	384.4	469.5	550.1	698.4	420.5	514.4	604.1	770.4
3	474 6	598.5	369.7	451.0	528.2	669.3	405.6	496.0	581.8	740.9
.5	453.4	570 7	355.1	433.0	506.3	640.9	390.6	477.4	559.8	711.7
.1	432 6	543 6	340.6	415.0	485.0	612.8	376.0	459.3	538.0	683.4
3	412.7	517.7	326.6	397.6	464.5	585.9	361.6	441.2	516.7	655.1
8.2	393.6	493.0	313.0	380.7	444.4	559.7	347.6	423.8	495.9	628.0
2.7	375.3	469.4	299.9	364.5	425.2	534.9	333.9	406.9	475.9	601.8
8.0	357.9	446.9	287.2	348.9	406.7	510.9	320 7	390.6	456.6	576.6
4 0	341.4	425.7	275.1	334.0	389.1	488.1	308.0	374.9	438.0	552 5
0.6	325.6	405 5	263.6	319.7	372.2	466.5	295.8	359.9	420.1	529.4
28.0	310.8	386.5	252.5	306 2	356.2	445.9	284.1	345.4	403.0	507.3
56 1	296.7	368.6	242.0	293 3	341.0	426.3	272.9	331.6	386.8	486.3
44 7	283.5	351.8	232.0	281.0	326.5	407.8	262.1	318.4	371.2	466 2
34.0	270.9	335.9	222.5	269.3	312.8	390.3	251.9	305.9	356.4	447.2
24.0	259.1	320.9	213.4	258.3	299.8	373.7	242.2	293.9	342.3	429.1
14.4	248.0	306.8	204.9	247.8	287.5	358.1	232.9	282 5	328 8	411.9
05.4	237.5	294.1	196.7	237.8	275.9	343.2	224.0	271.6	316.1	395.6

ECCENTRIC LOADING OF COLUMNS.

given rectangular cross-section, such as a masonry joint under press-e stress will be distributed uniformly over the section only when the nt passes through the centre of the section; any deviation from such al position will bring a maximum unit pressure to one edge and a um to the other; when the distance of the resultant from one edge is ird of the entire width of the joint, the pressure at the nearer edge is the mean pressure, while that at the farther edge is zero, and that the resultant approaches still nearer to the edge the pressure at the r edge becomes less than zero; in fact, becomes a tension, if the ial (mortar, etc., there is capable of resisting tension. Or, if, as usual sonry joints, the material is practically incapable of resisting tension, ressure at the nearer edge, when the resultant approaches it nearer he third of the width, increases very rapidly and dangerously, becom-eoretically infinite when the resultant reaches the edge.

h a given position of the resultant relatively to one edge of the joint or n, a similar redistribution of the pressures throughout the section may ought about by simply adding to or diminishing the width of the n.

P = the total pressure on any section of a bar of uniform thickness.
= the width of that section = the area of the section, when thickness

$\frac{P}{w}$ = the mean unit pressure on the section.

= the maximum unit pressure on the section.
= the minimum unit pressure on the section.
= the eccentricity of the resultant = its distance from the centre of ection.

en $M = p\left(1 + \frac{6d}{w}\right)$ and $m = p\left(1 - \frac{6d}{w}\right)$.

hen $d = \frac{1}{6}w$ then $M = 2p$ and $m = O$.

hen d is *greater than 1/6w, the resultant* in that case being less than *third of the width from one edge*. p becomes negative. (J. C. Trau *Jr., Engineering News, Nov. 23, 1893.*)

BUILT COLUMNS.

From experiments by T. D. Lovett, discussed by Burr, the values of f an a in several cases are determined, giving empirical forms of Gordon's fo mula as follows: p = pounds crushing strength per square inch of sectio l = length of column in inches, r = radius of gyration in inches.

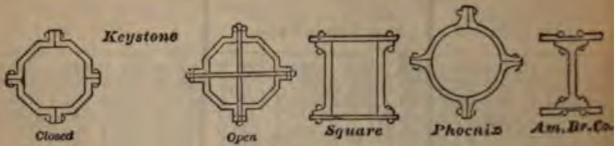


FIG. 76.

Flat Ends.

	Keystone Columns.	Square Columns.	Phœnix Columns.	American Bridge Co. Columns.
$p =$	$\dfrac{39{,}500}{1+\dfrac{1}{18{,}300}\dfrac{l^2}{r^2}}$ (1)	$\dfrac{39{,}000}{1+\dfrac{1}{35{,}000}\dfrac{l^2}{r^2}}$ (4)	$\dfrac{42{,}000}{1+\dfrac{1}{50{,}000}\dfrac{l^2}{r^2}}$ (6)	$\dfrac{36{,}000}{1+\dfrac{1}{46{,}000}\dfrac{l^2}{r^2}}$

Flat Ends, Swelled.

$p =$	$\dfrac{36{,}000}{1+\dfrac{1}{18{,}300}\dfrac{l^2}{r^2}}$ (2)	....	....	....

Pin Ends.

$p =$	....	$\dfrac{39{,}000}{1+\dfrac{1}{17{,}000}\dfrac{l^2}{r^2}}$ (5)	$\dfrac{42{,}000}{1+\dfrac{1}{22{,}700}\dfrac{l^2}{r^2}}$ (7)	$\dfrac{36{,}000}{1+\dfrac{1}{21{,}500}\dfrac{l^2}{r^2}}$ (1

Pin Ends, Swelled.

$p =$	$\dfrac{36{,}000}{1+\dfrac{1}{15{,}000}\dfrac{l^2}{r^2}}$ (3)	....	....	....

Round Ends.

$p =$	....	....	$\dfrac{42{,}000}{1+\dfrac{1}{12{,}500}\dfrac{l^2}{r^2}}$ (8)	$\dfrac{36{,}000}{1+\dfrac{1}{11{,}500}\dfrac{l^2}{r^2}}$ (11

With great variations of stress a factor of safety of as high as 6 or 8 may be used, or it may be as low as 3 or 4, if the condition of stress is uniform or essentially so.

Burr gives the following general principles which govern the resistance of built columns:

The material should be disposed as far as possible from the neutral axis of the cross-section, thereby increasing r;

There should be no initial internal stress;

The individual portions of the column should be mutually supporting;

The individual portions of the column should be so firmly secured to each other that no relative motion can take place, in order that the column may fail as a whole, thus maintaining the original value of r.

Stoney says: "When the length of a rectangular wrought-iron tube column does not exceed 30 times its least breadth, it fails by the bulging buckling of a short portion of the plates, not by the flexure of the pillar as a whole."

In Trans. A. S. C. E., Oct. 1880, are given the following formulæ for the *ultimate resistance of* wrought-iron columns designed by C. Shaler Smith

Flat Ends.

re in.	Phœnix Column.	American Bridge Co. Column.	Common Column.
$\frac{\ldots 00}{\ldots \frac{l^2}{\ldots 0\, d^2}}$ (12)	$\frac{42{,}500}{1+\frac{1}{4500}\frac{l^2}{d^2}}$ (15)	$\frac{36{,}500}{1+\frac{1}{3750}\frac{l^2}{d^2}}$ (18)	$\frac{36{,}500}{1+\frac{1}{2700}\frac{l^2}{d^2}}$ (21)

One Pin End.

$\frac{\ldots 500}{\ldots \frac{l^2}{\ldots 00\, d^2}}$ (13)	$\frac{40{,}000}{1+\frac{1}{2250}\frac{l^2}{d^2}}$ (16)	$\frac{36{,}500}{1+\frac{1}{2250}\frac{l^2}{d^2}}$ (19)	$\frac{36{,}500}{1+\frac{1}{1500}\frac{l^2}{d^2}}$ (22)

Two Pin Ends.

$\frac{\ldots 500}{\ldots \frac{l^2}{\ldots 00\, d^2}}$ (14)	$\frac{36{,}600}{1+\frac{1}{1500}\frac{l^2}{d^2}}$ (17)	$\frac{36{,}500}{1+\frac{1}{1750}\frac{l^2}{d^2}}$ (20)	$\frac{36{,}500}{1+\frac{1}{1200}\frac{l^2}{d^2}}$ (23)

nmon " column consists of two channels, opposite, with flanges ith a plate on one side and a lattice on the other.

ula for "square" columns may be used without much error for n-chord section composed of two channel-bars and plates, with the pin passing through the centre of gravity of the cross-Burr).

sion members composed of two channels connected by zigzag ay be treated by formulæ 4 and 5, using $f = 36{,}000$ instead of

ents on full-sized Phœnix columns in 1873 showed a close agree-e results with formulæ 6–8. Experiments on full-sized Phœnix the Watertown testing-machine in 1881 showed considerable dis-when the value of $l \div r$ became comparatively small. The fol-dified form of Gordon's formula gave tolerable results through ange of experiments:

$$\text{columns, flat end,}\quad p = \frac{40{,}000\left(1+\frac{2r}{l}\right)}{1+\frac{1}{50{,}000}\frac{l^2}{r^2}}. \qquad (24)$$

results of three series of experiments on Phœnix columns, a le formula than Gordon's is reached as follows:

lumns, flat ends, $p = 39{,}640 - 46\frac{l}{r}$, when $l \div r$ is from 30 to 140;

$$p = 64{,}700 - 4600\sqrt{\frac{l}{r}} \text{ when } l \div r \text{ is less than 30.}$$

Dimensions of Phœnix Columns.

(Phœnix Iron Co.)

ensions are subject to slight variations, which are unavoidable in shapes.

hts of columns given are those of the 4, 6, or 8 segments of are composed. The *rivet-heads* add from 2 to 5 per cent to the ven. Rivets are spaced 3, 4, or 6 inches apart from centre to somewhat more closely at the ends than towards the centre of

ns have 8 segments. *E* columns 6 segments, *C*, *B*[2], *B*[1], and *A* have . *Least radius of gyration* $= D \times .3636$.

One Segment.		Diameters in inches.			One Column.		
Thickness in inches.	Weight in lbs. per yard.	d inside.	D Outside.	D^1 Over Flanges.	Area of Cross-section, sq. inches	Weight per ft. in pounds.	Least Radius of Gyration, in inches.
3-16	9½	A 3⅝	4	6 1-16	3.8	12.6	1.45
¼	12		4⅛	6 3-16	4.8	16.0	1.50
5-16	14½		4¼	6 5-16	5.8	19.3	1.55
⅜	17		4⅜	6 7-16	6.8	22.6	1.59
¼	16	B^1 4 13/16	5 5-16	8 1-16	6.4	21.3	1.92
5-16	19½		5 7-16	8⅛	7.8	26.0	1.96
⅜	23		5 9-16	8¼	9.2	30.6	2.02
7-16	26½		5 11-16	8⅜	10.6	35.3	2.07
½	30		5 13-16	8 7-16	12.0	40.0	2.11
9-16	33½		5 15-16	8½	13.4	44.6	2.16
⅝	37		6 1-16	8⅝	14.8	49.3	2.20
¼	18½	B_2 5 15/16	6 7-16	9⅛	7.4	24.6	2.34
5-16	22½		6 9-16	9¼	9.0	30.0	2.39
⅜	26½		6 11-16	9 5-16	10.6	35.3	2.43
7-16	30½		6 13-16	9⅜	12.2	40.6	2.48
½	34½		6 15-16	9½	13.8	46.0	2.52
9-16	38½		7 1-16	9⅝	15.4	51.3	2.57
⅝	42½		7 3-16	9 11-16	17.0	56.6	2.61
¼	25½	C 7 3/16	7 11-16	11 9-16	10.2	34.	2.80
5-16	31		7 13-16	11⅝	12.4	41.3	2.85
⅜	36		7 15-16	11 11-16	14.4	48.0	2.90
7-16	41		8 1-16	11¾	16.4	54.6	2.94
½	46		8 3-16	11 13-16	18.4	61.3	2.98
9-16	51		8 5-16	11⅞	20 4	68.	3.03
⅝	56		8 7-16	12	22.4	74.6	3.08
11-16	62		8 9-16	12 1-16	24.8	82.6	3.12
¾	68		8 11-16	12 3-16	27.2	90.6	3.16
13-16	73		8 13-16	12 5-16	29.2	97.3	3.21
⅞	78		8 15-16	12 7-16	31.2	104.	3.26
1	89		9 3-16	12 9-16	35.6	118.6	3.34
1⅛	99		9 7-16	12¾	39.6	132.	3.43
1¼	109		9 11-16	12 15-16	43.6	145.3	3.52
¼	28	E 11	11½	15 7-16	16.8	56.	4.18
5-16	32½		11⅝	15 9-16	19.5	65.	4.23
⅜	37		11¾	15 11-16	22.2	74.	4.28
7-16	42		11⅞	15 13-16	25.2	84.	4.32
½	47		12	15⅞	28.2	94.	4.36
9-16	52		12⅛	16	31.2	104.	4.40
⅝	57		12¼	16 1-16	34.2	114.	4 45
11-16	62		12⅜	16 3-16	37.2	124.	4.50
¾	68		12½	16 5-16	40.8	136.	4.55
13-16	73		12⅝	16 7-16	43.8	146.	4.60
⅞	78		12¾	16⅝	46.8	156.	4.64
1	88		13	16¾	52.8	176.	4.73
1⅛	98		13¼	17	58 8	196.	4.82
1¼	108		13½	17 3-16	64.8	216.	4.91
5-16	31	G 14⅜	15	19⅛	24.8	82.6	5.45
⅜	36		15⅛	19¼	28.8	96.	5.50
7-16	41		15¼	19⅜	32.8	109.3	5.55
[illegible]	46		15⅜	19 7-16	36.8	122 6	5.59
[illegible]	[illegible]		15½	19½	40.8	136.	5.63
[illegible]	[illegible]		15⅝	19⅝	44.8	149.3	5.68

Segment.		Diameters in inches.			One Column.			Safe Load in net tons for 16-feet Lengths.
	Weight in lbs. per yard.	d inside.	D Outside.	D^1 Over Flanges.	Area of Cross-Section, sq. inches.	Weight per ft. in pounds.	Least Radius of Gyration, in inches.	
5	61		15¾	19¾	48.8	162.6	5.72	325.21
	66		15⅞	19⅞	52.8	176.	5.77	352.02
6	71		16	20	56.8	189.3	5.82	378.85
	76	G	16⅛	20⅛	60.8	202.6	5 87	405.70
	86	14⅜	16⅜	20⅜	68.8	229.3	5.95	464.38
	96		16⅝	20⅝	76.8	256.	6.04	513.17
	106		16⅞	20¾	84.8	282.6	6.14	567.06
	116		17⅛	21	92.8	309.3	6.23	620.98

rking Formulæ for Wrought-iron and Steel Struts arious Forms.—Burr gives the following practical formulæ, which lieves to possess advantages over Gordon's:

Kind of Strut.	p = Ultimate Strength, lbs. per sq. in. of Section.	p_1 = Working Strength = 1/5 Ultimate, lbs. per sq. in. of Section.
nd fixed end iron angles and tees	$44000-140\frac{l}{r}$ (1)	$8800-28\frac{l}{r}$ (2)
d-end iron angles and tees.......	$46000-175\frac{l}{r}$ (3)	$9200-35\frac{l}{r}$ (4)
nd iron channels and I beams....	$40000-110\frac{l}{r}$ (5)	$8000-22\frac{l}{r}$ (6)
nd mild-steel angles............	$52000-180\frac{l}{r}$ (7)	$10400-36\frac{l}{r}$ (8)
nd high-steel angles..............	$76000-290\frac{l}{r}$ (9)	$15200-58\frac{l}{r}$ (10)
d solid wrought-iron columns....	$\left.\begin{array}{l}32000-80\frac{l}{r}\\ 32000-277\frac{l}{d}\end{array}\right\}$ (11)	$\left.\begin{array}{l}6400-16\frac{l}{r}\\ 6400-55\frac{l}{d}\end{array}\right\}$ (12)

uations (1) to (4) are to be used only between $\frac{l}{r}=40$ and $\frac{l}{r}=200$

" (5) and (6) " " " " " " " = 20 " " = 200
" (7) to (10) " " " " " " " = 40 " " = 200
" (11) and (12) " " " " " " " = 20 " " = 200

or $\frac{l}{d}=6$ and $\frac{l}{d}=65$

l columns, properly made, of steel ranging in specimens from 65,000 to lbs. per square inch should give a resistance 25 to 33 per cent in ex- f that of wrought-iron columns with the same value of $l \div r$, provided atio does not exceed 140.

unsupported width of a plate in a compression member should not d 30 times its thickness.

built columns the transverse distance between centre lines of rivets ing plates to angles or channels, *etc.*, *should not* exceed 35 times the thickness. If *this width is exceeded*, longitudinal buckling of the plate takes place, and the column ceases to fail as a whole, bu detail.

The same tests show that the thickness of the leg of an angl latticing is riveted should not be less than 1/9 of the length of t side if the column is purely and wholly a compression member. ' limit may be passed somewhat in stiff ties and compression me signed to carry transverse loads.

The panel points of latticing should not be separated by a great than 60 times the thickness of the angle-leg to which the latticing if the column is wholly a compression member.

The rivet pitch should never exceed 16 times the thickness of th metal pierced by the rivet, and if the plates are very thick it sh nearly equal that value.

Merriman's Rational Formula for Columns (*E* July 19, 1894).

$$C = \frac{B}{1 - \frac{nB}{\pi^2 E}\frac{l^2}{r^2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad .$$

$$B = \frac{C}{1 + \frac{nC}{\pi^2 E}\frac{l^2}{r^2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad .$$

B = unit-load on the column = total load $P \div$ area of cross-
C = maximum compressive unit-stress on the concave side of th
l = length of the column; r = least radius of gyration of the cro
E = coefficient of elasticity of the material; $n = 1$ for both en
$n = 4/9$ for one end round and one fixed; $n = \frac{1}{4}$ for both ends fi
formula is for use with strains within the elastic limit only:
hold good when the strain C exceeds the elastic limit.

Prof. Merriman takes the mean value of E for timber = 1,500,0 iron = 15,000,000, for wrought-iron = 25,000,000, and for steel = and $\pi^2 = 10$ as a close enough approximation. With these value putes the following tables from formula (1):

I.—Wrought-iron Columns with Round En

Unit-load.	Maximum Compressive Unit-stress C.						
$\frac{P}{A}$ or B.	$\frac{l}{r} = 20$	$\frac{l}{r} = 40$	$\frac{l}{r} = 60$	$\frac{l}{r} = 80$	$\frac{l}{r} = 100$	$\frac{l}{r} = 120$	$\frac{l}{r} =$ 1
5,000	5,040	5,170	5,390	5,730	6,250	6,980	8,2
6,000	6,055	6,240	6,560	7,090	7,890	9,090	11,3
7,000	7,080	7,330	7,780	8,530	9,720	11,610	15,51
8,000	8,100	8,430	9,040	10,060	11,660	14,640	21,4
9,000	9,130	9,550	10,340	11,690	14,060	18,380	......
10,000	10,160	10,680	11,680	13,440	16,670	23,090	......
11,000	11,200	11,750	13,070	15,310	19,640	....	......
12,000	12,240	13,000	14,500	17,320	23,080	....	......
13,000	13,280	14,180	15,990	19,480	....	....	......

6,010	6,060	6,130	6,240	6,380	6,570	6,800	7,090
7,020	7,080	7,180	7,330	7,530	7,780	8,110	8,530
8,025	8,100	8,240	8,430	8,700	9,040	9,490	10,060
9,030	9,130	9,300	9,550	9,890	10,340	10,930	11,690
10,040	10,160	10,370	10,710	11,110	11,680	12,440	13,440
11,050	11,200	11,450	11,830	12,360	13,070	14,020	15,310
12,060	12,240	12,540	13,000	13,640	14,510	15,690	17,320
13,070	13,280	13,640	14,210	14,940	15,990	17,440	19,480
14,080	14,320	14,740	15,380	16,280	17,530	19,290	21,820

III.—Steel Columns with Round Ends.

Maximum Compressive Unit-stress C.

$\frac{l}{r} = 20$	$\frac{l}{r} = 40$	$\frac{l}{r} = 60$	$\frac{l}{r} = 80$	$\frac{l}{r} = 100$	$\frac{l}{r} = 120$	$\frac{l}{r} = 140$	$\frac{l}{r} = 160$
6,050	6,200	6,470	6,880	7,500	8,430	9,870	12,300
7,070	7,270	7,650	8,230	9,130	10,540	12,900	17,400
8,090	8,380	8,770	9,650	10,870	12,990	16,760	24,590
9,110	9,450	10,090	11,140	12,850	15,850	20,930	....
10,130	10,560	11,360	12,710	15,000	19,230	28,850	....
11,160	11,690	12,670	14,370	17,370	23,300	....	....
12,200	12,820	14,020	16,130	20,000	28,300	....	....
13,330	13,970	15,400	18,000	22,940	....	....	....
14,250	15,130	16,830	19,960	26,250	....	....	....

IV.—Steel Columns with Fixed Ends.

Maximum Compressive Unit-stress C.

$\frac{l}{r} = 20$	$\frac{l}{r} = 40$	$\frac{l}{r} = 60$	$\frac{l}{r} = 80$	$\frac{l}{r} = 100$	$\frac{l}{r} = 120$	$\frac{l}{r} = 140$	$\frac{l}{r} = 160$

computing C by formula (1). If the agreement between the speci computed values is not sufficiently close, new dimensions must be and the computation be repeated. By the use of the above tables t will be shortened.

The formula (1) may be put in another form which in some cases breviate the numerical work. For B substitute its value $P \div A$, Ar^2 write I, the least moment of inertia of the cross-section; then

$$I - \frac{P}{C}r^2 = \frac{nPl^2}{\pi^2 E}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad .$$

in which I and r^2 are to be determined.

For example, let it be required to find the size of a square oak with fixed ends when loaded with 24,000 lbs. and 16 ft. long, so maximum compressive stress C shall be 1000 lbs. per square inch $I = 24{,}000$, $C = 1000$, $n = \frac{1}{4}$, $\pi^2 = 10$, $E = 1{,}500{,}000$, $l = 16 \times 12$, an comes

$$I - 24r^2 = 14.75.$$

Now let x be the side of the square; then

$$I = \frac{x^4}{12} \quad \text{and} \quad r^2 = \frac{x^2}{12},$$

so that the equation reduces to $x^4 - 24x^2 = 177$, from which x^2 is fou 29.92 sq. in., and the side $x = 5.47$ in. Thus the unit-load B is a lbs. per square inch.

WORKING STRAINS ALLOWED IN BRIDG MEMBERS.

Theodore Cooper gives the following in his Bridge Specifications:

Compression members shall be so proportioned that the maxim shall in no case cause a greater strain than that determined by th ing formula:

$P = \dfrac{8000}{1 + \dfrac{l^2}{40{,}000r^2}}$ for square-end compression members;

$P = \dfrac{8000}{1 + \dfrac{l^2}{30{,}000r^2}}$ for compression members with one pin and one squ

$P = \dfrac{8000}{1 + \dfrac{l^2}{20{,}000r^2}}$ for compression members with pin-bearings;

(These values may be increased in bridges over 150 ft. span. See Specifications.)

P = the allowed compression per square inch of cross-section;
l = the length of compression member, in inches;
r = the least radius of gyration of the section in inches.

No compression member, however, shall have a length exceeding its least width.

The Phœnix Bridge Company give the following:

The greatest working stresses in wrought-iron compression mem spans 150 feet in length and under shall be the following:

	Flat Ends.	Pin En
Phœnix column....................	$P = \dfrac{8400}{1 + \dfrac{l^2}{50{,}000r^2}}$	$P = \dfrac{84}{1 + \dfrac{}{30}}$
Latticed or common column......	$P = \dfrac{8000}{1 + \dfrac{l^2}{40{,}000r^2}}$	$P = \dfrac{7}{1 + \dfrac{}{3}}$
....	$P = 9000 - 30\dfrac{l}{r}$	$P = 9000 -$

ords shall be proportioned by the flat-end formula.
etween flat-end and pin-end results shall be used for one pin end end.
nd transverse struts shall be designed by taking working stresses e and four tenths those given by the preceding formulæ.

Stresses allowed in Bridge Tension Members.

(Theodore Cooper's Specifications.)

of the structure shall be so proportioned that the maximum in no case cause a greater tension than the following (except in eding 150 feet):

	Pounds per sq. in.
ral bracing	15,000
l rolled beams, used as cross floor-beams and stringers.	9,000
om chords and main diagonals (forged eye-bars)	10,000
om chords and main diagonals (plates or shapes), net tion	8,000
nter rods and long verticals (forged eye-bars)	8,000
nter and long verticals (plates or shapes), net section..	6,500
om flange of riveted cross-girders, net section	8,000
om flange of riveted longitudinal plate girders over t. long, net section	8,000
om flange of riveted longitudinal plate girders under t. long, net section	7,000
r-beam hangers, and other similar members liable to den loading (bar iron with forged ends)	6,000
r-beam hangers, and other similar members liable to den loading (plates or shapes), net section	5,000

subject to alternate strains of tension and compression shall be ed to resist each kind of strain. Both of the strains shall, how- nsidered as increased by an amount equal to 8/10 of the least of rains, for determining the sectional area by the above allowed

enix Bridge Company specify: The greatest working stresses in it-iron tensile members of railway spans 150 feet in length and ll be as follows:

	Pounds per sq. in.
nter web members	8,000
verticals	8,000
n-web and lower-chord members (eye-bars)	10,000
ension loops	7,000
ension plates (net section)	7,000
ion members of lateral and transverse bracing	15,000
nter rods and long verticals of lattice girders (net sec- n)	7,000
er chords and main tension members o' lattice girders t section)	8,000
om flange of plate girders (net section)	8,000
om flange of rolled beams	8,000
le-iron lateral ties (net section)	12,000

over 150 feet in length, the greatest working tensile stresses per h of wrought iron, lower-chord and end main-web eye-bars, shall

$$8000\left(1+0.9\times\frac{\text{min. total stress}}{\text{max. total stress}}\right)$$

this quantity exceeds 10,000.

Working Stresses for Steel.

test allowed working stresses for steel tension members, for 0 feet in length *and less, shall be as follows:*

	Pounds sq. in
In counter web members	10,00
In long verticals	10,00
In all main-web and lower-chord eye-bars	12,0
In plate hangers (net section)	9,00
In tension members of lateral and transverse bracing	19,00
In steel-angle lateral ties (net section)	15,00

For spans over 200 feet in length the greatest allowed working str per square inch, in lower-chord and end main-web eye-bars, shall be tak

$$10{,}000\left(1+\frac{\text{min. total stress}}{\text{max. total stress}}\right)$$

whenever this quantity exceeds 13,200.

The greatest allowable stress in the main-web eye-bars nearest the c of such spans shall be taken at 13,200 pounds per square inch; and for the intermediate eye-bars shall be found by direct interpolation bet the preceding values.

The greatest allowable working stresses in steel plate and lattice gi and rolled beams shall be taken as follows:

	Pounds sq. in
Upper flange of plate girders (gross section)	10,00
Lower flange of plate girders (net section)	10,00
In counters and long verticals of lattice girders (net section)	9,00
In lower chords and main diagonals of lattice girders (net section)	10,00
In bottom flanges of rolled beams	10,00
In top flanges of rolled beams	10,00

RESISTANCE OF HOLLOW CYLINDERS TO COLLAPSE.

Fairbairn's empirical formula (*Phil. Trans.* 1858) is

$$p = 9{,}675{,}600\,\frac{t^{2.19}}{ld} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad .$$

where p = pressure in lbs. per square inch, t = thickness of cylinder diameter, and l = length, all in inches; or,

$$p = 806{,}600\,\frac{t^{2.19}}{Ld}, \text{ if } L \text{ is in feet.} \quad . \quad . \quad .$$

He recommends the simpler formula

$$p = 9{,}675{,}600\,\frac{t^2}{ld} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad .$$

as sufficiently accurate for practical purposes, for tubes of conside diameter and length.

The diameters of Fairbairn's experimental tubes were 4″, 6″, 8″, 10″ 12″, and their lengths, between the cast-iron ends, ranged between 19 in and 60 inches.

His formula (3) has been generally accepted as the basis of rules ascertaining the strength of boiler flues. In some cases, however, limit fixed to its application by a supplementary formula.

Lloyd's Register contains the following formula for the strength of cir boiler-flues, viz.,

$$P = \frac{89{,}600t^2}{Ld} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad .$$

The English Board of Trade prescribes the following formula for cir flues, when the longitudinal joints are welded, or made with riveted straps, viz.,

$$P = \frac{90{,}000t^2}{(L+1)d} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad .$$

For lap- and for inferior workmanship the numerical factor m *redu* 60,000.

Lloyd's Register, as well as those of the Board of Trade, pre- that in no case the value of P must exceed the amount given ig equation, viz.,

$$P = \frac{8000t}{d}. \quad \ldots \ldots \ldots \quad (6)$$

(4), (5), (6) P is the highest working pressure in pounds per and d are the thickness and diameter in inches, L is the lue in feet measured between the strengthening rings, in case such. Formula (4) is the same as formula (3), with a factor In formula (5) the length L is increased by 1; the influence dition has on the value of P is, of course, greater for short long ones.

s deduced from Fairbairn's experiments the following formula sing strength of flues:

$$p = \frac{4Tt^2}{d\sqrt{L}}, \quad \ldots \ldots \ldots \quad (7)$$

d d have the same meaning as in formula (1), L is the length in the tensile strength of the metal in pounds per square inch. to T the value 50,000, and express the length of the flue in ion (7) assumes the following form, viz.,

$$p = 692{,}800 \frac{t^2}{d\sqrt{l}}. \quad \ldots \ldots \ldots \quad (8)$$

siders a factor of safety of 4 sufficient in applying his formula. Treatise on Steam Engineering," by J. W. Nystrom, p. 106.)), (4), and (8) have the common defect that they make the essure decrease indefinitely with increase of length, and *vice* ove has deduced from Fairbairn's experiments an equation of rm, which, reduced to English measures, is as follows, viz.,

$$p = 5{,}358{,}150 \frac{t^2}{ld} + 41{,}906 \frac{t^2}{d} + 1323 \frac{t}{d}, \quad \ldots \ldots \quad (9)$$

tation is the same as in formula (1).

t, in his "Manual of Rules," etc., p. 696, gives the dimensions of ected from the reports of the Manchester Steam-Users Associa- which collapsed while in actual use in boilers. These flues 24 to 60 inches in diameter, and from 3-16 to ⅝ inch in thickness. ed of rings of plates riveted together, with one or two longitud- ut all of them unfortified by intermediate flanges or strength- At the collapsing pressures the flues experienced compressions 1.53 to 2.17 tons, or a mean compression of 1.82 tons per square ion. From these data Clark deduced the following formula rage resisting force of common boiler-flues," viz.,

$$p = t^2\left(\frac{50{,}000}{d} - 500\right), \quad \ldots \ldots \ldots \quad (10)$$

e collapsing pressure in pounds per square inch, and d and t eter and thickness expressed in inches.

ter, in *Van Nostrand's Magazine*, March, 1881, discussing the her formulæ, shows that experimental data are as yet insuffi- rmine the value of any of the formulæ. He says that Nystrom's gives a closer agreement of the calculated with the actual col- ures in experiments on flues of every description than any of mulæ.

sing Pressure of Plain Iron Tubes or Flues.

(Clark, S. E., vol. i. p. 643.)

nce to collapse of plain-riveted flues is directly as the square of of the plate, and inversely as the square of the diameter. The e two ends of the flue does not *practically* extend over a length ter than *twice or three times the diameter*. The collapsing *long flues is therefore practically* independent of the length.

Instances of collapsed flues of Cornish and Lancashire boilers col Clark, showed that the resistance to collapse of flues of 3/8-inch plat 43 feet long, and 30 to 50 inches diameter, varied as the 1.75 power diameter. Thus,

for diameters of........................	30	35	40	45	50	inches,
the collapsing pressures were..........	76	58	45	37	30	lbs. per s
for 7-16-inch plates the collapsing pressures were......................	..	..	60	49	42	" "

For collapsing pressures of plain iron flue-tubes of Cornish and shire steam-boilers, Clark gives:

$$P = \frac{200,000t^2}{d^{1.75}}.$$

P = collapsing pressure, in pounds per square inch;
t = thickness of the plates of the furnace tube, in inches.
d = internal diameter of the furnace tube, in inches.

For short lengths the longitudinal tensile resistance may be effec augmenting the resistance to collapse. Flues efficiently fortified by joints or hoops at intervals of 3 feet may be enabled to resist from to 60 lbs. or 70 lbs. pressure per square inch more than plain tubes, ing to the thickness of the plates.

Strength of Small Tubes.—The collapsing resistance of drawn tubes of small diameter, and from .134 inch to .109 inch in thi has been tested experimentally by Messrs. J. Russell & Sons. The for wrought-iron tubes varied from 14.33 to 20.07 tons per square-in tion of the metal, averaging 18.20 tons, as against 17.57 to 24.28 tons, ing 22.40 tons, for the bursting pressure.

(For strength of Segmental Crowns of Furnaces and Cylinders see S. E., vol. i, pp. 649-651 and pp. 627, 628.)

Formula for Corrugated Furnaces (*Eng'g*, July 24, 1 '92).—As the result of a series of experiments on the resistance to c of Fox's corrugated furnaces, the Board of Trade and Lloyd's R altered their formulæ for these furnaces in 1891 as follows:

Board of Trade formula is altered from

$$\frac{12,500 \times T}{D} = WP \text{ to } \frac{14,000 \times T}{D} = WP.$$

T = thickness in inches;
D = mean diameter of furnace;
WP = working pressure in pounds per square inch.

Lloyd's formula is altered from

$$\frac{1000 \times (T^2)}{D} = WP \text{ to } \frac{1234 \times (T^2)}{D} = WP.$$

T = thickness in sixteenths of an inch;
D = greatest diameter of furnace;
WP = working pressure in pounds per square inch.

TRANSVERSE STRENGTH.

In transverse tests the strength of bars of rectangular section is fo vary directly as the breadth of the specimen tested, as the square depth, and inversely as its length. The deflection under any load va the cube of the length, and inversely as the breadth and as the cube depth. Represented algebraically, if S = the strength and D the defl l the length, b the breadth, and d the depth,

$$S \text{ varies as } \frac{bd^2}{l} \text{ and } D \text{ varies as } \frac{l^3}{bd^3}.$$

the purpose of reducing the strength of pieces of various n standard, the term *modulus of rupture* (represented b value is obtained by experiment on a bar of rectangular ed at the ends and loaded in the middle and substituting numerical n the following formula :

$$R = \frac{3}{2}\frac{Pl}{bd^2},$$

n $P =$ the breaking load in pounds, $l =$ the length in inches, b the , and d the depth.

odulus of rupture is sometimes defined as the strain at the instant ure upon a unit of the section which is most remote from the neutral the side which first ruptures. This definition, however, is based theory which is yet in dispute among authorities, and it is better to t as a numerical value, or experimental constant, found by the ap-n of the formula above given.

the above formula, making l 12 inches, and b and d each 1 inch, it that the modulus of rupture is 18 times the load required to break a inch square, supported at two points one foot apart, the load being l in the middle.

$$\text{icient of transverse strength} = \frac{\text{span in feet} \times \text{load at middle in lbs.}}{\text{breadth in inches} \times (\text{depth in inches})^2}.$$

$$= \frac{1}{18}\text{th of the modulus of rupture.}$$

ndamental Formulæ for Flexure of Beams (Merriman).

sting shear = vertical shear;

sting moment = bending moment;

of tensile stresses = sum of compressive stresses;

sting shear = algebraic sum of all the vertical components of the in-stresses at any section of the beam.

be the area of the section and S_s the shearing unit stress, then resist-ear $= AS_s$; and if the vertical shear $= V$, then $V = AS_s$.

vertical shear is the algebraic sum of all the external vertical forces e side of the section considered. It is equal to the reaction of one sup-considered as a force acting upward, minus the sum of all the vertical ward forces acting between the support and the section.

resisting moment = algebraic sum of all the moments of the inter-rizontal stresses at any section with reference to a point in that sec-$= \frac{SI}{c}$, in which $S =$ the horizontal unit stress, tensile or compressive case may be, upon the fibre most remote from the neutral axis, $c =$ hortest distance from that fibre to said axis, and $I =$ the moment of a of the cross-section with reference to that axis.

bending moment M is the algebraic sum of the moment of the ex-l forces on one side of the section with reference to a point in that sec-= moment of the reaction of one support minus sum of moments of between the support and the section considered.

$$M = \frac{SI}{c}.$$

bending moment is a compound quantity = product of a force by the nce of its point of application from the section considered, the distance measured on a line drawn from the section perpendicular to the tion of the action of the force.

ncerning the above formula, Prof. Merriman, *Eng. News*, July 21, 1894, The formula just quoted is true when the unit-stress S on the part of eam farthest from the neutral axis is within the elastic limit of the rial. It is not true when this limit is exceeded, because then the neutral does not pass through the centre of gravity of the cross-section, and use also the different longitudinal stresses are not proportional to their nces from that axis, these two requirements being involved in the de-on of the formula. But in all cases of design the permissible unit-es should not exceed the elastic limit, and hence the formula applies ally, without regarding the ultimate strength of the material or any circumstances regarding rupture. Indeed so great reliance is placed this formula that the practice of testing beams by rupture has been t entirely abandoned, and the *allowable* unit-stresses are mainly [illegible] from tensile *and compressive tests.*

GENERAL FORMULÆ FOR TRANSVERSE STRENGTH OF BEAMS OF UNIFORM CROSS-SECTION.

Beam.	Rectangular Beam.		Beam of any Section.		
	Breaking Load.	Deflection for Load P or W.	Maximum Moment of Stress.	Moment of Rupture.	Deflection. Δ
Fixed at one end, load at the other...........	$P = \frac{1}{6}\frac{Rbd^2}{l}$	$\frac{4Pl^3}{Ebd^3}$	$Pl =$	$\frac{RI}{c}$	$\frac{1}{3}\frac{Pl^3}{EI}$
Same with load distributed uniformly.........	$W = \frac{1}{3}\frac{Rbd^2}{l}$	$\frac{3}{2}\frac{Wl^3}{Ebd^3}$	$\frac{1}{2}Wl =$	$\frac{RI}{c}$	$\frac{1}{8}\frac{Wl^3}{EI}$
Supported at ends, loaded in middle..........	$P = \frac{2}{3}\frac{Rbd^2}{l}$	$\frac{Pl^3}{4Ebd^3}$	$\frac{1}{4}Pl =$	$\frac{RI}{c}$	$\frac{1}{48}\frac{Pl^3}{EI}$
Same loaded uniformly.......................	$W = \frac{4}{3}\frac{Rbd^2}{l}$	$\frac{5}{32}\frac{Wl^3}{Ebd^3}$	$\frac{1}{8}Wl =$	$\frac{RI}{c}$	$\frac{5}{384}\frac{Wl^3}{EI}$
Same, loaded at middle, and also with uniform load, ...	$2P+W = \frac{4}{3}\frac{Rbd^2}{l}$	$\frac{1}{4}\left(P+\frac{1}{8}W\right)\frac{l^3}{Ebd^3}$	$\left(\frac{1}{4}P+\frac{1}{8}W\right)l =$	$\frac{RI}{c}$	$\frac{1}{48}\left(P+\frac{5}{8}W\right)\frac{l^3}{EI}$
Fixed at both ends, loaded in middle.........	$P = \frac{4}{3}\frac{Rbd^2}{l}$	$\frac{1}{16}\frac{Pl^3}{Ebd^3}$	$\frac{1}{8}Pl =$	$\frac{RI}{c}$	$\frac{P}{192}\frac{l^3}{EI}$
Same, Barlow's Experiments.................	$P = \frac{Rbd^2}{l}$		$\frac{1}{6}Pl =$	$\frac{RI}{c}$	
Same, uniformly loaded......................	$W = \frac{2Rbd^2}{l}$	$\frac{1}{32}\frac{Wl^3}{Ebd^3}$	$\frac{1}{12}Wl =$	$\frac{RI}{c}$	$\frac{W}{384}\frac{l^3}{EI}$
Fixed at one end, supported at the other, loaded at .634l from fixed end, ...		$\frac{.1148Pl^3}{Ebd^3}$	$\frac{3}{8}\left(2\sqrt{3}-3\right)Pl =$	$\frac{RI}{c}$	$\frac{P}{105}\frac{l^3}{EI}$ (nearly).
Same uniformly loaded.......................	$W = \frac{4}{3}\frac{Rbd^2}{l}$	$\frac{.0648Wl^3}{Ebd^3}$	$\frac{1}{8}Wl =$	$\frac{RI}{c}$	$\frac{W}{185}\frac{l^3}{EI}$ (nearly).

ulæ for Transverse Strength of Beams.—Referring to
receding page,
ad at middle;
tal load, distributed uniformly;
ngth, b = breadth, d = depth, in inches;
odulus of elasticity;
odulus of rupture, or stress per square inch of extreme fibre;
oment of inertia;
stance between neutral axis and extreme fibre.

eaking load of circular section, replace bd^2 by $0.59d^3$.
od wrought iron the value of R is about 80,000, for steel about 120,000,
entage of carbon apparently having no influence. (Thurston, Iron
l, p. 491).
st iron the value of R varies greatly according to quality. Thurston
,740 and 67,980 in No. 2 and No. 4 cast iron, respectively.
ams fixed at both ends and loaded in the middle, Barlow, by experi-
und the maximum moment of stress = $1/6Pl$ instead of $\frac{1}{8}Pl$, the
ven by theory. Prof. Wood (Resist. Matls. p. 155) says of this case:
nomena are of too complex a character to admit of a thorough and
alysis, and it is probably safer to accept the results of Mr. Barlow
ce than to depend upon theoretical results.

OXIMATE GREATEST SAFE LOADS IN LBS. ON STEEL BEAMS. (Pencoyd Iron Works.)

on fibre strains of 16,800 lbs. for steel. (For iron the loads should be
h less, corresponding to a fibre strain of 14,000 lbs. per square inch).
ength in feet between supports; a = interior area in square inches;
ectional area of beam in square inches; d = interior depth in inches.
epth of beam in inches. w = working load in net tons.

of ...n.	Greatest Safe Load in Pounds		Deflection in Inches.	
	Load in Middle.	Load Distributed.	Load in Middle.	Load Distributed.
...ect-	$\frac{940AD}{L}$	$\frac{1880AD}{L}$	$\frac{wL^3}{32AD^2}$	$\frac{wL^3}{52AD^2}$
Rect-	$\frac{940(AD-ad)}{L}$	$\frac{1880(AD-ad)}{L}$	$\frac{wL^3}{32(AD^2-ad^2)}$	$\frac{wL^3}{52(AD^2-ad^2)}$
...lin	$\frac{700AD}{L}$	$\frac{1400AD}{L}$	$\frac{wL^3}{24AD^2}$	$\frac{wL^3}{38AD^2}$
...ler.	$\frac{700(AD-ad)}{L}$	$\frac{1400(AD-ad)}{L}$	$\frac{wL^3}{24(AD^2-ad^2)}$	$\frac{wL^3}{38(AD^2-ad^2)}$
...gged or	$\frac{930AD}{L}$	$\frac{1860AD}{L}$	$\frac{wL^3}{32AD^2}$	$\frac{wL^3}{52AD^2}$
or	$\frac{1600AD}{L}$	$\frac{3200AD}{L}$	$\frac{wL^3}{53AD^2}$	$\frac{wL^3}{85AD^2}$
...eam	$\frac{1450AD}{L}$	$\frac{2900AD}{L}$	$\frac{wL^3}{50AD^2}$	$\frac{wL^3}{80AD^2}$
	$\frac{1780AD}{L}$	$\frac{3560AD}{L}$	$\frac{wL^3}{58AD^2}$	$\frac{wL^3}{93AD^2}$
	II	III	IV	V

The above formulæ for the strength and stiffness of rolled beams [illegible]rious sections are intended for convenient application in cases [illegible] strict accuracy is not required.

The rules for rectangular and circular sections are correct, while tho[illegible] the flanged sections are approximate, and limited in their application [illegible] standard shapes as given in the Pencoyd tables. When the section [illegible] beam is increased above the standard minimum dimensions, the flang[illegible] maining unaltered, and the web alone being thickened, the tendency [illegible] for the load as found by the rules to be in excess of the actual; but [illegible] the limits that it is possible to vary any section in the rolling, the[illegible] will apply without any serious inaccuracy.

The calculated safe loads will be approximately one half of loads [illegible] would injure the elasticity of the materials.

The rules for deflection apply to any load below the elastic limit, o[illegible] than double the greatest safe load by the rules.

If the beams are long without lateral support, reduce the loads fo[illegible] ratios of width to span as follows:

Length of Beam.	Proportion of Calculated Loa[illegible] forming Greatest Safe Load.
20 times flange width.	Whole calculated load.
30 " " "	9-10 " "
40 " " "	8-10 " "
50 " " "	7-10 " "
60 " " "	6-10 " "
70 " " "	5-10 " "

These rules apply to beams supported at each end. For beams supp[illegible] otherwise, alter the coefficients of the table as described below, referri[illegible] the respective columns indicated by number.

Changes of Coefficients for Special Forms of Beam[illegible]

Kind of Beam.	Coefficient for Safe Load.	Coefficient for De[illegible]tion.
Fixed at one end, loaded at the other.	One fourth of the coefficient, col. II.	One sixteenth of th[illegible] efficient of col. IV[illegible]
Fixed at one end, load evenly distributed.	One fourth of the coefficient of col. III.	Five forty-eighths [illegible] coefficient of col[illegible]
Both ends rigidly fixed, or a continuous beam, with a load in middle.	Twice the coefficient of col. II.	Four times the c[illegible]cient of col. IV.
Both ends rigidly fixed, or a continuous beam, with load evenly distributed.	One and one-half times the coefficient of col. III.	Five times the coeff[illegible] of col. V.

ELASTIC RESILIENCE.

In a rectangular beam tested by transverse stress, supported at th[illegible] and loaded in the middle,

$$P = \frac{2}{3}\frac{Rbd^2}{l};$$

$$\Delta = \frac{1}{4}\frac{Pl^3}{Ebd^3};$$

in which, if P is the load in pounds at the elastic limit, R = the modul[illegible] transverse strength, or the strain on the extreme fibre, at the elastic [illegible] E = modulus of elasticity, Δ = deflection, l, b, and d = length, breadth[illegible] depth in inches. Substituting for P in (2) its value in (1), we have

$$\Delta = \frac{1}{6}\frac{Rl^2}{Ed}.$$

silience = half the product of the load and deflection = $\frac{1}{2}P\Delta$,
: resilience per cubic inch

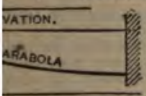
$$= \frac{1}{2}\frac{P\Delta}{lbd}.$$

the values of P and Δ, this reduces to elastic resilience per
$\frac{1}{18}\frac{R^2}{E}$, which is independent of the dimensions; and therefore
silience per cubic inch for transverse strain may be used as a
essing one valuable quality of a material.
r tension:

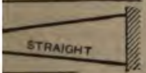
sile stress in pounds per square inch at the elastic limit;
ngation per unit of length at the elastic limit;
dulus of elasticity = $P \div e$; whence $e = P \div E$.
: resilience per cubic inch = $\frac{1}{2}Pe = \frac{1}{2}\frac{P^2}{E}$.

OF UNIFORM STRENGTH THROUGHOUT THEIR LENGTH.

n is supposed in all cases to be rectangular throughout. The
n in plan are of uniform depth throughout. Those shown in
of uniform breadth throughout.
B = breadth of beam. D = depth of beam.



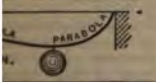
Fixed at one end, loaded at the other; curve parabola, vertex at loaded end; BD^2 proportional to distance from loaded end. The beam may be reversed, so that the upper edge is parabolic, or both edges may be parabolic.



Fixed at one end, loaded at the other; triangle, apex at loaded end; BD^2 proportional to the distance from the loaded end.

Fixed at one end; load distributed; triangle, apex at unsupported end; BD^2 proportional to square of distance from unsupported end.

Fixed at one end; load distributed; curves two parabolas, vertices touching each other at unsupported end; BD^2 proportional to distance from unsupported end.



Supported at both ends; load at any one point; two parabolas, vertices at the points of support, bases at point loaded; BD^2 proportional to distance from nearest point of support. The upper edge or both edges may also be parabolic.

Supported at both ends; load at any one point; two triangles, apices at points of support, bases at point loaded; BD^2 proportional to distance from the nearest point of support.

Supported at both ends; load distributed; curves two parabolas, vertices at the middle of the beam; bases centre line of beam; BD^2 proportional to product of distances from points of support.

Supported at both ends; load distributed; curve semi-ellipse; BD^2 proportional to the product of the distances from the points of support.

PROPERTIES OF ROLLED STRUCTURAL SH

Explanation of Tables of the Properties of Car I Beams, Channels, and Z Bars.

The tables of I beams are calculated for the minimum weight each pattern can be rolled. The tables of channels are calculat minimum and maximum weights of the various shapes, while the p of Z bars are given for thicknesses differing by 1/16 inch.

Columns 11 and 13, in the tables for I beams and channels, give co by the help of which the safe uniformly-distributed load may read termined. To do this, divide the coefficient given by the span or between supports in feet. If the weight of the section is interme tween the minimum and maximum weights given, add to the c for the minimum weight the value given in columns 12 or 14 (for o increase of weight), multiplied by the number of pounds the heavier than the minimum.

If a section is to be selected (as will usually be the case) in carry a certain load, for a length of span already determined on, the coefficient which this load and span will require, and refer to for a section having a coefficient of this value. The coefficient is by multiplying the load, in pounds uniformly distributed, by length in feet.

In case the load is not uniformly distributed, but is concentrat middle of the span, multiply the load by 2 and then consider formly distributed. The deflection will be 8/10 of the deflection latter load.

For other cases of loading obtain the bending moment in foot-po multiplied by 8 will give the coefficient required.

If the loads are quiescent, the coefficients for a fibre strain of per square inch for steel and 12,000 lbs. for iron may be use moving loads are to be provided for, the coefficients for 12,500 a lbs., respectively, should be taken. Inasmuch as the effects of im be very considerable (the strains produced in an unyielding, material by a load suddenly applied being double those produce same load in a quiescent state), it will sometimes be advisable to smaller fibre strains than those given in the tables. In such cas efficients can readily be determined by proportion. Thus, for a fi of 8000 lbs. per square inch the coefficient will equal the coeffi 10,000 lbs. fibre strain, from the table, multiplied by 8/10.

The moments of resistance given in column 9 are used to deter fibre strain per square inch in a beam, or other shape, subjected to or transverse strains, by dividing the same into the bending expressed in inch-pounds.

For Carnegie Z bars, complete tables of moments of inertia, mo resistance, radii of gyration, and values of the coefficients (C) are thicknesses varying by 1/16 inch. These coefficients may be ap explained above, for cases where the Z bars are subjected to t loading, as, for example, in the case of roof-purlins.

For more complete and detailed information concerning structur consult the pocket-books and circulars issued by the manufacturer

[A more correct term for what is called "moment of resistance and also in the tables on pages 274-277, is "moment of resisting are Johnson, *Eng'g News*, Feb. 27, 1896. Reuleaux calls it "section or section factor.]

STEEL.

Proper Distance in Feet, Centre to Centre of Beams.

Distance between Supports in feet.	20" I. 80 lbs.	20" I. 64 lbs.	15" I. 80 lbs.	15" I. 60 lbs.	15" I. 50 lbs.	15" I. 41 lbs.	12" I. 40 lbs.	12" I. 32 lbs.	10" I. 33 lbs.	10" I. 25½ lbs.	9" I. 27 lbs.	9" I. 21 lbs.	Distance between Supports in feet.	8" I. 22 lbs.	8" I. 18 lbs.	7" I. 20 lbs.	7" I. 15½ lbs.	6" I. 16 lbs.	6" I. 13 lbs.	5" I. 13 lbs.	5" I. 10 lbs.	4" I. 10 lbs.	4" I. 7½ lbs.
12	107.3	84.9	77.6	63.6	52.3	41.9	34.7	27.4	23.9	18.3	18.2	13.9	5	76.6	61.6	60.6	47.0	40.7	33.4	26.8	21.2	16.5	12.6
13	91.5	72.3	66.1	54.2	44.6	35.7	29.6	23.4	20.4	15.6	15.5	11.8	6	53.2	42.8	42.1	32.7	28.3	23.2	18.6	14.7	11.4	8.7
14	78.8	62.4	57.0	46.7	38.4	30.8	25.5	20.2	17.6	13.5	13.4	10.2	7	39.1	31.4	30.9	24.0	20.8	17.0	13.7	10.8	8.4	6.4
15	68.7	54.3	50.0	40.7	33.5	26.8	22.2	17.6	15.3	11.7	11.7	8.9	8	29.9	24.1	23.7	18.4	15.9	13.0	10.5	8.3	6.4	4.9
16	60.4	47.7	43.7	35.8	29.4	23.6	19.5	15.4	13.4	10.3	10.2	7.8	9	23.7	19.0	18.7	14.5	12.6	10.3	8.3	6.5	5.1	3.9
17	53.5	42.3	38.7	31.7	26.1	20.9	17.3	13.7	11.9	9.1	9.1	6.9	10	19.2	15.4	15.1	11.8	10.2	8.4	6.7	5.3	4.1	3.1
18	47.7	37.7	34.5	28.3	23.3	18.6	15.4	12.2	10.6	8.1	8.1	6.2	11	15.8	12.7	12.5	9.7	8.4	6.9	5.5	4.4	3.4	2.6
19	42.8	33.9	31.0	25.4	20.9	16.7	13.9	10.9	9.5	7.3	7.3	5.5	12	13.3	10.7	10.5	8.2	7.1	5.8	4.7	3.7	2.9	2.2
20	38.6	30.6	28.0	22.9	18.8	15.1	12.5	9.9	8.6	6.6	6.6	5.0	13	11.3	9.1	9.0	7.0	6.0	4.9	4.0	3.1	2.4	1.9
21	35.0	27.7	25.3	20.8	17.1	13.7	11.3	8.9	7.8	6.0	5.9	4.5	14	9.8	7.9	7.7	6.0	5.2	4.3	3.4	2.7	2.1	1.6
22	31.9	25.3	23.1	18.9	15.6	12.5	10.3	8.2	7.1	5.5	5.4	4.1	15	8.5	6.8	6.7	5.2	4.5	3.7	3.0	2.3	1.8	1.4
23	29.2	23.1	21.1	17.3	14.2	11.4	9.5	7.5	6.5	5.0	5.0	3.8	16	7.5	6.0	5.9	4.6	4.0	3.3	2.6	2.1	1.6	1.2
24	26.8	21.2	19.4	15.9	13.1	10.5	8.7	6.9	6.0	4.6	4.6	3.5	17	6.6	5.3	5.2	4.1	3.5	2.9	2.3	1.8	1.4	1.1
25	24.7	19.6	17.9	14.7	12.1	9.6	8.0	6.3	5.5	4.2	4.2	3.2	18	5.9	4.8	4.7	3.6	3.1	2.6	2.1	1.6	1.3	1.0
26	22.9	18.1	16.5	13.6	11.1	8.9	7.4	5.8	5.1	3.9	3.9	3.0	19	5.3	4.3	4.2	3.3	2.8	2.3	1.9	1.5	1.1	....
27	21.2	16.8	15.3	12.6	10.3	8.3	6.9	5.4	4.7	3.6	3.6	2.7	20	4.8	3.9	3.8	2.9	2.5	2.1	1.7	1.3	1.0	....
28	19.7	15.6	14.3	11.7	9.6	7.7	6.4	5.0	4.4	3.4	3.3	2.6	21	4.3	3.5	3.4	2.7	2.3	1.9	1.5	1.2	....	....
29	18.4	14.5	13.3	10.9	9.0	7.2	5.9	4.7	4.1	3.1	3.1	2.4	22	4.0	3.2	3.1	2.4	2.1	1.7	1.4	1.1	....	....
30	17.2	13.6	12.4	10.2	8.4	6.7	5.6	4.4	3.8	2.9	2.9	2.2											

For any other load than 100 lbs. per square foot, divide the spacing given by the ratio the given load per square foot bears to 100. … for a load of 150 lbs. per square foot divide by 1.5. Maximum fibre strain, 16,000 lbs. per square inch.

Only figures above the cross lines should be used for plastered ceilings, so that the deflection will not cause cracking of the plaster.

Properties of Carnegie I Beams—Steel.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Section Index.	Depth of Beam.	Weight per foot.	Area of Section.	Thickness of Web.	Width of Flange.	Increase of Thickness of Web for each lb. Increase of Weight.	Moment of Inertia, Neutral Axis Perpendicular to Web at Centre.	Moment of Resistance, Neutral Axis as before.	Radius of Gyration, Neutral Axis as before.	Coefficient of Strength for Fibre Strain of 16,000 lbs. per sq. inch. Used for Buildings.	Add to Coeff. for every lb. Increase in Wt. of Beam.	Coefficient of Strength for Fibre Strain 12,500 lbs. per sq. inch. Used for Bridges.	Add to Coeff. for every lb. Increase in Wt. of Beam	Mom. of Inertia Neutral Axis, coincident with Cent Line of Web.	Radius of Gyration, Neut. Axis as before.
		lbs.	sq. in.	in.	in.	inch.	I	R	r	C		C'		I'	r'
B 1	24″	80	23.5	.50	6.95	.0123	2059.3	171.6	9.42	1830500	12800	1430100	10000	41.6	1.34
B 2	20″	80	23.5	.60	7.00	.015	1449.2	144.9	7.85	1545600	10450	1207500	8200	45.6	1.39
B 3	20″	64	18.8	.50	6.25		1146.0	114.6	7.80	1222400		955000		27.3	1.20
B 4	15″	80	23.5	.77	6.41	.020	785.9	104.8	5.82	1117700	7800	873200	6100	42.2	1.35
B 5	15″	60	17.6	.54	6.04		644.0	85.9	6.04	916300		715800		30.4	1.32
B 6	15″	50	14.7	.45	5.75	.020	529.7	70.6	6.00	753300	7800	588500	6100	21.0	1.20
B 7	15″	41	12.0	.40	5.50		424.1	56.6	5.94	603200		471300		14.0	1.08
B 8	12″	40	11.7	.39	5.50	.025	281.3	46.9	4.90	500100	6300	390700	4900	16.8	1.20
B 9	12″	32	9.4	.35	5.25		222.3	37.0	4.85	395200		308800		10.3	1.04
B10	10″	33	9.7	.37	5.00	.029	161.3	32.3	4.08	344000	5200	268800	4100	11.8	1.10
B11	10″	25.5	7.5	.32	4.75		123.7	24.7	4.06	263800		206100		7.32	0.99
B12	9″	27	7.9	.31	4.75	.033	110.6	24.6	3.72	262200	4600	204900	3600	9.10	1.07
B13	9″	21	6.2	.27	4.50		84.3	18.7	3.70	199900		156100		5.56	0.95
B14	8″	22	6.5	.27	4.50	.037	71.9	18.0	3.33	191600	4200	149700	3300	6.62	1.01
B15	8″	18	5.3	.25	4.25		57.8	14.4	3.30	154000		120300		4.35	0.91
B16	7″	20	5.9	.27	4.25	.042	49.7	14.2	2.91	151400	3600	118300	2800	5.52	0.97
B17	7″	15.5	4.6	.23	4.00		38.6	11.0	2.91	117600		91900		3.47	0.87
B18	6″	16	4.7	.26	3.63	.049	28.6	9.54	2.47	101800	3100	79500	2400	3.24	0.83
B19	6″	13	3.8	.23	3.50		23.5	7.83	2.48	83500		65300		2.27	0.77
B20	5″	13	3.8	.26	3.13	.059	15.7	6.28	2.03	67000	2600	52400	2000	1.99	0.72
B21	5″	10	3.0	.22	3.00		12.4	4.96	2.05	52900		41300		1.29	0.66
B22	4″	10	2.9	.24	2.75	.074	7.7	3.86	1.62	41200	2100	32200	1600	1.22	0.65
B23	4″	7.5	2.2	.20	2.63		5.9	2.95	1.63	31400		24600		0.75	0.58
B24	4″	6	1.8	.18	2.18		4.6	2.30	1.61	24400		19060		0.36	0.45

Section Ind	Depth of Ch	Weight per lbs.	Area of Sect sq. in.	Thickness o inches.	Width of Fl inches.	Increase of ness of W each lb. In of Weight. inches.	Moment of I Neutral Axi pendicular t at Centre. I	Moment of F ance, Neut Axis as be R	Radius of Gy Neutral A before. r	Coefficient o Strength f Fibre Strai 16,000 lbs. square inc for Buildin C	Add to Coeff for every crease in W of Channel	for Fibre Strain of 12,500 lbs. per square inch, used for Bridges C'	Add to Coeff for every crease in W of Channel	Distance of C of Gravity Outside of inches.
C 1	15″	51	15.0	.78	3.78		390.0	52.0	5.12	554700		433400		0.77
C 1	15″	32	9.4	.40	3.40	.020	284.5	37.9	5.53	404700	7800	316200	6100	0.75
C 2	12″	30¼	8.9	.55	3.15		153.9	25.7	4.17	273600		213800		0.62
C 2	12″	20	5.9	.30	2.90	.025	117.9	19.7	4.49	209600	6300	163800	4900	0.62
C 3	10″	23¾	7.0	.51	2.91		84.6	16.9	3.50	180500		141000		0.61
C 3	10″	15¼	4.5	.26	2.66	.029	63.8	12.8	3.80	136100	5200	106300	4100	0.63
C 4	9″	20½	6.0	.49	2.69		58.5	13.0	3.14	138700		108400		0.56
C 4	9″	12¾	3.7	.24	2.44	.033	43.3	9.6	3.42	102700	4600	80200	3600	0.58
C 5	8″	17¼	5.0	.47	2.47		38.9	9.7	2.78	103700		81000		0.52
C 5	8″	10	2.9	.20	2.20	.037	27.4	6.8	3.07	73100	4200	57100	3300	0.54
C 6	7″	14½	4.3	.45	2.25		24.6	7.0	2.42	75000		58600		0.48
C 6	7″	8½	2.5	.20	2.00	.042	17.4	5.0	2.67	53100	3600	41500	2800	0.49
C 7	6″	12	3.6	.44	2.14		15.6	5.2	2.09	55400		43300		0.47
C 7	6″	7	2.1	.19	1.89	.049	11.1	3.7	2.31	39400	3100	30800	2400	0.48
C 8	5″	10¼	3.0	.43	2.03		9.1	3.7	1.75	39000		30500		0.47
C 8	5″	6	1.7	.18	1.78	.059	6.5	2.6	1.94	27900	2600	21800	2000	0.48
C 9	4″	8¼	2.4	.41	1.91		4.8	2.4	1.41	25700		20100		0.48
C 9	4″	5	1.4	.17	1.67	.074	3.5	1.8	1.57	18700	2100	14600	1600	0.48

L = Safe load in lbs. uniformly distributed; l = span in feet.
M = Moment of forces in foot-lbs.; C and C' = coefficients given above.

$L = \frac{C \text{ or } C'}{l}$; $M = \frac{C \text{ or } C'}{8}$; C or $C' = Ll = 8M$.

PROPERTIES OF CARNEGIE Z BARS.

IRON OR STEEL.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
							Moments of Inertia. I		Moments of Resistance. R		Radii of Gyration. r			Coefficient of Strength. C	C'
Section Index.	Depth of Web. in.	Width of Flange. in.	Thickness of Metal. in.	Weight per foot, Iron. lbs.	Weight per foot, Steel. lbs.	Area of Section. sq. in.	Neutral Axis through Centre of Gravity Perpendicular to Web.	Neutral Axis through Centre of Gravity Coincident with Web.	Neutral Axis through Centre of Gravity Perpendicular to Web.	Neutral Axis through Centre of Gravity Coincident with Web.	Neutral Axis through Centre of Gravity Perpendicular to Web.	Neutral Axis through Centre of Gravity Coincident with Web.	Least Radius, Neutral Axis Diagonal.	For Fibre Strain of 16,000 lbs. per square inch. Axis Perpendicular to Web at Centre.	For Fibre Strain of 12,000 lbs. per square inch. Axis Perpendicular to Web at Centre.
Z 1	6	3½	⅜	15.3	15.6	4.59	25.32	9.11	8.44	2.75	2.35	1.41	0.83	90000	67500
Z 1	6 1-16	3 9-16	7-16	18.0	18.3	5.39	29.80	10.95	9.83	3.27	2.35	1.43	0.84	104800	78600
Z 1	6⅛	3⅝	½	20.6	21.0	6.19	34.36	12.87	11.22	3.81	2.36	1.44	0.84	119700	89800
Z 2	6	3½	9-16	22.3	22.7	6.68	34.64	12.59	11.55	3.91	2.28	1.37	0.81	123200	92400
Z 2	6 1-16	3 9-16	⅝	24.9	25.4	7.46	38.86	14.42	12.82	4.43	2.28	1.39	0.82	136700	102600
Z 2	6⅛	3⅝	11-16	27.5	28.0	8.25	43.18	16.34	14.10	4.98	2.29	1.41	0.84	150400	112800
Z 3	6	3½	¾	28.8	29.3	8.63	42.12	15.44	14.04	4.94	2.21	1.34	0.81	149800	112300
Z 3	6 1-16	3 9-16	13-16	31.3	32.0	9.40	46.13	17.27	15.22	5.47	2.22	1.36	0.82	162300	121800
Z 3	6⅛	3⅝	⅞	33.9	34.6	10.17	50.22	19.18	16.40	6.02	2.22	1.37	0.83	174900	131200
Z 4	5	3¼	5-16	11.3	11.6	3.40	13.36	6.18	5.34	2.00	1.98	1.35	0.75	57000	42700
Z 4	5 1-16	3 5-16	⅜	13.7	13.9	4.10	16.18	7.65	6.39	2.45	1.99	1.37	0.76	68200	51100
Z 4	5⅛	3⅜	7-16	16.0	16.4	4.81	19.07	9.20	7.44	2.92	1.99	1.38	0.77	79400	59500
Z 5	5	3¼	½	17.5	17.8	5.25	19.19	9.05	7.68	3.02	1.91	1.31	0.74	81900	61400
Z 5	5 1-16	3 5-16	9-16	19.8	20.2	5.94	21.83	10.51	8.62	3.47	1.91	1.33	0.75	91900	69000
Z 5	5⅛	3⅜	⅝	22.1	22.6	6.64	24.53	12.06	9.57	3.94	1.92	1.35	0.76	102100	76600
Z 6	5	3¼	11-16	23.2	23.7	6.96	23.68	11.37	9.47	3.91	1.84	1.28	0.73	101000	75800
Z 6	5 1-16	3 5-16	¾	25.5	26.0	7.64	26.16	12.88	10.34	4.37	1.85	1.30	0.75	110300	82700

Properties of Carnegie Z Bars—(Continued.)

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
							Moments of Inertia. I		Moments of Resistance. R		Radii of Gyration. r			Coefficient of Strength. C	C'
Section Index.	Depth of Web. in.	Width of Flange. in.	Thickness of Metal. in.	Weight per foot, Iron. lb.	Weight per foot, Steel. lbs.	Area of Section. sq. in.	Neutral Axis through Centre of Gravity Perpendicular to Web.	Neutral Axis through Centre of Gravity Coincident with Web.	Neutral Axis through Centre of Gravity Perpendicular to Web.	Neutral Axis through Centre of Gravity Coincident with Web.	Neutral Axis through Centre of Gravity Perpendicular to Web.	Neutral Axis through Centre of Gravity Coincident with Web.	Least Radius, Neutral Axis Diagonal.	For Fibre Strain of 16,000 lbs. per square inch, Axis Perpendicular to Web at Centre.	For Fibre Strain of 12,000 lbs. per square inch, Axis Perpendicular to Web at Centre.
Z 6	5⅛	3⅜	13-16	27.8	28.3	8.33	29.31	14.86	11.44	4.84	1.88	1.31	0.76	122000	91500
Z 7	4	3 1-16	¼	8.0	8.2	2.41	6.28	4.23	3.14	1.44	1.62	1.33	0.67	33500	25100
Z 7	4 1-16	3⅛	5-16	10.1	10.3	3.03	7.94	5.46	3.91	1.84	1.62	1.34	0.68	41700	31300
Z 7	4⅛	3 3-16	⅜	12.2	12.4	3.66	9.63	6.77	4.67	2.26	1.62	1.36	0.69	49800	37400
Z 8	4	3 1-16	7-16	13.5	13.8	4.05	9.66	6.73	4.83	2.37	1.55	1.29	0.66	51500	38600
Z 8	4 1-16	3⅛	½	15.5	15.8	4.66	11.18	7.96	5.50	2.77	1.55	1.31	0.67	58700	44000
Z 8	4⅛	3 3-16	9-16	17.6	17.9	5.27	12.74	9.26	6.18	3.19	1.55	1.33	0.69	65900	49400
Z 9	4	3 1-16	⅝	18.5	18.9	5.55	12.11	8.73	6.05	3.18	1.48	1.25	0.66	64500	48400
Z 9	4 1-16	3⅛	11-16	20.5	20.9	6.14	13.52	9.95	6.65	3.58	1.48	1.27	0.67	70900	53200
Z 9	4⅛	3 3-16	¾	22.5	22.9	6.75	14.97	11.24	7.26	4.00	1.49	1.29	0.69	77400	58100
Z 10	3	2 11-16	¼	6.6	6.7	1.97	2.87	2.81	1.92	1.10	1.21	1.19	0.55	20500	15400
Z 10	3 1-16	2¾	5-16	8.3	8.4	2.48	3.64	3.64	2.38	1.40	1.21	1.21	0.56	25400	19000
Z 11	3	2 11-16	⅜	9.5	9.7	2.86	3.85	3.92	2.57	1.57	1.16	1.17	0.55	27400	20600
Z 11	3 1-16	2¾	7-16	11.2	11.4	3.36	4.57	4.75	2.98	1.88	1.17	1.19	0.56	31800	23800
Z 12	3	2 11-16	½	12.3	12.5	3.69	4.59	4.85	3.06	1.99	1.12	1.15	0.55	32600	24500
Z 12	3 1-16	2¾	9-16	13.9	14.2	4.18	5.26	5.70	3.43	2.31	1.12	1.17	0.56	36600	27400

TRENTON IRON BEAMS AND CHANNELS

(New Jersey Steel and Iron Co.)

I Beams.

Height in inches.	Least Weight per Yard, in pounds.	Width of Flange, in inches.	Thickness of Web, in inches.	Coefficient in lbs. for Transverse Strength.
20	272	6¾	11-16	1,320,000
20	200	6	½	990,000
15⅛	200	5¾	.6	748,000
15 3-16	150	5	½	551,000
15⅛	125	5	.42	460,000
12 5-16	170	5½	.6	511,000
12¼	125	4.8	.47	377,000
12	120	5½	.39	375,000
12	96	5¼	.32	306,000
10½	135	5	.47	360,000
10½	105	4½	⅜	286,000
10½	90	4½	5-16	250,000
9	125	4½	.57	268,000
9	85	4½	⅜	199,000
9	70	4	.3	167,000
8	80	4½	⅜	168,000
8	65	4	.3	135,000
7	55	3¾	.3	101,000
6	120	5¼	⅝	172,000
6	90	5	½	132,000
6	50	3½	.3	76,800
6	40	3	¼	62,600
5	40	3	5-16	49,100
5	30	2¾	¼	38,700
4	37	3	5-16	36,800
4	30	2¾	¼	30,100
4	18	2	3-16	18,000

Height in inches.	Least Weight per Yard, in pounds.	Width of Flange, in inches.	Thickness of Web, in inches.
Channels.			
15	190	4¾	¾
15	120	4	½
12¼	140	4	11-16
12¼	70	3	.33
10½	60	2¾	⅝
10	48	2½	5-16
9	70	3½	7-16
9	50	2½	.33
8	45	2½	.26
8	33	2.2	.20
7	36	2½	¼
7	25½	2	.20
6	45	2½	.40
6	33	2¼	.28
6	22½	1⅞	.18
5	19	1⅝	.20
4	16½	1½	.20
3	15	1½	.20
Deck Beams.			
8	65	4½	⅝
7	55	4½	5-16

Trenton Beams and Channels.

To find which beam, supported at both ends, will be required to
with safety a given uniformly distributed load:

Multiply the load in pounds by the span in feet, and take the bea
" Coefficient for Strength " is nearest to and exceeds the number s
The weight of the beam itself should be included in the load.

The deflection in inches, for such distributed load, will be found
ing the square of the span taken in feet, by 70 times the depth of t
taken in inches, for iron beams, and by 52.5 times the depth for ste

EXAMPLE.—Which beam will be required to support a uniforml
uted load of 12 tons (= 24,000 lbs.) on a span of 15 feet ?

24,000 × 15 = 360,000, which is less than the coefficient of the 12¼-
lb. iron beam. The weight of the beam itself would be 625 lbs., w
ded to the load and multiplied by the span, would still give a pro
than the coefficient; thus,

$$24{,}625 \times 15 = 369{,}375.$$

The deflection will be

$$\frac{15 \times 15}{70 \times 12\frac{1}{4}} = 0.26 \text{ inch.}$$

or each beam can be found by dividing
nd subtracting the weight of the beam

When the load is concentrated entirely at the centre of the span, one ha of this amount must be taken.

The beams must be secured against yielding sideways, or the safe loa will be much less.

For beams used with plastered ceilings, the deflection allowed should n exceed 1/30 inch per foot of span, to avoid cracking of the plaster.

TRENTON ANGLE-BARS.

Size of Bar.	Approximate Weight, in pounds per yard, for each thickness in inches.								Coeff. fo Transvers Strength
	7/16	½	9/16	⅝	11/16	¾	13/16	⅞	Thinnest B
6 × 6	......	57.5	64.3	71.1	77.8	84.4	91.0	97.3	36,900 lbs
4½ × 4½	37.5	42.5	47.5	52.3	57.2	61.9	......	......	18,000 "
	⅜	7 16	½	9/16	⅝	11/16	¾		
4 × 4	28.6	33 1	37.5	41.8	46.1	50.5	54.4	......	12,184 "
3½ × 3½	24.8	28.7	32.5	36.2	39.8	43.4	......	......	9,200 "
	¼	5/16	⅜	7/16	½	9/16	⅝	11/16	
3 × 3	14.4	17.7	21.1	24 4	27.5	30.6	33.6	36.5	4,611 "
	5/16	⅜	13/32	7/16	15/32	½	17/32	9/16	
2¾ × 2¾	16.2	19.2	20.7	22.2	23.6	25.0	26.3	27.7	4,710 "
	¼	5/16	11/32	⅜	13/32	7/16	15/32	½	
2½ × 2½	11.9	14.7	16.0	17.3	18.6	20.0	21.2	22.5	3,156 "
	¼	9/32	5/16	11/32	⅜	13/32	7/16		
2¼ × 2¼	10.6	11.9	13.1	14.3	15.5	16.8	17.8	......	2,530 "
	7/32	¼	9/32	5/16	11/32	⅜			
2 × 2	8.3	9.4	10.4	11.5	12.6	13 6	......	......	1,752 "
	3/16	7/32	¼	9/32	5/16	11/ 2	⅜		
1¾ × 1¾	6.21	7.18	8.13	9 05	9.96	10 8	11.7	......	1,150 "
1½ × 1½	5.27	6.09	6.88	7.64	8.40	9.13	......	......	832 "
	⅛	5/32	3/16	7/32	¼				
1¼ × 1¼	2.97	3.66	4 34	4.99	5.63	......	......	......	393 "
1 × 1	2.34	2.88	3.40	3.91	4.38	......	......	......	246 "
⅞ × ⅞	2.03	2.48	2.93	......	......	......	......	......	186 "
¾ × ¾	1.72	2.09	2.46	......	......	......	......	......	133 "

Uneven Legs.

			7/16	½	9/16	⅝	11/16	¾	
6 × 4	......	......	41.8	47.5	53.1	58.6	64.0	69.4	30,680, 6″ wa 14,750, 4″
		⅜							
5 × 3½	....	30 5	35.3	40.0	44.7	19.2	53.7	58.1	18,353, 5″ 9,651, 3½″
4½ × 3	......	26.7	30.9	35.0	39.0	43.0	46.8	50.6	14,580, 4½″ 7,020, 3″
		5/16	⅜	7/16	½	9/16	⅝	11/16	
4 × 3	......	20.9	24.8	28.7	32.5	36.2	39.8	43.4	9,850, 4″ 5,871, 3″
	¼								
3½ × 3	15.6	19.3	23.9	26.5	30.0	33.4	36.7	40 0	6,180, 3½″ 4,710, 3″
3½ × 2½	14.4	17.7	21.1	24.4	27.5	30.6	33.6	36.5	6,037, 3½″ 3,296, 2½″
		¼							
3½ × 1½	......	11.9	......	......	......	......	......	......	5,515, 3½″ 1,148, 1½″
	¼	5/16	11/32	⅜	13/32	7/16	½	9/16	
3 × 2½	13.1	16.2	17.7	19.2	20.7	22.2	25.0	27.7	4,490, 3″ 3,233, 2½″
		7/32	¼	9/32	5/16	⅜	7/16	½	
3 × 2	......	10.4	11.9	13.3	14.6	17.3	20.0	22.5	3,833, 3″ 1,850, 2″

TRENTON TEE BARS.

Designation of Bar. Table.	Leg.	Approximate Weight, in pounds per yard, for each thickness in inches.		Coefficient for Transverse Strength. Thinnest Bar
4"	× 4"	....	½" 37.5 lbs.	15,800 lbs.
3½"	× 3½"	7-16" 28.7 lbs.	½" 32.5 "	10,550 "
3"	× 3"	⅜" 21.1 "	½" 27.5 "	6,680 "
2½"	× 2½"	5-16" 14.7 "	⅜" 17.3 "	3,850 "
2¼"	× 2¼"	5-16" 13 09 "	....	3,087 "
2"	× 2"	¼" 9.4 "	5-16" 11.5 "	1,970 "
1½"	× 1½"	¼" 6.68 "	....	1,033 "
1¼"	× 1¼"	7-32" 4.87 "	¼" 5.5 "	596 "
1"	× 1"	5-32" 2.80 "	3-16" 3.3 "	268 "
5"	× 2½"	....	½" 35.0 "	6,344 "
3"	× 2"	5-16" 14.6 lbs.	⅜" 17.3 "	2,540 "
2½"	× 3"	⅜" 19.2 "	....	6,404 "
2"	× 3"	⅜" 17.3 "	....	6,173 "
2"	× 1½"	9-32" 9.1 "	....	1,355 "
2¼"	× 1⅛"	¼" 7.4 "	....	604 "
2"	× 1"	¼" 6.5 "	....	457 "
1½"	× 1"	¼" 5.6 "	....	421 "

SIZE OF BEAMS, AND THEIR DISTANCE APART, Suitable for Floors having Loads per square foot from 100 lbs. to 300 lbs.

(New Jersey Steel and Iron Co.)

Clear Span in feet.	Load per sq. ft. 100 lbs.			Load per sq. ft. 150 lbs.			Load per sq. ft. 200 lbs.			Load per sq. ft. 250 lbs.			Load per sq. ft. 300 lbs.		
	Size and Weight per yard.		Distance from Centre to Centre.	Size and Weight per yard.		Distance from Centre to Centre.	Size and Weight per yard.		Distance from Centre to Centre.	Size and Weight per yard.		Dist. from Centre to Centre.	Size and Weight per yard.		Dist. from Centre to Centre.
	in.	lb.	feet	in.	lb.	feet	in.	lb.	feet	in.	lb.	feet	in.	lb.	feet
8	4	30	4.6	4	30	3.1	5	30	3.0	6	40	3.9	6	40	3.2
	5	30	5.9	5	30	4.0	6	40	4.8	6	50	4.7	5	50	3.9
10	5	30	3.8	6	40	4.1	6	40	3.0	6	50	3.0	7	55	3.3
	5	40	4.8	6	50	5.0	6	50	3.7	7	55	4.0	8	65	4.4
12	6	40	4.2	6	50	3.4	7	55	3.4	8	65	3.6	8	65	3.0
	6	50	5.2	7	55	4.6	8	65	4.5	9	70	4.5	9	70	3.8
14	7	55	5.0	7	55	3.3	8	65	3.3	9	70	3.3	9	85	3.3
	8	65	6.7	8	65	4.5	9	70	4.1	10½	90	5.0	10½	90	4.2
16	8	65	5.0	8	65	3.3	9	85	3.7	10½	90	3.8	10½	105	3.6
	9	70	6.3	9	70	4.2	10½	90	4.7	10½	105	4.3	12¼	125	4.8
18	9	70	4.9	9	85	3.9	10½	105	4.2	10½	105	3.4	10½	135	3.6
	9	85	5.9	10½	90	4.9	12	96	4.6	12¼	125	4.5	12¼	125	3.7
20	10½	90	6.0	10½	105	4.5	10½	105	3.4	12¼	125	3.6	12¼	125	3.0
	.....	.....	.....	12¼	125	6.0	12¼	125	4.5	12¼	170	4.9	15	150	4.4
22	10½	90	4.9	12	96	4.0	12¼	125	3.7	12¼	125	3.0	12¼	170	3.3
	10½	105	5.6	12¼	125	4.9	15	125	4.5	15	125	3.6	15	150	3.6
24	12	96	5.0	12¼	125	4.1	12¼	125	3.0	12¼	170	3.3	15	150	3.0
	12¼	125	6.1	15	125	5.0	15	150	4.5	15	150	3.6	15	200	4.1
26	12¼	125	5.1	15	125	4.3	15	150	3.8	15	150	3.0	15	200	3.3
	.....	.....	.....	15	150	5.1	15	200	5.2	15	200	4.2	20	200	4.7
28	15	125	5.5	15	150	4.3	15	200	4.4	15	200	3.5	20	200	3.9
	.....	.....	.....	15	200	5.9	20	200	6.0	20	200	4.8	20	272	5.3
30	15	150	5.6	15	150	3.7	15	200	3.8	20	200	4.1	20	200	3.4
	.....	.....	.....	15	200	5.1	20	200	5.2	20	272	5.5	20	272	[illegible]

FLOORING MATERIAL.

re-proof flooring, the space between the floor-beams may be spanned k arches, or with hollow brick made especially for the purpose, the ing much lighter than ordinary brick.

s 4 inches deep of solid brick weigh about 70 lbs. per square foot, g the concrete levelling material, and substantial floors are thus to 6 feet span of arch, or much greater span if the skew backs at ging of the arch are made deeper, the rise of the arch being prefer- less than 1/10 of the span. Hollow brick for floors are usually in out ½ of the span, and are used up to, and even exceeding, spans et. The weight of the latter material will vary from 20 lbs. per oot for 3-foot spans up to 60 lbs. per square foot for spans of 10 feet. ticulars of this construction are given by the manufacturers. For ng brick floors the beams should be securely tied with rods to resist al pressure.

following cases the loads, in addition to the weight of the floor ay be assumed as:

street bridges for general public traffic........	80 lbs. per sq. ft.
floors of dwellings	40 lbs. " "
churches, theatres, and ball-rooms...........	80 lbs. " "
hay-lofts	80 lbs. " "
storage of grain	100 lbs. " "
warehouses and general merchandise..........	250 lbs. " "
factories..................................	200 to 400 lbs. " "
snow thirty inches deep......................	16 lbs. " "
maximum pressure of wind	50 lbs. " "
brick walls	112 lbs. per cu. ft.
masonry walls..............................	116–144 lbs. " "

allowing thirty pounds per square foot for wind and snow:

corrugated iron laid directly on the purlins...	37 lbs. per sq. ft.
corrugated iron laid on boards..................	40 lbs. " "
slate nailed to laths	43 lbs. " "
slate nailed on boards.........................	46 lbs. " "

tered below the rafters, the weight will be about ten pounds per ot additional.

E-RODS FOR BEAMS SUPPORTING BRICK ARCHES.

rizontal thrust of brick arches is as follows:

$$\frac{1.5WS^2}{R} = \text{pressure in pounds. per lineal foot of arch:}$$

W = load in pounds. per square foot;
S = span of arch in feet;
R = rise in inches.

he tie-rods as low through the webs of the beams as possible and o that the pressure of arches as obtained above will not produce a tress than 15,000 lbs. per square inch of the least section of the bolt.

TORSIONAL STRENGTH.

horizontal shaft of diameter $= d$ be fixed at one end, and at the free end, at a distance $= l$ from the fixed end, let there be fixed a al lever arm with a weight $= P$ acting at a distance $= a$ from the he shaft so as to twist it; then Pa = moment of the applied force.

ng moment = twisting moment $= \frac{SJ}{c}$, in which S = unit shearing se, J = polar moment of inertia of the section with respect to the d c = distance of the most remote fibre from the axis, in a cross- For a circle with diameter d,

$$J = \frac{\pi d^4}{32}; \qquad c = \tfrac{1}{2}d;$$

$$Pa = \frac{SJ}{c} = \frac{\pi d^3 S}{16} = \frac{d^3 S}{5.1} = .1963 d^3 S; \quad d = \sqrt[3]{\frac{5.1Pa}{S}}.$$

For hollow shafts of external diameter d and internal diameter d_1,

$$Pa = .1963 \frac{d^4 - d_1^4}{d} S; \qquad d = \sqrt[3]{\frac{5.1 Pa}{\left(1 - \frac{d_1^4}{d^4}\right) S}}.$$

For a square whose side $= d$,

$$J = \frac{d^4}{6}; \quad c = d\sqrt{\tfrac{1}{2}}; \quad \frac{SJ}{c} = Pa = \frac{d^3 S}{4.2426} = 0.236 d^3 S.$$

For a rectangle whose sides are b and d,

$$J = \frac{bd^3}{12} + \frac{b^3 d}{12}; \quad c = \tfrac{1}{2}\sqrt{b^2 + d^2}; \quad \frac{SJ}{c} = Pa = \frac{(bd^3 + b^3 d)S}{6\sqrt{b^2 + d^2}}.$$

The above formulæ are based on the supposition that the shearing resistance at any point of the cross-section is proportional to its distance from the axis; but this is true only within the elastic limit. In materials capable of flow, while the particles near the axis are strained within the elastic limit those at some distance within the circumference may be strained nearly to the ultimate resistance, so that the total resistance is something greater than that calculated by the formulæ. (See Thurston, "Matls. of Eng.," Part II. p. 527.) Saint Venant finds for square shafts $Pa = 0.281 d^3 S$ (Rankine, "Mach. and Millwork," p. 504). For working strength, however, the formulæ may be used, with S taken at the safe working unit resistance.

The ultimate torsional shearing resistance S is about the same as the direct shearing resistance, and may be taken at 20,000 to 25,000 lbs. per square inch for cast iron, 45,000 lbs. for wrought iron, and 50,000 to 150,000 lbs. for steel, according to its carbon and temper. Large factors of safety should be taken, especially when the direction of stress is reversed, as in reversing engines, and when the torsional stress is combined with other stresses, as is usual in shafting. (See "Shafting.")

Elastic Resistance to Torsion.—Let l = length of bar being twisted, d = diameter, P = force applied at the extremity of a lever arm of length $= a$, Pa = twisting moment, G = torsional modulus of elasticity, θ = angle through which the free end of the shaft is twisted, measured in arc of radius = 1.

For a cylindrical shaft

$$Pa = \frac{\pi \theta G d^4}{32 l}; \qquad \theta = \frac{32 Pal}{\pi d^4 G}; \qquad G = \frac{32 Pal}{\theta \pi d^4}; \qquad \frac{32}{\pi} = 10.186.$$

If α = angle of torsion in degrees,

$$\theta = \frac{\alpha \pi}{180}; \qquad \alpha = \frac{180\theta}{\pi} = \frac{180 \times 32 Pal}{\pi^2 d^4 G} = \frac{583.6 Pal}{d^4 G}.$$

The value of G is given by different authorities as from ⅓ to 2/5 of E, the modulus of elasticity for tension.

COMBINED STRESSES.

(From Merriman's "Strength of Materials.")

Combined Tension and Flexure.—Let A = the area of a bar subjected to both tension and flexure, P = tensile stress applied at the ends, $P \div A$ = unit tensile stress, S = unit stress at the fibre on the tensile side most remote from the neutral axis, due to flexure alone, then maximum tensile unit stress $= (P \div A) + S$. A beam to resist combined tension and flexure should be designed so that $(P \div A) + S$ shall not exceed the proper allowable working unit stress.

Combined Compression and Flexure.—If $P \div A$ = unit stress due to compression alone, and S = unit compressive stress at fibre most remote from neutral axis, due to flexure alone, then maximum compressive unit stress

$$= \frac{P}{A} + S.$$

Combined Tension (or Compression) and Shear.—If sp-tension (or compression) unit stress $= p$, applied shearing unit stress hen from the combined action of the two forces

Max. $S = \pm \sqrt{v^2 + \frac{1}{4}p^2}$, Maximum shearing unit stress;

$= \frac{1}{2}p + \sqrt{v^2 + \frac{1}{4}p^2}$, Maximum tensile (or compressive) unit stress.

nbined Flexure and Torsion.—If S = greatest unit stress flexure alone, and S_s = greatest torsional shearing unit stress due to alone, then for the combined stresses

lax. tension or compression unit stress $t = \frac{1}{2}S + \sqrt{S_s^2 + \frac{1}{4}S^2}$;

Max. shear $s = \pm \sqrt{S_s^2 + \frac{1}{4}S^2}$.

ula for diameter of a round shaft subjected to transverse load while itting a given horse-power (see also Shafts of Engines):

$$d^3 = \frac{16M}{\pi t} + \frac{16}{t}\sqrt{\frac{M^2}{\pi^2} + \frac{402,500,000H^2}{n^2}},$$

M = maximum bending moment of the transverse forces in pound- H = horse-power transmitted, n = No. of revs. per minute, and t = allowable tensile or compressive working strength of the material.

bined Compression and Torsion.—For a vertical round rrying a load and also transmitting a given horse-power, the result- ximum compressive unit stress

$$t = \frac{4P}{\pi d^2} + \sqrt{321,000^2 \frac{H^2}{\pi^2 d^6} + \frac{16P^2}{\pi^2 d^4}},$$

n P is the load. From this the diameter d may be found when t and r data are given.

s due to Temperature.—Let l = length of a bar, A = its sec- rea, c = coefficient of linear expansion for one degree, t = rise or emperature in degrees, E = modulus of elasticity, λ the change of ue to the rise or fall t; if the bar is free to expand or contract, λ =

bar is held so as to prevent its expansion or contraction the stress d by the change of temperature $= S = ActE$. The following are values of the coefficients of linear expansion for a change in temper- one degree Fahrenheit:

For brick and stone.... $a = 0.0000050$,
For cast iron........... $a = 0.0000062$,
For wrought iron....... $a = 0.0000067$,
For steel................ $a = 0.0000065$.

ress due to temperature should be added to or subtracted from the aused by other external forces according as it acts to increase or to the existing stress.

stress will be caused in a steel bar 1 inch square in area by a change perature of 100° F.? $S = ActE = 1 \times .0000065 \times 100 \times 30,000,000 =$ s. Suppose the bar is under tension of 19,500 lbs. between rigid abut- efore the change in temperature takes place, a cooling of 100° F. ble the tension, and a heating of 100° will reduce the tension to zero.

STRENGTH OF FLAT PLATES.

circular plate supported at the edge, uniformly loaded, according to r,

$$f = \frac{5}{6}\frac{r^2}{t^2}p; \qquad t = \sqrt{\frac{5r^2p}{6f}}; \qquad p = \frac{6ft^2}{5r^2}.$$

circular plate fixed at the edge, uniformly loaded,

$$f = \frac{2}{3}\frac{r^2}{t^2}p; \qquad t = \sqrt{\frac{2}{3}\frac{r^2p}{f}}; \qquad p = \frac{3ft^2}{2r^2};$$

h *f denotes the working stress*; r, the radius in inches; t *inches; and p, the pressure* in pounds per square inch.

For mathematical discussion, see Lanza, "Applied Mechanics," p. 900, Lanza gives the following table, using a factor of safety of 8, with tensile strength of cast iron 20,000, of wrought iron 40,000, and of steel 80,000:

	Supported.	Fixed.
Cast iron	$t = .0182570r\sqrt{p}$	$t = .0163300r\sqrt{p}$
Wrought iron	$t = .0117850r\sqrt{p}$	$t = .0105410r\sqrt{p}$
Steel	$t = .0091287r\sqrt{p}$	$t = .0081649r\sqrt{p}$

For a circular plate supported at the edge, and loaded with a concentrated load P applied at a circumference the radius of which is r_0:

$$f = \left(\frac{4}{3}\log\frac{r}{r_0} + 1\right)\frac{P}{\pi t^2} = c\frac{P}{\pi t^2};$$

for $\frac{r}{r_0} =$ 10 20 30 40 50;

$c =$ 4.07 5.00 5.53 5.92 6.22;

$$t = \sqrt{\frac{cP}{\pi f}}; \qquad P = \frac{\pi t^2 f}{c}.$$

The above formulæ are deduced from theoretical considerations, and give thicknesses much greater than are generally used in steam-engine cylinder heads. (See empirical formulæ under Dimensions of Parts of Engines.) The theoretical formulæ seem to be based on incorrect or incomplete hypotheses, but they err in the direction of safety.

The Strength of Unstayed Flat Surfaces.—Robert Wilson (*Eng'g*, Sept. 24, 1877) draws attention to the apparent discrepancy between the results of theoretical investigations and of actual experiments on the strength of unstayed flat surfaces of boiler-plate, such as the unstayed crowns of domes and of vertical boilers.

Rankine's "Civil Engineering" gives the following rules for the strength of a circular plate *supported* all round the edge, prefaced by the remark that "the formula is founded on a theory which is only approximately true, but which nevertheless may be considered to involve no error of practical importance:"

$$M = \frac{Wb}{6\pi} = \frac{Pb^3}{24}.$$

Here

M = greatest bending moment;

W = total load uniformly distributed $= \frac{Pb^2\pi}{4}$;

b = diameter of plate in inches;

P = bursting pressure in pounds per square inch.

Calling t the thickness in inches, for a plate *supported* round the edges,

$$M = \frac{1}{6}\,42{,}000bt^2; \qquad \therefore\ \frac{Pb^2}{24} = 7000t^2.$$

For a plate fixed round the edges,

$$\frac{2}{3}\frac{Pb^2}{24} = 7000t^2; \quad \text{whence } P = \frac{t^2 \times 63{,}000}{r^2},$$

where r = radius of the plate.

Dr. Grashof gives a formula from which we have the following rule:

$$P = \frac{t^2 \times 72{,}000}{r^2}.$$

This formula of Grashof's has been adopted by Professor Unwin in his "Elements of Machine Design." These formulæ by Rankine and Grashof may be regarded as being practically the same.

…ying to make the rules given by these authorities agree with his experience of the strength of unstayed flat ends of cylindrical domes that had given way after long use, Mr. Wilson was led … the above rules give the breaking strength much lower than

specially when aided by the action of the corrosive acids in the
steam, will in time reduce the thickness of the plate, and bring
e destruction of an unstayed surface at a very low pressure.
e flat plates commence to deflect at very low pressures, they should
used without stays; but it is better to dish the plates when they are
ed by flues, tubes, etc.
inst the commonly accepted opinion that the limit of elasticity
ever be reached in testing a boiler or other structure, these experi-
iow that an exception should be made in the case of an unstayed
plate of a boiler, which will be safer when it has assumed a perma-
that will prevent its becoming grooved by the continual variation
ure in working. The hydraulic pressure in this case simply does
uld have been done before the plate was fixed, that is, dishes it.
se experiments appear to show that the mode of attaching by flange
inside or outside angle-iron exerts an important influence on the
in which the plate is strained by the pressure.
the plate is secured to an angle-iron, the stretching under pressure is,
ain extent, concentrated at the line of rivet-holes, and the plate par-
ther of a beam supported than fixed round the edge. Instead of the
increasing as the square of the thickness, when the plate is attached
gle-iron, it is probable that the strength does not increase even
as the thickness, since the plate gives way simply by stretching at
-holes, and the thicker the plate, the less uniformly is the strain
the different layers of which the plate may be considered to be
. When the plate is flanged, the flange becomes compressed by the
against the body of the plate, and near the rim, as shown by the
flexure, the inside of the plate is stretched more than the outside,
ay be by a kind of shearing action that the plate gives way along
where the crushing and stretching meet.
se tests appear to show that the rules deduced from the theoretical
tions of Lamé, Rankine, and Grashof are not confirmed by experi-
d are therefore not trustworthy.
raced Wrought-iron Heads of Boilers, etc. (*The Loco-*
Feb. 1890).—Few experiments have been made on the strength of
ls, and our knowledge of them comes largely from theory. Experi-
ave been made on small plates 1-16 of an inch thick, yet the data so
cannot be considered satisfactory when we consider the far thicker
at are used in practice, although the results agreed well with Ran-
ormula. Mr. Nichols has made experiments on larger heads, and
em he has deduced the following rule: "To find the proper thick-
a flat unstayed head, multiply the area of the head by the pressure

3. Head 26¼ inches in diameter, and ⅜ inch thick. The area 54 inches. Then, ⅜ × 44,800 × 10 = 168,000, and 168,000 ÷ 541 = 311 This head burst at 370 pounds.

4. Head 28½ inches in diameter and ⅜ inch thick. The are square inches; then, ⅜ × 44,800 × 10 = 168,000, and 168,000 ÷ 6 pounds. The actual bursting pressure was 300 pounds.

In the third experiment, the amount the plate bulged under d pressures was as follows:

At pounds per sq. in.....	10	20	40	80	120	140	17
Plate bulged............	1/32	1/16	⅛	¼	⅜	½	⅝

The pressure was now reduced to zero, "and the end sprang b inch, leaving it with a permanent set of 9-16 inch. The pressure of was again applied on 36 separate occasions during an interval of fi the bulging and permanent set being noted on each occasion, but any appreciable difference from that noted above.

The experiments described were confined to plates not widely diff their dimensions, so that Mr. Nichols's rule cannot be relied upon f that depart much from the proportions given in the examples.

Thickness of Flat Cast-iron Plates to resist Bu Pressures.—Capt. John Ericsson (Church's Life of Ericsson) g following rules: The proper thickness of a square cast-iron plate wi tained by the following: Multiply the side in feet (or decimals of a ¼ of the pressure in pounds and divide by 850 times the side in inc quotient is the square of the thickness in inches.

For a circular plate, multiply 11-14 of the diameter in feet by ¼ pressure on the plate in pounds. Divide by 850 times 11-14 of the d in inches. [Extract the square root.]

Prof. Wm. Harkness, *Eng'g News*, Sept. 5, 1895, shows that these r be put in a more convenient form, thus:

$$\text{For square plates} \quad T = 0.00495S\sqrt{p},$$

and

$$\text{For circular plates} \quad T = 0.00439D\sqrt{p},$$

where T = thickness of plate, S = side of the square, D = diamete circle, and p = pressure in lbs. per sq. in. Professor Harkness, h doubts the value of the rules, and says that no satisfactory theoreti tion has yet been obtained.

Strength of Stayed Surfaces.—A flat plate of thickness ported uniformly by stays whose distance from centre to centre is a, load p lbs. per square inch. Each stay supports pa^2 lbs. The stress on the plate is

$$f = \frac{2}{9}\frac{a^2}{t^2}p. \text{ (Unwin).}$$

SPHERICAL SHELLS AND DOMED BOILER-HE

To find the Thickness of a Spherical Shell to r given Pressure.—Let d = diameter in inches, and p the intern ure per square inch. The total pressure which tends to produce around the great circle will be $\frac{1}{4}\pi d^2p$. Let S = safe tensile st square inch, and t the thickness of metal in inches; then the resistan pressure will be πdtS. Since the resistance must be equal to the pr

$$\tfrac{1}{4}\pi d^2p = \pi dtS. \quad \text{Whence } t = \frac{pd}{4S}.$$

The same rule is used for finding the thickness of a hemispheri to a cylinder, as of a cylindrical boiler.

Thickness of a Domed Head of a boiler.—If S = saf stress per square inch, d = diameter of the shell in inches, and t = t of the shell, $t = pd \div 2S$; but the thickness of a hemispherical he same diameter is $t = pd \div 4S$. Hence if we make the *radius* of c of a domed head equal to the diameter of the boiler, we shall h $\frac{2pd}{4S} = \frac{pd}{2S}$, or the thickness of such a domed head will be equal to t *ness of the* [illegible]

tresses in Steel Plating due to Water-pressure, as in ting of vessels and bulkheads (*Engineering*, May 22, 1891, page 629). Ir. J. A. Yates has made calculations of the stresses to which steel plates subjected by external water-pressure, and arrives at the following consions:

ssume $2a$ inches to be the distance between the frames or other rigid ports, and let d represent the depth in feet, below the surface of the ter, of the plate under consideration, t = thickness of plate in inches, he deflection from a straight line under pressure in inches, and P = stress square inch of section.

or outer bottom and ballast tank plating, $a = 420\frac{t}{d}$, D should not be ater than $.05\frac{2a}{12}$, and $\frac{P}{2}$ not greater than 2 to 3 tons; while for bulkheads, $a = 2352\frac{t}{d}$, D should not be greater than $.1\frac{2a}{12}$, and $\frac{P}{2}$ not greater than ns. To illustrate the application of these formulæ the following cases e been taken:

For Outer Bottom, etc.			For Bulkheads, etc.		
ick-s of ling.	Depth below Water.	Spacing of Frames should not exceed	Thick-ness of Plating	Depth of Water.	Maximum Spacing of Rigid Stiffeners.
n.	ft.	in.	in.	ft.	ft. in.
[illegible]	20	About 21	$\frac{1}{2}$	20	9 10
[illegible]	10	" 42	$\frac{3}{8}$	20	7 4
[illegible]	18	" 18	$\frac{3}{8}$	10	14 8
[illegible]	9	" 36	$\frac{1}{4}$	20	4 10
[illegible]	10	" 20	$\frac{1}{4}$	10	9 8
[illegible]	5	" 40	$\frac{1}{8}$	10	4 10

would appear that the course which should be followed in stiffening heads is to fit substantially rigid stiffening frames at comparatively intervals, and only work such light angles between as are necessary making a fair job of the bulkhead.

HICK HOLLOW CYLINDERS UNDER TENSION.

rr, "Elasticity and Resistance of Materials," p. 36, gives

$$\left\{ \left(\frac{h+p}{h-p}\right)^{\frac{1}{2}} - 1 \right\}.$$

t = thickness; r = interior radius;
h = maximum allowable hoop tension at the interior of the cylinder;
p = intensity of interior pressure.

erriman gives

s = unit stress at inner edge of the annulus;
r = interior radius; t = thickness;
l = length.

e total stress over the area $2tl = 2sl\frac{rt}{r+t}$. . . . . (1)

e total interior pressure which tends to rupture the cylinder is $2rl \times p$. be the unit pressure, then $p = \frac{st}{r+t}$, from which one of the quantities r, or t can be found when the other three are given.

$$s = \frac{p(r+t)}{t}; \quad r = \frac{(s-p)t}{p}; \quad t = \frac{rp}{s-p}.$$

In eq. (1), if t be neglected in comparison with r, it reduces to $2st$ is the same as the formula for thin cylinders. If $t = r$, it become only half the resistance of the thin cylinder.

The formulæ given by Burr and by Merriman are quite different be seen by the following example: Let maximum unit stress at th edge of the annulus = 8000 lbs. per square inch, radius of cylinder = 4 interior pressure = 4000 lbs. per square inch. Required the thickne

By Burr, $t = 4\left\{\left(\frac{8000 + 4000}{8000 - 4000}\right)^{\frac{1}{2}} - 1\right\} = 4(\sqrt{3} - 1) = 2.928$ in

By Merriman, $t = \frac{4 \times 4000}{8000 - 4000} = 4$ inches.

Limit to Useful Thickness of Hollow Cylinders Jan. 4, 1884).—Professor Barlow lays down the law of the resisting of thick cylinders as follows:

"In a homogeneous cylinder, if the metal is incompressible, the on every concentric layer, caused by an internal pressure, varies in as the square of its distance from the centre."

Suppose a twelve-inch gun to have walls 15 inches thick.

$$\frac{\text{Pressure on exterior}}{\text{Pressure on interior}} = \frac{6^2}{21^2} = 1 : 12.25.$$

So that if the stress on the interior is 12¼ tons per square inch, th on the exterior is only 1 ton.

Let s = the stress on the inner layer, and s_1 that at a distance x fr axis; r = internal radius, R = external radius.

$$s_1 : s :: r^2 : x^2, \text{ or } s_1 = s\frac{r^2}{x^2}.$$

The whole stress on a section 1 inch long, extending from the int the exterior surface, is $S = sr \times \frac{R - r}{R}$.

In a 12-inch gun, let s = 40 tons, r = 6 in., R = 21 in.

$$S = 40 \times 6 \times \frac{21 - 6}{21} = 172 \text{ tons}.$$

Suppose now we go on adding metal to the gun outside: then R come so large compared with r, that $R - r$ will approach the val that the fraction $\frac{R - r}{R}$ becomes nearly unity.

Hence for an infinitely thick cylinder the useful strength coul exceed Sr (in this case 240 tons).

Barlow's formula agrees with the one given by Merriman.

Another statement of the gun problem is as follows: Using the

$$p = \frac{st}{r + t},$$

s = 40 tons, t = 15 in., r = 6 in., $p = \frac{40 \times 15}{21} = 28\frac{4}{7}$ tons per sq. in radius = 172 tons, the pressure to be resisted by a section 1 inch lo thickness of the gun on one side. Suppose thickness were doubled, t = 30 in.: $p = \frac{40 \times 30}{36} = 33\frac{1}{3}$ tons, or an increase of only 16 per ce

For short cast-iron cylinders, such as are used in hydraulic pre doubtful if the above formulæ hold true, since the strength of the cal portion is reinforced by the end. In that case the bursting would be higher than that calculated by the formula. A rule practice for such presses is to make the thickness = 1/10 of the i cumference, for pressures of 3000 to 4000 lbs. per square inch. T pressure would bring a stress upon the inner layer of 10,350 lbs. pe inch, as calculated by the formula; which would necessitate the u *best charco*[illegible]e the press reasonably safe.

HIN CYLINDERS UNDER TENSION.

e working pressure in lbs. per sq. in.;
meter in inches;
sile strength of the material, lbs. per sq. in.;
ckness in inches;
tor of safety;
io of strength of riveted joint to strength of solid plate.

$$fpd = 2Ttc; \quad p = \frac{2Ttc}{df}; \quad t = \frac{fpd}{2Tc}.$$

, $f = 5$, and $c = 0.7$; then

$$p = \frac{14000t}{d}; \quad t = \frac{dp}{14000}.$$

epresents the strength resisting rupture along a longitudinal esistance to rupture in a circumferential seam, due to pressure f the cylinder, we have $\frac{p\pi d^2}{4} = \frac{Tt\pi dc}{f}$;

$$\text{whence } p = \frac{4Ttc}{df}.$$

th to resist rupture around a circumference is twice as great ist rupture longitudinally; hence boilers are commonly single- circumferential seams and double-riveted in the longitudinal

HOLLOW COPPER BALLS.

per balls are used as floats in boilers or tanks, to control feed e valves, and regulate the water-level.

un up in halves from sheet copper, and a rib is formed on one is rib the other half fits, and the two are then soldered or er. In order to facilitate the brazing, a hole is left on one side allow air to pass freely in or out; and this hole is made use of o secure the float to its stem. The original thickness of the anything up to about 1-16 of an inch, if the spinning is done he, though thicker metal may be used when special machinery r forming it. In the process of spinning, the metal is thinned s by stretching; but the thinnest place is neither at the equator ., along the rib) nor at the poles. The thinnest points lie along assing around the ball parallel to the rib, one on each side of it, to a half of the way to the poles. Along these lines the thick- 10, 15, or 20 per cent less than elsewhere, the reduction depend t on the skill of the workman.

otive for October, 1891, gives two empirical rules for determin- kness of a copper ball which is to work under an external ollows:

$$\text{ss} = \frac{\text{diameter in inches} \times \text{pressure in pounds per sq. in.}}{16{,}000}.$$

$$\text{ss} = \frac{\text{diameter} \times \sqrt{\text{pressure}}}{1240}.$$

give the same result for a pressure of 166 lbs. only. Example: thickness of a 5-inch copper ball to sustain

f...........	50	100	150	166	200	250	lbs. per sq. in.
first rule...	.0156	.0312	.0469	.0519	.0625	.0781	inch.
second rule	.0285	.0403	.0494	.0518	.0570	.0637	"

DING-POWER OF NAILS, SPIKES, AND SCREWS.

. W. Wright, Western Society of Engineers, 1881.)

Spikes driven into dry cedar (cut 18 months):

s........................	5 × ¼ in. sq.	6 × ¼	6 × ½	5 × ⅜
n in........................	4¼ in.	5 in.	5 in.	4¼ in.
ance to drawing. *Av'ge, lbs.*	*857*	821	1691	1202
sts each......... Max. "	*1159*	923	2129	1556
Min. "	*766*	766	1120	687

A. M. Wellington found the force required to draw spikes 9/16 × 9/1 driven 4¼ inches into seasoned oak, to be 4281 lbs.; same spikes, etc., seasoned oak, 6523 lbs.

"Professor W. R. Johnson found that a plain spike ⅜ inch driven 3⅜ inches into seasoned Jersey yellow pine or unseasoned che required about 2000 lbs. force to extract it; from seasoned white oak 4000 and from well-seasoned locust 6000 lbs."

Experiments in Germany, by Funk, give from 2465 to 3940 lbs. (me many experiments about 3000 lbs.) as the force necessary to extract a ½-inch square iron spike 6 inches long, wedge-pointed for one inch driven 4½ inches into white or yellow pine. When driven 5 inches the required was about 1/10 part greater. Similar spikes 9/16 inches squ inches long, driven 6 inches deep, required from 3700 to 6745 lbs. to ex them from pine; the mean of the results being 4873 lbs. In all cases twice as much force was required to extract them from oak. The s were all driven across the grain of the wood. When driven with the g spikes or nails do not hold with more than half as much force.

Boards of oak or pine nailed together by from 4 to 16 tenpenny comm nails and then pulled apart in a direction lengthwise of the boards across the nails, tending to break the latter in two by a shearing a averaged about 300 to 400 lbs. per nail to separate them, as the res many trials.

Resistance of Drift-bolts in Timber.—Tests made by Ro Coolidge, in 1878.

						P
1st Test.	1 in. square	iron drove	30 in.	in white pine,	15/16-in. hole.	....
2d "	1 in. round	" "	34 "	" " "	13/16-in. "	....
3d "	1 in. square	" "	18 "	" " "	15/16-in. "	....
4th "	1 in. round	" "	22 "	" " "	13/16-in. "	....
5th "	1 in. round	" "	34 "	" Norw'y pine,	13/16-in. "	....
6th "	1 in. square	" "	30 "	" " "	15/16-in. "	....
7th "	1 in. square	" "	18 "	" " "	15/16-in. "	....
8th "	1 in. round	" "	22 "	" " "	13/16-in. "	....

NOTE.—In test No. 6 drift-bolts were not driven properly, holes not in line, and a piece of timber split out in driving.

Force required to draw Screws out of Norway Pin

½" diam. drive screw 4 in. in wood.	Power required, average	[illegible]
" " 4 threads per in. 5 in. in wood.	" " "	[illegible]
" " D'ble thr'd, 3 per in., 4 in. in "	" " "	[illegible]
" " Lag-screw, 7 per in., 1½ " "	" " "	1[illegible]
" " " " 6 " " 2½ " "	" " "	[illegible]
½ inch R.R. spike... .... 5 " "	" " "	[illegible]

Force required to draw Wood Screws out of Dry W —Tests made by Mr. Bevan. The screws were about two inches in l .22 diameter at the exterior of the threads, .15 diameter at the botto depth of the worm or thread being .035 and the number of threads inch equal 12. They were passed through pieces of wood half an in thickness and drawn out by the weights stated: Beech, 460 lbs.; as lbs.; oak, 760 lbs.; mahogany, 770 lbs.; elm, 665 lbs.; sycamore, 830 lbs.

Tests of Lag-screws in Various Woods were made by Cox, University of Iowa, 1891:

Kind of Wood.	Size Screw.	Size Hole bored.	Length in Tie.	Max. Resist. lbs.	T
Seasoned white oak...............	⅝ in.	½ in.	4½ in.	8037	
" " "...............	9/16 "	7/16 "	3 "	6480	
" " "...............	½ "	⅜ "	4½ "	8780	
Yellow-pine stick...	⅝ "	½ "	4 "	3800	
White cedar, unseasoned.........	⅝ "	½ "	4 "	3405	

In figuring area for lag-screws, the surface of a cylinder whose diam equal to that of the screw was taken. The length of the screw part in case was 4 inches.—*Engineering News*, 1891.

Cut *versus* **Wire Nails.**—Experiments were made at the Water Arsenal in 1893 on the comparative direct tensile adhesion, in pin *spruce*, of c nails. The results are stated by Prof. W. H. *as follows* are 18 series of tests, ten pairs of nails (a cut and a wire nail in each) 1, making a total of 1160 nails drawn. The tests were made in od in most instances, but some extra ones were made in white "box nails." The nails were of all sizes, varying from 1⅛ inches to length. In every case the cut nails showed the superior holding y a large percentage. In spruce, in nine different sizes of nails, lard and light weight, the ratio of tenacity of cut to wire nail 3 to 2, or, as he terms it, "a superiority of 47.45% of the former." 'finishing" nails the ratio was roughly 3.5 to 2; superiority 72%. nails (1⅛ to 4 inches long) the ratio was roughly 3 to 2; superiority ean superiority in spruce wood was 61%. In white pine, cut nails, h taper along the grain, showed a superiority of 100%, and with s the grain of 135%. Also when the nails were driven in the end t, i.e., along the grain, the superiority of cut nails was 100%, or the t to wire was 2 to 1. The total of the results showed the ratio of be about 3.2 to 2 for the harder wood, and about 2 to 1 for the for the whole taken together the ratio was 3.5 to 2. We are lude that under these circumstances the cut nail is superior to il in direct tensile holding-power by 72.74%.

Nail-holding Power of Various Woods.

(Watertown Experiments.)

f Wood.	Size of Nail.	Holding-power per square inch of Surface in Wood, lbs. Wire Nail.	Cut Nail.	Mean.
....	8d	167	450	405
	9 "		455	
	20 "		477	
	50 "		347	
			363	
	60 "		340	
. .. .	8 "	318	695	662
	10 "		755	
	50 "		596	
	60 "		604	
....	8 "	940	1340	1216
	20 "		1292	
	60 "		1018	
....	50 "		664	683
	60 "		702	
. ...	9 "	651	1179	1200
	20 "		1221	

Nail-holding Power of Various Woods.

F. W. Clay's Experiments. *Eng'g News*, Jan. 11, 1894.)

Wood.	Tenacity of 6d nails Plain.	Barbed.	Blued.	Mean.
....	106	94	135	111
....	190	180	270	196
....	78	132	219	143
....	226	300	555	360
....	141	201	319	220

le at the University of Illinois gave the resistance of a 1-in. round .6-inch hole perpendicular to the grain, as 6000 lbs. per lin. ft. in ,600 lbs. in oak. Experiments made at the East River Bridge nces of 12,000 and 15,000 lbs. per lin. ft. for a 1-in. round rod in n. and 14/16-in. diameter, respectively, in Georgia pine.

Holding-power of Bolts in White Pine.

(*Eng'g News*, September 26, 1891.)

	Round. Lbs.	Square. Lbs.
all plain 1-in. bolts.................. ...	8224	8200
all plain bolts, ⅝ to 1⅛ in..............	7805	8110
all bolts.....	8383	8598

t-bolts should be driven in holes 13/16 of their diameter, and *bolts in holes whose diameter is* 14/16 of the side of the square.

STRENGTH OF WROUGHT IRON BOLTS.

(Computed by A. F. Nagle.)

Diameter of Bolt, Inches.	Number of Threads.	Diameter of Bottom of Thread, Inches.	Area at Bottom of Thread, Square Inches.	Stress upon Bolt upon Basis of				
				3000 lbs. per sq. inch. lbs.	4000 lbs. per sq. inch. lbs.	5000 lbs. per sq. inch. lbs.	7000 lbs. per sq. inch. lbs.	10000 lbs. per sq. inch. lbs.
½	13	.38	.12	350	460	580	810	1160
9-16	12	.44	.15	450	600	750	1050	1500
⅝	11	.49	.19	560	750	930	1310	1870
¾	10	.60	.28	750	1130	1410	1980	2830
⅞	9	.71	.39	1180	1570	1970	2760	3940
1	8	.81	.52	1550	2070	2600	3630	5180
1⅛	7	.91	.65	1950	2600	3250	4560	6510
1¼	7	1.04	.84	2520	3360	4200	5900	8410
1⅜	6	1.12	1.00	3000	4000	5000	7000	10000
1½	6	1.25	1.23	3680	4910	6140	8600	12280
1⅝	5½	1.35	1.44	4300	5740	7180	10000	14360
1¾	5	1.45	1.65	4950	6600	8250	11560	16510
1⅞	5	1.57	1.95	5840	7800	9800	13640	19500
2	4½	1.66	2.18	6540	8720	10900	15260	21800
2¼	4½	1.92	2.88	8650	11530	14400	20180	28800
2½	4	2.12	3.55	10640	14200	17730	24830	35500
2¾	4	2.37	4.43	13290	17720	22150	31000	44300
3	3½	2.57	5.20	15580	20770	26000	36360	52000
3½	3¼	3.04	7.25	21760	29000	36260	50760	72500
4	3	3.50	9.62	28860	38500	48100	67350	96200

When it is known what load is to be put upon a bolt, and the judg the engineer has determined what stress is safe to put upon the ir down in the proper column of said stress until the required load i The area at the bottom of the thread will give the equivalent area bar to that of the bolt.

Effect of Initial Strain in Bolts.—Suppose that bolts a to connect two parts of a machine and that they are screwed up tig fore the effective load comes on the connected parts. Let P_1 = th tension on a bolt due to screwing up, and P_2 = the load afterward The greatest load may vary but little from P_1 or P_2, according former or the latter is greater, or it may approach the value P_1 + pending upon the relative rigidity of the bolts and of the parts co Where rigid flanges are bolted together, metal to metal, it is proba the extension of the bolts with any additional tension relieves th tension, and that the total tension is P_1 or P_2, but in cases wher packing, as india rubber, is interposed, the extension of the bolts m little affect the initial tension, and the total strain may be nearly Since the latter assumption is more unfavorable to the resistanc bolt, this contingency should usually be provided for. (See Unwi ments of Machine Design" for demonstration.)

STAND-PIPES AND THEIR DESIGN.

(Freeman C. Coffin, New England Water Works Assoc., *Eng. New* 16, 1893.) See also papers by A. H. Howland, Eng. Club of Phil. 18 Stephens, Amer. Water Works Assoc., *Eng. News*, Oct. 6 and 13, Kiersted, Rensselaer Soc. of Civil Eng., *Eng'g Record*, April 25 an 1891, and W. D. Pence, *Eng. News*, April and May, 1894.

The question of diameter is almost entirely independent of that o The efficient capacity must be measured by the length from the hi line to a point below which it is undesirable to draw the water on ac *loss of pressure* for fire-supply, whether that point is the actual b *the stand-pi*... above it. This allowable fluctuation ought not t *50 ft., in* ... This makes the diameter dependent upon t is, the first of which is the amount of the consumption during the ordi- interval between the stopping and starting of the pumps. This should er draw the water below a point that will give a good fire stream and e a margin for still further draught for fires. The second condition is maximum number of fire streams and their size which it is considered essary to provide for, and the maximum length of time which they are le to have to run before the pumps can be relied upon to reinforce n.

nother reason for making the diameter large is to provide for stability nst wind-pressure when empty.

e following table gives the height of stand-pipes beyond which they are afe against wind-pressures of 40 and 50 lbs. per square foot. The area rface taken is the height multiplied by one half the diameter.

eights of Stand-pipe that will Resist Wind-pressure by its Weight alone, when Empty.

Diameter, feet.	Wind, 40 lbs. per sq. ft.	Wind, 50 lbs. per sq. ft.
20	45	35
25	70	55
30	150	80
35	...	160

ave the above degree of stability the stand-pipes must be designed e outside angle-iron at the bottom connection.

form of anchorage that depends upon connections with the side near the bottom is unsafe. By suitable guys the wind-pressure is re- by tension in the guys, and the stand-pipe is relieved from wind that tend to overthrow it. The guys should be attached to a band or other shaped iron that completely encircles the tank, and rests ome sort of bracket or projection, and not be riveted to the tank. ould be anchored at a distance from the base equal to the height of t at which they are attached, if possible.

est plan is to build the stand-pipe of such diameter that it will resist d by its own stability.

Thickness of the Side Plates.

ressure on the sides is outward, and due alone to the weight of the or pressure per square inch, and increases in direct ratio to the and also to the diameter. The strain upon a section 1 inch in height oint is the total strain at that point divided by two—for each side is d to bear the strain equally. The total pressure at any point is the diameter in inches, multiplied by the pressure per square inch, he height at that point. It may be expressed as follows:

H = height in feet, and f = factor of safety;
d = diameter in inches;
p = pressure in lbs. per square inch;
$.434 = p$ for 1 ft. in height;
s = tensile strength of material per square inch;
T = thickness of plate.

the total strain on each side per vertical inch

$$= \frac{.434Hd}{2} = \frac{pd}{2}; \qquad T = \frac{.434Hdf}{2s} = \frac{pdf}{2s}.$$

offin takes $f = 5$, not counting reduction of strength of joint, equiv- an actual factor of safety of 3 if the strength of the riveted joint is 60 per cent of that of the plate.

nount of the wind strain per square inch of metal at any joint can l by the following formula, in which

H = height of stand-pipe in feet above joint;
T = thickness of plate in inches;
p = wind-pressure per square foot;
W = wind-pressure per foot in height above joint;
$W = Dp$ where D is the diameter in feet;
m = average leverage or movement about neutral axis *or central points in the circumference*; or,
m = *sine of 45°, or .707 times the radius* in feet,

Then the strain per square inch of plate

$$= \frac{(Hw)\frac{H}{2}}{\text{circ. in ft.} \times mT}.$$

Mr. Coffin gives a number of diagrams useful in the design of stan together with a number of instances of failures, with discussion probable causes.

Mr. Kiersted's paper contains the following : Among the most pr strains a stand-pipe has to bear are: that due to the static pressure water, that due to the overturning effect of the wind on an empty pipe, and that due to the collapsing effect, on the upper rings, of wind storms.

For the thickness of metal to withstand safely the static pres water, let

t = thickness of the plate iron in inches;
H = height of stand-pipe in feet;
D = diameter of stand-pipe in feet.

Then, assuming a tensile strength of 48,000 lbs. per square inch, a of safety of 4, and efficiency of double-riveted lap-joint equalling 0. strength of the solid plate,

$$t = .00036H \times D; \qquad H = \frac{10,000t}{3.6D};$$

which will give safe heights for thicknesses up to ⅝ to ¾ of an inch same formula may also apply for greater heights and thicknesses practical limits, if the joint efficiency be increased by triple riveting.

The conditions for the severest overturning wind strains exist wh stand pipe is empty.

Formula for wind-pressure of 50 pounds per square foot, when

d = diameter of stand-pipe in inches;
x = any unknown height of stand-pipe;
$x = \sqrt{80\pi dt} = 15.85\sqrt{dt}.$

The following table is calculated by these formulæ. The stand-p intended to be self-sustaining; that is, without guys or stiffeners.

Heights of Stand-pipes for Various Diameters an Thicknesses of Plates.

Thickness of Plate in Fractions of an Inch.	Diameters in Feet.											
	5	6	7	8	9	10	12	14	15	16	18	20
3-16	50	55	60	65	55	50	35	..	..	....	....	[illegible]
7-32	55	..	....	..	65	60	50	40	40	....	...	[illegible]
4-16	60	65	70	75	75	70	55	50	45	40	35	[illegible]
5-16	70	75	80	85	90	85	70	60	55	50	45	[illegible]
6-16	75	80	90	95	100	100	85	75	70	65	55	[illegible]
7-16	80	90	95	100	110	115	100	85	80	75	65	[illegible]
8-16	85	95	100	110	115	120	115	100	90	85	75	[illegible]
9-16	...	....	....	115	125	130	130	110	100	95	85	[illegible]
10-16	....	..	...	...	130	135	145	120	115	105	95	[illegible]
11-16	....	....	...	....	..	145	155	135	125	120	105	[illegible]
12-16	...	....	..	....	....	150	165	145	135	130	115	[illegible]
13-16	....	...	...	..	....	...	...	160	150	140	125	[illegible]
14-16	....	....	...	...	....	...	....	....	160	150	135	[illegible]
15-16	....	....	....	...	....	....	....	....	....	160	145	[illegible]
16-16	..	....	...	....	..	...	....	...	....	..	155	[illegible]

Heights to nearest 5 feet. Rings are to build 5 feet vertically.

Failures of Stand-pipes have been numerous in recent yea list showing 23 important failures inside of nine years is given in a pa Prof. W. D. Pence, *Eng'g. News*, April 5, 12, 19 and 26, May 3, 10 and 2 *June 7, 1894*. His discussion of the probable causes of the failures is valua

Allen, Engineers Club of Philadelphia, 1886, gives the following ickness of plates for stand pipes.

Wrought iron plate T. S. 48,000 pounds in direction of fibre, and pounds across the fibre. Strength of single riveted joint .4 that , and of double riveted joint, .7 that of the plate; wind pressure is per square foot; safety factor = 3.

otal height in feet; r = outer radius in feet; r' = inner radius = pressure per square inch; t = thickness in inches; d = outer n feet.

pipe filled and longitudinal seams double riveted

$$t = \frac{pr \times 12}{48,000 \times .7 \times \frac{1}{3}} = \frac{hd}{4301};$$

e empty and lateral seams, single riveted, we have by equating

$$r\left(\frac{h}{2}\right)^2 = 144 \times 6000\,(r^4 - r'^4)\,\frac{.7854}{r}, \text{ whence } r^4 - r'^4 = \frac{h^2 r^2}{27144}.$$

showing required Thickness of Bottom Plate.

in	Diameter.					
	5 feet.	10 feet.	15 feet.	20 feet.	25 feet.	30 feet.
	"	"	"	"	"	"
	† 7-64*	1/8 *	11-64*	15-64	19-64	23-64
	†11-64*	9-64*	7-32	9-32	23-64	27-64
	† 7-32	11-64*	1/4	21-64	13-32	31-64
	†19-64	3-16	9-32	3/8	15-32	9-16
	† 3/8	7-32	5-16	27-64	17-32	5/8
	†29-64	†15-64	23-64	15-32	37-64	45-64
		†23-64	7-16	37-64	47-64	7/8
		†33-64	17-32	45-64	7/8	1 3-64
		†11-16	39-64	13-16	1 1-32	1 7-32
		†29-32	45-64	15-16	1 11-64	1 25-64

* The minimum thickness should = 3-16".

N.B.—Dimensions marked † determined by wind-pressure.

r Tower at Yonkers, N. Y.—This tower, with a pipe 122 feet 20 feet diameter, is described in *Engineering News*, May 18, 1892. ickness of the lower rings is 11-16 of an inch, based on a tensile of 60,000 lbs. per square inch of metal, allowing 65% for the strength d joints, using a factor of safety of 3½ and adding a constant of The plates diminish in thickness by 1-16 inch to the last four the top, which are ¼ inch thick.

ntract for steel requires an elastic limit of at least 33,000 lbs. per nch; an ultimate tensile strength of from 56,000 to 66,000 lbs. per nch; an elongation in 8 inches of at least 20%, and a reduction of t least 45%. The inspection of the work was made by the Pittsburgh Laboratory. According to their report the actual conditions de- were as follows: Elastic limit from 34,020 to 39,420; the tensile from 58,330 to 65,390; the elongation in 8 inches from 22½ to 32%; n in area from 52.72 to 71.32%; 17 plates out of 141 were rejected in ection.

ROUGHT-IRON AND STEEL WATER-PIPES.

ted Steel Water-pipes (*Engineering News*, Oct. 11, 1890, and 91.)—The use of riveted wrought-iron pipe has been common in fic States for many years, the largest being a 44-inch conduit in on with the works of the Spring Valley Water Co., which supplies ncisco. The use of wrought iron and steel pipe has been neces- the West, owing to the *extremely high* pressures to be withstood *difficulties of transportation.* As an example: In conne[illegible] with the water supply of Virginia City and Gold Hill, Nev., there wa 1872 an 11½-inch riveted wrought-iron pipe, a part of which is und of 1720 feet.

In the East, the most important example of the use of riveted st pipe is that of the East Jersey Water Co., which supplies the city of The contract provided for a maximum high service supply of 25,00 lons daily. In this case 21 miles of 48-inch pipe was laid, some of it feet head. The plates from which the pipe is made are about 13 by 7 feet wide, open-hearth steel. Four plates are used to make on of pipe about 27 feet long. The pipe is riveted longitudinally with row, and at the end joints with a single row of rivets of varying corresponding to the thickness of the steel plates. Before being r the trench, two of the 27-feet lengths are riveted together, thus di still further the number of joints to be made in the trench and excavation to give room for jointing. All changes in the grade of line are made by 10° curves and all changes in line by 2¼, 5, 7½ curves. To lay on curved lines a standard bevel was used, and the curves are secured by varying the number of beveled joints in certain length of pipe.

The thickness of the plates varies with the pressure, but only th nesses are used, ¼, 5-16, and ⅜ inches, the pipe made of these th having a weight of 160, 185, and 225 lbs. per foot, respectively. At all the pipe was tested to pressure 1½ times that to which it is s jected when in place.

Mannesmann Tubes for High Pressures.—At the mann Works at Komotau, Hungary, more than 600 tons or 25 miles and 4-inch tubes averaging ¼ inch in thickness have been suc tested to a pressure of 2000 lbs. per square inch. These tubes were for a high-pressure water-main in a Chilian nitrate district.

This great tensile strength is probably due to the fact that, in a being much more worked than most metal, the fibres of the m spirally, as has been proved by microscopic examination. While tubes will hardly stand more than 200 lbs. per square inch, and wel are not safe above 1000 lbs. per square inch, the Mannesmann tu withstands 2000 lbs. per square inch. The length up to which be readily made is shown by the fact that a coil of 3-inch tube 70 was made recently.

For description of the process of making Mannesmann tubes s A. I. M. E., vol xix., 384.

STRENGTH OF VARIOUS MATERIALS. EXTR FROM KIRKALDY'S TESTS.

The recent publication, in a book by W. G. Kirkaldy, of the results thousand tests made during a quarter of a century by his father, D kaldy, has made an important contribution to our knowledge c the range of variation in strength of numerous materials. A c abstract of these results was published in the *American Machini* and 18, 1893, from which the following still further condensed ex taken:

The figures for tensile and compressive strength, or, as Kirk them, pulling and thrusting stress, are given in pounds per squar original section, and for bending strength in pounds of actual pounds per BD^2 (breadth × square of depth) for length of 36 inches supports. The contraction of area is given as a percentage of the area, and the extension as a percentage in a length of 10 inches, ex otherwise stated. The abbreviations T. S., E. L., Contr., and Ext. for the sake of brevity, to represent tensile strength, elastic limit, centages of contraction of area, and elongation, respectively.

Cast Iron.—44 tests: T. S. 15,468 to 28,740 pounds; 17 of these sound, the strength ranging from 15,468 to 21,257 pounds. Avera 23,805 pounds.

Thrusting stress, specimens 2 inches long, 1.34 to 1.5 in. diamete all sound, 94,352 to 131,912; one, unsound, 93,759; average of all, 11

Bending stress, bars about 1 in. wide by 2 in deep, cast on e mate stress 2876 to 3854; stress per BD^2 = 725 to 892; average, 830, dulus of rupture, R, = stress per BD^2 × length, = 29,520. Ult ... .29 to .40 in.; average .31 inch.

... tests of cast iron, 460 tests, 16 lots from various source th total range as follows: Pulling stress, 12,688 to 33,616 pounds; g stress, 66,363 to 175,950 pounds; bending stress, per BD^2, 505 to nds; modulus of rupture, R, 18,180 to 40,608. Ultimate deflection, inch.

ecimen which was the highest in thrusting stress was also the high-nding, and showed the greatest deflection, but its tensile strength 26,502.

ecimen with the highest tensile strength had a thrusting stress of and a bending strength, per BD^2, of 979 pounds with 0.41 deflection. imen lowest in T. S. was also lowest in thrusting and bending, but deflection. The specimen which gave .21 deflection had T. S., 19,188; 2. 104,281; and bending, 561.

Castings.—69 tests; tensile strength, 10,416 to 31,652; thrusting ltimate per square inch, 53,502 to 132,031.

nel Irons.—Tests of 18 pieces cut from channel irons. T. S. 53,141 pounds per square inch; contr. of area from 3.9 to 32.5 %. 0 in. from 2.1 to 22.5 %. The fractures ranged all the way from 100 % to 100 % crystalline. The highest T. S., 53,141, with 8.1 % contr. and was 100 % crystalline; the lowest T. S., 40,693, with 3.9 contr. and was 75 % crystalline. All the fibrous irons showed from 12.2 to ., 17.8 to 32.5 contr., and T. S. from 43,426 to 49,615. The fibrous therefore of medium tensile strength and high ductility. The e irons are of variable T. S., highest to lowest, and low ductility.

moor Iron Bars.—Three rolled bars 2¼ inches diameter; ten-elastic, 23,200 to 24,200; ultimate, 50,875 to 51,905; contraction, 44.4 xtension, 29.2 to 24.3. Three hammered bars, 4¼ inches diameter, ,100 to 24,200; ultimate, 46,810 to 49,223; contraction, 20.7 to 46.5; , 10.8 to 31.6. Fractures of all, 100 per cent fibrous. In the ham-s the lowest T. S. was accompanied by lowest ductility.

Bars, Various.—Of a lot of 80 bars of various sizes, some rolled hammered (the above Lowmoor bars included) the lowest T. S. ne) 40,808 pounds per square inch, was shown by the Swedish " bar 3¼ inches diameter, rolled. Its elastic limit was 19,150 contraction 68.7 % and extension 37.7 % in 10 inches. It was also ductile of all the bars tested, and was 100 % fibrous. The highest 0 pounds, with elastic limit, 29,400; contr., 36.6; and ext., 24.3 %, n by a "Farnley" 2-inch bar, rolled. It was also 100 % fibrous. t ductility 2.6 % contr., and 4.1 % ext., was shown by a 3¾-inch l bar, without brand. It also had the lowest T. S., 40,278 pounds, high elastic limit, 25,700 pounds. Its fracture was 95 % crystal-s of the two bars showing the lowest T. S., one was the most duc-e other the least ductile in the whole series of 80 bars.

y, high ductility is accompanied by low tensile strength, as in the rs, but the Farnley bars showed a combination of high ductility ensile strength.

otive Forgings, Iron.—17 tests: average, E. L., 30,420; T. S., tr., 36.5; ext. in 10 inches, 23.8.

n Anchor Forgings, Iron.—4 tests: average, E. L., 23,825; ; contr., 3.0; ext. in 10 inches, 3.8.

places these two irons in contrast to show the difference between bad work. The broken anchor material, he says, is of a most character, and a disgrace to any manufacturer.

Plate Girder.—Tensile tests of pieces cut from a riveted iron twenty years' service in a railway bridge. Top plate, average E. L., 26,600; T. S., 40,806; contr. 16.1; ext. in 10 inches, 7.8. te, average of 3 tests, E. L., 31,200; T. S., 44,288; contr., 13.3; ext. , 6.3. Web-plate, average of 3 tests, E. L., 28,000; T. S., 45,902; ext. in 10 inches, 8.9. Fractures all fibrous. The results of 30 different parts of the girder prove that the iron has undergone during twenty years of use.

lates.—Six plates 100 inches long, 2 inches wide, thickness vari-7 inch. T. S., 55,485 to 60,805; E. L., 29,600 to 33,200; contr., 52.9 17.05 to 18.57.

ridge Links.—40 links from Hammersmith Bridge, 1886.

	T. S.	E. L.	Contr.	Ext. in 100 in.	Fracture. Silky.	Granular.
Average of all.............	67,294	38,294	34.5%	14.11%		
Lowest T. S................	60,753	36,030	30.1	15.51	30%	70
Highest T. S. and E. L.....	75,936	44,166	31.2	12.42	15	85
Lowest E. L.............	64,044	32,441	34.7	13.43	30	70
Greatest Contraction.....	63,745	38,118	52.8	15.46	100	0
Greatest Extension.......	65,980	36,792	40.8	17.78	35	65
Least Contr. and Ext......	63,980	39,017	6.0	6.62	0	100

The ratio of elastic to ultimate strength ranged from 50.6 to 65.2 per ce[nt]; average, 56.9 per cent.

Extension in lengths of 100 inches. At 10,000 lbs. per sq. in., .018 to .0[...] mean, .020 inch; at 20,000 lbs. per sq. in. .049 to .063; mean, .055 inch; [at] 30,000 lbs. per sq. in., .083 to .100; mean, .090; set at 30,000 pounds per sq. [in.] 0 to .002; mean, 0.

The mean extension between 10,000 to 30,000 lbs. per sq. in. increased re[gu]larly at the rate of .007 inch for each 2000 lbs. per sq. in. increment of str[ess]. This corresponds to a modulus of elasticity of 28,571,429. The least incre[ase] of extension for an increase of load of 20,000 lbs. per sq. in., .065 inch, c[or]responds to a modulus of elasticity of 30,769,231, and the greatest, .076 in[ch], to a modulus of 26,315,789.

Steel Rails.—Bending tests, 5 feet between supports, 11 tests of fla[nge] rails 72 pounds per yard, 4.63 inches high.

	Elastic stress. Pounds.	Ultimate stress. Pounds.	Deflection at 50,000 Pounds.	Ultima[te] Deflecti[on].
Hardest...	34,200	60,960	3.24 ins.	8 ins.
Softest	32,000	56,740	3.76 "	8 "
Mean	32,763	59,209	3.53 "	8 "

All uncracked at 8 inches deflection.

Pulling tests of pieces cut from same rails. Mean results.

	Elastic Stress. per sq. in.	Ultimate Pounds. per sq. in.	Coutraction of area of fracture.	Extens[ion] in 10 in[ches].
Top of rails..........	44,200	83,110	19.9%	12.5%
Botton of rails.	40,900	77,820	30.9%	22.8%

Steel Tires.—Tensile tests of specimens cut from steel tires.

KRUPP STEEL.—262 Tests.

	E. L.	T. S.	Contr.	Ext. in 5 inches.
Highest........	69,250	119,079	31.9	18.1
Mean	52,869	104,112	29.5	19.7
Lowest	41,700	90,523	45.5	23.7

VICKERS, SONS & CO.—70 Tests.

	E. L.	T. S.	Contr.	Ext. in 5 inches.
Highest........	58,600	120,789	11.8	8.4
Mean......... ..	51,066	101,264	17.6	12.4
Lowest....	48,700	87,697	24.7	14.0

Note the correspondence between Krupp's and Vickers' steels as to te[n]sile strength and elastic limit, and their great difference in contraction a[nd extensi]on. *The fractures of the Krupp steel averaged 22 per cent sil[ky], [78 per cen]t granular*; of the Vicker steel, 7 per cent silky, 93 per cent gra[nular].

eel Axles.—Tensile tests of specimens cut from steel axles.

PATENT SHAFT AND AXLE TREE CO.—157 Tests.

	E. L.	T. S.	Contr.	Ext. in 5 inches.
st..	49,800	99,009	21.1	16.0
... .	36,267	72,099	33.0	23.6
t.	31,800	61,382	34.8	25.3

VICKERS, SONS & CO.—125 Tests.

	E. L.	T. S.	Contr.	Ext. in 5 inches.
st........	42,600	83,701	18.9	13.2
....	37,618	70,572	41.6	27.5
t	30,250	56,388	49.0	37.2

average fracture of Patent Shaft and Axle Tree Co. steel was 33 per lky, 67 per cent granular.

average fracture of Vickers' steel was 88 per cent silky, 12 per cent ar.

le tests of specimens cut from locomotive crank axles.

VICKERS'.—82 Tests, 1879.

	E. L.	T. S.	Contr.	Ext. in 5 inches.
t.... ...	26,700	68,057	28.3	18.4
... ...	24,146	57,922	32.9	24.0
....	21,700	50,195	52.7	36.2

VICKERS'.—78 Tests, 1884.

	E. L.	T. S.	Contr.	Ext. in 5 inches.
t...	27,600	64,873	27.0	20.8
....	23,573	56,207	32.7	25.9
....	17,600	47,695	35.0	27.2

FRIED. KRUPP.—43 Tests, 1889.

	E. L.	T. S.	Contr.	Ext. in 5 inches.
t........	31,650	66,868	48.6	35.6
... ...	29,491	61,774	47.7	32.3
....	21,950	55,172	55.3	35.6

l Propeller Shafts.—Tensile tests of pieces cut from two shafts, of four tests each. Hollow shaft, Whitworth. T. S., 61,290; E. L., contr., 52.8; ext. in 10 inches, 28.6. Solid Shaft, Vickers', T. S., E. L. 20,425; contr., 44.4; ext. in 10 inches, 30.7.

sting tests, Whitworth, ultimate, 56,201; elastic, 29,300; set at 30,000 18 per cent; set at 40,000 lbs., 2.04 per cent; set at 50,000 lbs., 3.82 per

sting tests, Vickers', ultimate, 44,602; elastic, 22,250; set at 30,000 lbs., r cent; set at 40,000 lbs., 4.69 per cent.

ring strength of the Whitworth shaft, mean of four tests, was 40,654 r square inch, or 66.3 per cent of the pulling stress. Specific gravity Whitworth steel, 7.867: of the Vickers', 7.856.

ing Steel.—Untempered, 6 tests, average, E. L., 67,916; T. S., contr., 37.8; ext. in 10 inches, 16.6. Spring steel untempered, 15 average, E. L., 38,785; T. S., 69,496; contr., 19.1; ext. in 10 inches, 29.8. two lots were shipped for the same purpose, viz., railway carriage rings.

l Castings.—44 tests, E. L., 31,816 to 35,567; T. S., 54,928 to 63,840; 1.67 to 15.8; ext., 1.45 to 15.1. Note the great variation in ductility. eel of the highest strength was also the most ductile.

eted Joints, Pulling Tests of Riveted Steel Plates, Triple Riveted Lap Joints, Machine Riveted, Holes Drilled.

width and thickn ss, *inches :*

9.50 × .25	*13.00 × .51*	*11.75 × .78*	12.25 × 1.01	14.00 × .77

ross sectional area square inches :

3.375	*6.63*	*9.165*	12.372	10.780

l. pounds :

20	*332,640*	*423,180*	528,000	455,210

Stress per square inch of gross area, joint:

59,058	50,172	46,173	42,696	4

Stress per square inch of plates, solid:

70,765	65,300	64,050	62,280	6

Ratio of strength of joint to solid plate:

83.46	76.83	72.09	68.55	(

Ratio net area of plate to gross:

73.4	65.5	62.7	64.7	

Where fractured:

plate at holes.	plate at holes.	plate at holes.	plate at holes.	sl

Rivets, diameter, area and number:

.45, .159, 24	.64, .321, 21	.95, .708, 12	1.08, .916, 12	.95,

Rivets, total area:

3.816	6.741	8.496	10.992

Strength of Welds.—Tensile tests to determine ratio of stre weld to solid bar.

IRON TIE BARS.—28 Tests.

Strength of solid bars varied from	43,201 to 57
Strenth of welded bars varied from	17,816 to 4
Ratio of weld to solid varied from	37.0

IRON PLATES.—7 Tests.

Strength of solid plate from	44,851 to 4
Strength of welded plate from	26,442 to 3
Ratio of weld to solid	57.7

CHAIN LINKS.—216 Tests.

Strength of solid bar from	49,122 to 5
Strength of welded bar from	39,575 to 4
Ratio of weld to solid	72.1

IRON BARS.—Hand and Electric Machine Welded.

32 tests, solid iron, average	52,444
17 " electric welded, average	46,836 ra
19 " hand " "	46,899

STEEL BARS AND PLATES.—14 Tests.

Strength of solid	54,226
Strength of weld	28,553
Ratio weld to solid	52.

The ratio of weld to solid in all the tests ranging from 37.0 to 95. of the great variation of workmanship in welding.

Cast Copper.—4 tests, average, E. L., 5900; T. S., 24,781; con ext., 21.8.

Copper Plates.—As rolled, 22 tests, .26 to .75 in. thick; E. L 18,650; T. S., 30,993 to 34,281; contr., 31.1 to 57.6; ext., 39.9 to 52.2. riation in elastic limit is due to difference in the heat at which t were finished. Annealing reduces the T. S. only about 1000 pound E. L. from 3000 to 7000 pounds.

Another series, .38 to .52 thick; 148 tests, T. S., 29,099 to 31,924; co to 56.7; ext. in 10 inches, 28.1 to 41.8. Note the uniformity i strength.

Drawn Copper.—74 tests (0.88 to 1.08 inch diameter); T. S., 40,557; contr., 37.5 to 64.1; ext. in 10 inches, 5.8 to 48.2.

Bronze from a Propeller Blade.—Means of two tests e centre and edge. Central portion (sp. gr. 8.320). E. L., 7550; T. contr., 25.4; ext. in 10 inches, 32.8. Edge portion (sp. gr. 8550). E T. S., 35,960; contr., 37.8; ext. in 10 inches, 47.9.

Cast German Silver.—10 tests: E. L., 13,400 to 29,100; T. S., 46,540; contr., 3.2 to 21.5; ext. in 10 inches, 0.6 to 10.2.

Thin Sheet Metal.—Tensile Strength.

German silver, 2 lots	75,816
Bronze, 4 lots	73,380
Brass, 2 lots	44,398
Copper, 9 lots	30,470
on, 13 lots, lengthway	44,331
on, 13 lots, crossway	39,838
el, 6 lots	49,253
rossway	55,948

Wire.—Tensile Strength.

silver, 5 lots	81,735 to 92,224
1 lot	78,049
s drawn, 4 lots	81,114 to 98,578
as drawn, 3 lots	37,607 to 46,494
annealed, 3 lots	34,936 to 45,210
(another lot), 4 lots	35,052 to 62,190
(extension 36.4 to 0.6%).	
ots	59,246 to 97,908
tension 15.1 to 0.7%).	
ots	103,272 to 318,823

eel of 318,823 T. S. was .047 inch diam., and had an extension of only ent; that of 103,272 T. S. was .107 inch diam. and had an extension r cent. One lot of .044 inch diam. had 267,114 T. S., and 5.2 per cent n.

Wire Ropes.

Selected Tests Showing Range of Variation.

ption.	Circumference, inches.	Weight per Fathom.	Strands. No. of Strands.	No. of Wires.	Diameter of Wires, inches.	Hemp Core.	Ultimate Strength, lbs.
ed.	7.70	53.00	6	19	.1563	Main	339,780
ized.	7.00	53.10	7	19	.1495	Main and Strands	314,860
ized.	6.38	42.50	7	19	.1347	Wire Core	295,920
ed.	7.10	37.57	6	30	.1004	Main and Strands	272,750
ized.	6.18	40.46	7	19	.1302	Wire Core	268,470
ized.	6.19	40.33	7	19	.1316	Wire Core	221,820
ed.	4.92	20.86	6	30	.0728	Main and Strands	190,890
ed.	5.36	18.94	6	12	.1104	Main and Strands	136,550
ed.	4.82	21.50	6	7	.1693	Main	129,710
ized.	3.65	12.21	6	19	.0755	Main	110,180
ized.	3.50	12.65	7	7	.122	Wire Core	101,440
ized.	3.82	14.12	6	7	.135	Main	98,670
ed.	4.11	11.35	6	12	.080	Main and Strands	75,110
ed.	3.31	7.27	6	12	.068	Main and Strands	55,095
ized.	3.02	8.62	6	7	.105	Main	49,555
ized.	2.68	6.26	6	6	.0963	Main and Strands	41,205
ed.	2.87	5.43	6	12	.0560	Main and Strands	38,555
ed.	2.46	3.85	6	12	.0472	Main and Strands	28,075
ized.	1.75	2.80	6	7	.0619	Main	24,552
ed.	2.04	2.72	6	12	.0378	Main and Strands	20,415
ed.	1.76	1.85	6	12	.0305	Main	14,634

p Ropes, Untarred.—15 tests of ropes from 1.53 to 6.90 inches rence, weighing 0.42 to 7.77 pounds per fathom, showed an ultim- gth of from 1670 to 33,808 pounds, the strength per fathom weight rom 2872 to 5534 pounds.

p Ropes, Tarred.—15 tests of ropes from 1.44 to 7.12 inches rence, weighing from 0.38 to 10.39 pounds per fathom, showed an strength of from 1046 to 31,549 pounds, the strength per fathom rying from 1767 to 5149 pounds.

n Ropes.—5 ropes, 2.48 to 6.51 inches circumference, 1.08 to 8.17 er fathom. Strength 3089 to 23,258 pounds, or 2474 to 3346 pounds m weight.

la Ropes.—35 tests; 1.19 to 8.90 inches circumference, 0.20 to nds per fathom. Strength 1280 to 65,550 pounds, or 3003 to 7394 er fathom weight.

Belting.

No. of lots.		Tensile str per square
11	Leather, single, ordinary tanned	3248 t
4	Leather, single, Helvetia	5631 t
7	Leather, double, ordinary tanned	2160 t
8	Leather, double Helvetia	4078 t
6	Cotton, solid woven	5648 t
14	Cotton, folded, stitched	4570 t
1	Flax, solid, woven	99
1	Flax, folded, stitched	63
6	Hair, solid, woven	3852 t
2	Rubber, solid, woven	4271 t

Canvas.—35 lots: Strength, lengthwise, 113 to 408 pounds per crossways, 191 to 468 pounds per inch.

The grades are numbered 1 to 6, but the weights are not given. strengths vary considerably, even in the same number.

Marbles.—Crushing strength of various marbles. 38 tests, 8 k Specimens were 6-inch cubes, or columns 4 to 6 inches diameter, and 12 inches high. Range 7542 to 13,720 pounds per square inch.

Granite.—Crushing strength, 17 tests; square columns 4 × 4 and 4 to 24 inches high, 3 kinds. Crushing strength ranges 10,026 to pounds per square inch. (Very uniform.)

Stones.—(Probably sandstone, local names only given.) 11 kin tests, 6 × 6, columns 12, 18 and 24 inches high. Crushing strength ra from 2105 to 12,122. The strength of the column 24 inches long is gen from 10 to 20 per cent less than that of the 6-inch cube.

Stones.—(Probably sandstone) tested for London & Northwestern way. 16 lots, 3 to 6 tests in a lot. Mean results of each lot ranged 3785 to 11,956 pounds. The variation is chiefly due to the stones being different lots. The different specimens in each lot gave results which erally agreed within 30 per cent.

Bricks.—Crushing strength, 8 lots; 6 tests in each lot; mean r ranged from 1835 to 9209 pounds per square inch. The maximum var in the specimens of one lot was over 100 per cent of the lowest. In the uniform lot the variation was less than 20 per cent.

Wood.—Transverse and Thrusting Tests.

	Tests.	Sizes abt. in square.	Span, inches.	Ultimate Stress.	$S = \frac{LW}{4BD^2}$.
Pitch pine	10	11½ to 12½	144	45,856 to 80,520	1096 to 1403
Dantzic fir	12	12 to 13	144	37,948 to 54,152	657 to 790
English oak	8	4½ × 12	120	32,856 to 39,084	1505 to 1779
American white oak	5	4½ × 12	120	23,624 to 26,952	1190 to 1372

Demerara greenheart, 9 tests (thrusting) 816
Oregon pine, 2 tests 58
Honduras mahogany, 1 test ....
Tobasco mahogany, 1 test ....
Norway spruce, 2 tests 525
American yellow pine, 2 tests 387
English ash, 1 test ....

Portland Cement.—(Austrian.) Cross-sections of specime ...lling tests only; cubes, 3 × 3 inches for thrusting tests nds per imperial bushel; residue, 0.7 per cent with sieve 2500 meshes re inch; 38.8 per cent by volume of water required for mixing; time g, 7 days; 10 tests to each lot. The mean results in lbs. per sq. in. follows:

Cement alone, Pulling.	Cement alone, Thrusting.	1 Cement, 2 Sand, Thrusting.	1 Cement, 3 Sand, Thrusting.	1 Cement, 4 Sand, Thrusting.
376	2910	893	407	228
420	3342	1023	494	275
451	3724	1172	594	338

land Cement.—Various samples pulling tests, 2 × 2¼ inches ction, all aged 10 days, 180 tests; ranges 87 to 643 pounds per square

TENSILE STRENGTH OF WIRE.

(From J. Bucknall Smith's Treatise on Wire.)

	Tons per sq. in. sectional area.	Pounds per sq. in. sectional area.
r annealed iron wire........................	25	56,000
hard drawn............................	35	78,400
er, steel wire............................	40	89,600
emens-Martin steel wire...............	60	134,000
arbon ditto (or "improved")	80	179,200
le cast-steel "improved" wire...	100	224,000
oved" cast-steel "plough".....	120	268,800
l qualities of tempered and improved cast-el wire may attain..........................	150 to 170	336,000 to 380,800

MISCELLANEOUS TESTS OF MATERIALS.

orts of Work of the Watertown Testing-machine in 1883.

TESTS OF RIVETED JOINTS, IRON AND STEEL PLATES.

Thickness Plate.	Diameter, Rivets, inches.	Diameter, Punched Holes, inches.	Width Plate Tested, inches.	No. Rivets.	Pitch Rivets, inches.	Tensile Strength Joint in Net Section of Plate per square inch, pounds.	Tensile Strength Plate per square inch, pounds.	Efficiency of Joint, Per Cent.
	11-16	¾	10½	6	1¾	39,300	47,180	47.0 ‡
	11-16	¾	10½	6	1¾	41,000	47,180	49.0 ‡
	¾	13-16	10	5	2	35,650	44,615	45.6 ‡
	¾	13-16	10	5	2	35,150	44,615	44.9 ‡
	11-16	¾	10	5	2	46,360	47,180	59.9 §
	11-16	¾	10	5	2	46,875	47,180	60.5 §
	¾	13-16	10	5	2	46,400	44,615	59.4 §
	¾	13-16	10	5	2	46,140	44,615	59.2 §
	1	1 1-16	10½	4	2⅝	44,260	44,635	57.2 §
	1	1 1-16	10½	4	2⅝	42,350	44,635	54.9 §
	1⅛	1 3-16	11.9	4	2.9	42,310	46,590	52.1 §
	1⅛	1 3-16	11.9	4	2.9	41,920	46,590	51.7 §
	¾	13-16	10½	6	1¾	61,270	53,330	59.5 ‡
	¾	13-16	10½	6	1¾	60,830	53,330	59.1 ‡
	15-16	1	10	5	2	47,530	57,215	40.2 ‡
	15-16	1	10	5	2	49,840	57,215	42.8 ‡
	11-16	¾	10	5	2	62,770	53,330	71.7 §
	11-16	¾	10	5	2	61,210	53,330	69.8 §
	15-16	1	10	5	2	68,920	57,215	57.1 §
	15-16	1	10	5	2	66,710	57,215	55.0 §
	1	1 1-16	9½	4	2⅜	62,180	52,445	63.4 §
	1	1 1-16	9½	4	2⅜	62,590	52,445	63.8 §
	1⅛	1 3-16	10	4	2½	51,650	51,545	54.0 §
	1⅛	1 3-16	10	4	2½	54,200	51,545	53.4 §

rou. † Steel. ‡ Lap-joint. § Butt-joint

The efficiency of the joints is found by dividing the maximum tens stress on the gross sectional area of plate by the tensile strength of t material.

COMPRESSION TESTS OF 3 × 3 INCH WROUGHT-IRON BARS

Length, inches.	Tested with Two Pin Ends, Pins 1½ inch in Diameter. Ultimate Compressive Strength pounds per square inch.	Tested with Two Flat Ends, Ultimate Compressive Strength, pounds per square inch.	Tested with On Flat and One P End. Ultimate Compressive Strength, poun per square inc
30	28,260 31,990	....	....
60	26,310 26,640	....	....
90	24,030 25,380	26,780 25,580	25,120 25,190
120	20,660 20,200	23,010 22,450	22,450 21,870
150	16,520 17,840	....	....
180	13,010 15,700	....	....

Tested with two pin-ends. Length of bars 120 inches. Diameter of Pins.	Ult. Comp. St per sq. in., lb
⅞ inch	16,250
1⅛ inches	17,740
1⅞ "	21,400
2¼ "	22,210

TENSILE TEST OF SIX STEEL EYE-BARS.

COMPARED WITH SMALL TEST INGOTS.

The steel was made by the Cambria Iron Company, and the eye-bar h made by Keystone Bridge Company by upsetting and hammering. All bars were made from one ingot. Two test pieces, ¾-inch round, rolled f a test-ingot, gave elastic limit 48,040 and 42,210 pounds; tensile stren 73,150 and 69,470 pounds, and elongation in 8 inches, 22.4 and 25.6 per c respectively. The ingot from which the eye-bars were made was 14 in square, rolled to billet, 7 × 6 inches. The eye-bars were rolled to 6½ × 1 Chemical tests gave carbon .27 to .30; manganese, .61 to .73; phospho .074 to .098.

Gauged Length, inches.	Elastic limit, lbs. per sq. in.	Tensile strength per sq. in., lbs.	Elongation per cent, in Gauged Lengt
160	37,480	67,800	15.8
160	36,650	64,000	6.96
160	......	71,560	8.6
200	37,600	68,720	12.3
200	35,810	65,850	12.0
200	33,230	64,410	16.4
200	37,640	68,290	13.9

The average tensile strength of the ¾-inch test pieces was 71,310 lbs., of the eye-bars 67,230 lbs., a decrease of 5.7%. The average elastic lim the test pieces was 45,150 lbs., that of the eye-bars 36,402 lbs., a decreas 19.4%. The elastic limit of the test pieces was 63.3% of the ultimate stren that of the eye-bars 54.2% of the ultimate strength.

MPRESSION OF WROUGHT-IRON COLUMNS, LATTICED BOX AND SOLID WEB.

ALL TESTED WITH PIN ENDS.

Columns made of	Length, feet.	Sectional Area, square inch.	Total Weight of Column, pounds.	Ultimate Strength, per square inch, pounds.
channel, solid web..................	10.0	9.831	432	30,220
" " "	15.0	9.977	592	21,050
" " "	20.0	9.762	755	16,220
" " "	20.0	16.281	1,290	22,540
" " "	26.8	16.141	1,645	17,570
channels, with 5-16-in. continuous es	26.8	19.417	1,940	25,290
ch continuous plates and angles. th of plates, 12 in., 1 in. and 7.35 in.	26.8	16.168	1,765	28,020
ch continuous plates and angles. es 12 in. wide.	26.8	20.954	2,242	25,770
channels, latticed..................	13.3	7.628	679	33,910
" "	20.0	7.621	924	34,120
" "	26.8	7.673	1,255	29,870
channels, latticed, swelled sides..	13.4	7.624	684	33,530
" " " " ..	20.0	7.517	921	33,390
" " " " ..	26.8	7.702	1,280	30,770
" "	16.8	11.944	1,470	33,740
" "	25.0	12.175	1,926	32,440
channels, latticed, swelled sides.	16.7	12.366	1,549	31,130
" " " " .	25.0	11.932	1,962	32,740
ch channels, latticed one side; con-ous plate one side	25.0	17.622	1,848	26,190
ch channels, latticed one side; con-ous plate one side................	25.0	17.721	1,827	17,270

s in centre of gravity of channel bars and continuous plate, 1.63 from centre line of channel bars.
s placed in centre of gravity of channel bars.

EFFECT OF COLD-DRAWING ON STEEL.

e tensile bars and two compression bars, cut from the same bar of lled steel, from the Norway Steel and Iron Company:

	Tensile strength per sq. in., lbs.	Elongation, per cent.
e of the original hot-rolled bar, length inches, diameter 2.03 inches. Gauged ngth 30 inches...........................	55,400	23.9
meter reduced in compressing dies (one ss), .094 inch. Gauged length 20 inches.	70,420	2.7
meter reduced in compression dies (one ss), .222 inch. Gauged length 20 inches.	81,890	0.075

	Compress. Stress, lbs. per sq. in.	Amount of Compress., in.	Compress. set, in.
npression test of cold-drawn bar (same No. 3). Length 4 inches, diameter 808 inches	75,000	.0562	.0395
, same as No. 4............................	75,000	.0578	.0400

es 4 and 5 both had diameters increased in the middle to 1.821 in... the ends to 1.813 inches.

TESTS OF AMERICAN WOODS. (See also pa

In all cases a large number of tests were made of each and maximum results only are given. All of the test speci tional area of 1.575 × 1.575 inches. The transverse test spec inches between supports, and the compressive test spec

inches long. Modulus of rupture calculated from formul

load in pounds at the middle, l = length in inches, b = bre

Name of Wood.	Transverse Tests. Modulus of Rupture.	
	Min.	Max.
Cucumber tree (*Magnolia acuminata*)..	7,440	12,050
Yellow poplar white wood (*Liriodendron tulipifera*)....	6,560	11,756
White wood, Basswood (*Tilia Americana*)....	6,720	11,530
Sugar-maple, Rock-maple (*Acer saccharinum*....	9,680	20,130
Red maple (*Acer rubrum*) ..	8,610	13,450
Locust (*Robinia pseudacacia*)....	12,200	21,730
Wild cherry (*Prunus serotina*)..........	8,310	16,800
Sweet gum (*Liquidambar styraciflua*)..	7,470	11,130
Dogwood (*Cornus florida*)..........	10,190	14,560
Sour gum, Pepperidge (*Nyssa sylvatica*).	9,830	14,300
Persimmon (*Diospyros Virginiana*). ...	10,290	18,500
White ash (*Fraxunis Americana*).......	5,950	15,800
Sassafras (*Sassafras officinale*).....	5,180	10,150
Slippery elm (*Ulmus fulva*).......	10,220	13,952
White elm (*Ulmus Americana*).........	8,250	15,070
Sycamore; Buttonwood (*Platanus occidentalis*)........	6,720	11,360
Butternut; white walnut (*Juglans cinerea*)..	4,700	11,740
Black walnut (*Juglans nigra*).....	8,400	16,320
Shellbark hickory (*Carya alba*)..... ...	14,870	20,710
Pignut (*Carya porcina*)..	11,560	19,430
White oak (*Quercus alba*)..	7,010	18,360
Red oak (*Quercus rubra*).	9,760	18,370
Black oak (*Quercus tinctoria*)...........	7,900	18,420
Chestnut (*Castanea vulgaris*).	5,950	12,870
Beech (*Fagus ferruginea*)	13,850	18,840
Canoe-birch, paper-birch (*Betula papyracea*).....	11,710	17,610
Cottonwood (*Populus monilifera*).......	8,390	13,430
White cedar (*Thuja occidentalis*).......	6,310	9,530
Red cedar (*Juniperus Virginiana*). ...	5,640	15,100
Cypress (*Saxodium Distichum*)..........	9,530	10,030
White pine (*Pinus strobus*)..............	5,610	11,530
Spruce pine (*Pinus glabra*).............	3,780	10,980
Long-leaved pine, Southern pine (*Pinus palustris*)	9,220	21,060
White spruce (*Picea alba*)......	9,900	11,650
Hemlock (*Tsuga Canadensis*)...........	7,590	14,680
Red fir, yellow fir (*Pseudotsuga Douglasii*)...	8,220	17,920
Tamarack (*Larix Americana*)	10,080	16,770

SHEARING STRENGTH OF IRON AN

H. V. Loss in American Engineer and Railroad Journal *1893*, describes an extensive series of experiments on t and steel bars in shearing machines. Some of his resul of penetration at point of maximum resistance for soft steel bars ndent of the width, but varies with the thickness. If d = depth of on and t = thickness, $d = .3t$ for a flat knife, $d = .25\ t$ for a 4° bevel d $d = .16\ \sqrt{t^2}$ for an 8° bevel knife. The ultimate pressure per inch in flat steel bars is approximately 50,000 lbs. $\times\ t$. The energy con- foot pounds per inch width of steel bars is, approximately: 1″ 0 ft.-lbs.; 1⅛″, 2500; 1¾″, 3700; 1⅞″, 4500; the energy increasing wer rate than the thickness. Iron angles require more energy el angles of the same size; steel breaks while iron has to be For hot-rolled steel the resistance per square inch for rectan- tions varies from 4400 lbs. to 20,500 lbs., depending partly upon its and partly upon the size of its cross-area, which latter element y but greatly indicates the temperature, as the smaller dimensions considerably longer time to reduce them down to size, which time eans loss of heat.

ot probable that the resistance in practice can be brought very low the lowest figures here given—viz., 4400 lbs. per square inch— rease of 1000 lbs. will henceforth mean a considerable increase in ction and temperature.

DING-POWER OF BOILER-TUBES EXPANDED INTO TUBE-SHEETS.

ments by Chief Engineer W. H. Shock, U. S. N., on brass tubes, 2½ diameter, expanded into plates ¾-inch thick, gave results ranging 50 to 46,000 lbs. Out of 48 tests 5 gave figures under 10,000 lbs., 12 n 10,000 and 20,000 lbs., 18 between 20,000 and 30,000 lbs., 10 between nd 40,000 lbs., and 3 over 40,000 lbs.

riments by Yarrow & Co., on steel tubes, 2 to 2¼ inches diameter, sults similarly varying, ranging from 7900 to 41,715 lbs., the majority g from 20,000 to 30,000 lbs. In 15 experiments on 4 and 5 inch tubes ain ranged from 20,720 to 68,040 lbs. Beading the tube does not neces- give increased resistance, as some of the lower figures were obtained eaded tubes. (See paper on Rules Governing the Construction of Boilers, Trans. Engineering Congress, Section G, Chicago, 1893.)

CHAINS.

eight per Foot, Proof Test and Breaking Weight.

(Pennsylvania Railroad Specifications.)

ninal meter Wire, ches.	Description.	Specifications.		
		Weight per foot, lbs.	Proof Test, lbs.	Breaking Weight, lbs.
32	Lock-chain	0.20	....	....
16	Fire-door chain	0.35	....	....
16	Crossing-gate chain	0.70	1500	3000
	Sprocket-wheel chain	1.10	3000	5500
	Brake-chain	1.50	3500	7000
16	Crane-chain	1.50	4000	7500
16	Drop-bottom branch chain	1.90	5000	9500
	Crane-chain	1.90	5500	10,000
	Drop-bottom main chain	2.50	7000	12,500
	Crane-chain	2.50	7500	13,000
	Safety "	4.00	11,000	20,000
	Crane "	4.00	11,000	20,000
	Log "	5.50	16,000	29,000
	Crane "	5.50	16,000	29,000
	" "	7.40	22,000	40,000
	" "	9.50	30,000	55,000
	" "	12.00	40,000	66,000
	" "	15.00	50,000	82,000
	" "	21.00	70,000	116,000

gation of all sizes, 10 per cent. All chain must stand the prescribed test without deformation.

British Admiralty Proving Tests of Chain Cables.—links. Minimum size in inches and 16ths. Proving test in tons of 22[illegible]

Min. Size:	$\frac{11}{16}$	$\frac{12}{16}$	$\frac{13}{16}$	$\frac{14}{16}$	$\frac{15}{16}$	1	$1\frac{1}{16}$	$1\frac{2}{16}$	$1\frac{3}{16}$	$1\frac{4}{16}$	$1\frac{5}{16}$
Test, tons:	$8\frac{10}{20}$	$10\frac{2}{20}$	$11\frac{18}{20}$	$13\frac{15}{20}$	$15\frac{15}{20}$	18	$20\frac{6}{20}$	$22\frac{15}{20}$	$25\frac{7}{20}$	$28\frac{2}{20}$	31
Min. Size:	1^{8}	1^{9}	1^{10}	1^{11}	1^{12}	1^{13}	1^{14}	1^{15}	2	2^{1}	2
Test, tons:	$40\frac{10}{20}$	$43\frac{18}{20}$	$47\frac{10}{20}$	$51\frac{5}{20}$	$55\frac{2}{20}$	$59\frac{2}{20}$	$63\frac{5}{20}$	$67\frac{11}{20}$	72	$76\frac{11}{20}$	81

Wrought-iron Chain Cables.—The strength of a chain less than twice that of a straight bar of a sectional area equal to that side of the link. A weld exists at one end and a bend at the other, quiring at least one heat, which produces a decrease in the strength report of the committee of the U. S. Testing Board, on tests of wrough and chain cables contains the following conclusions. That beyond when made of American bar iron, with cast-iron studs, the studded inferior in strength to the unstudded one.

"That when proper care is exercised in the selection of material, tion of 5 to 17 per cent of the strongest may be expected in the res of cables. Without this care, the variation may rise to 25 per cent.

"That with proper material and construction the ultimate resist the chain may be expected to vary from 155 to 170 per cent of that bar used in making the links, and show an average of about 163 per

"That the proof test of a chain cable should be about 50 per cent ultimate resistance of the weakest link."

The decrease of the resistance of the studded below the unstudd is probably due to the fact that in the former the sides of the link remain parallel to each other up to failure, as they do in the latter, sult is an increase of stress in the studded link over the unstudded proportion of unity, to the secant of half the inclination of the side former to each other.

From a great number of tests of bars and unfinished cables, the tee considered that the average ultimate resistance, and proof tests cables made of the bars, whose diameters are given, should be such shown in the accompanying table.

ULTIMATE RESISTANCE AND PROOF TESTS OF CHAIN CABLES.

Diam. of Bar.	Average resist. = 163% of Bar.	Proof Test.	Diam. of Bar.	Average resist. = 163% of Bar.	Pro
Inches.	Pounds.	Pounds.	Inches.	Pounds.	P
1 1/16	71,172	33,840	1 9/16	162,283	[illegible]
1 1/16	79,544	37,820	1⅝	174,475	[illegible]
1⅛	88,445	42,053	1 11/16	187,075	[illegible]
1 3/16	97,731	46,468	1¾	200,074	[illegible]
1¼	107,440	51,084	1 13/16	213,475	[illegible]
1 5/16	117,577	55,903	1⅞	227,271	[illegible]
1⅜	128,129	60,920	1 15/16	241,463	[illegible]
1 7/16	139,103	66,138	2	256,040	[illegible]
1½	150,485	71,550			

STRENGTH OF GLASS.

(Fairbairn's "Useful Information for Engineers," Second Seri

	Best Flint Glass.	Common Green Glass.	Ex Cr
Mean specific gravity	3.078	2.528	
Mean tensile strength, lbs. per sq. in., bars..	2,413	2,896	
do. thin plates.	4,200	4,800	
Mean crush'g strength, lbs. p. sq. in., cyl'drs.	27,582	39,876	
do. cubes.	13,130	20,206	

The bars in tensile tests were about ½ inch diameter. The crush were made on cylinders about ¾ inch diameter and from 1 to 2 inch and on cubes approximately 1 inch on a side. The mean transverse of glass, as calculated by Fairbairn from a mean tensile strength lbs. and a mean compressive strength of 30,150 lbs. per sq. in., is, supported at the ends and loaded in the middle,

$$w = 3140\frac{bd^2}{l},$$

w = breaking weight in lbs., b = breadth, d = depth, and l = length, s. Actual tests will probably show wide variations in both direc- m the mean calculated strength.

NGTH OF COPPER AT HIGH TEMPERATURES.

itish Admiralty conducted some experiments at Portsmouth Dock- 877, on the effect of increase of temperature on the tensile strength r and various bronzes. The copper experimented upon was in rods ameter, having a tensile strength of about 25 tons per square inch. llowing table shows some of the results:

rature hr.	Tensile Strength in lbs. per sq. in.	Temperature Fahr.	Tensile Strength in lbs. per sq. in.
pheric.	23,115	Atmospheric.	
0°	23,366	400°	21,105
0°	22,110	500°	19,597
0°	21,607		

a temperature of 400° F. the loss of strength was only about 10 per l at 500° F. the loss was 16 per cent. The temperature of steam at ressure is 382° F., so that according to these experiments the loss th at this point would not be a serious matter. Above a tempera- 00° the strength is seriously affected.

STRENGTH OF TIMBER.

gth of Long-leaf Pine (Yellow Pine, *Pinus Palustris*) from (Bulletin No. 8, Forestry Div., Dept. of Agriculture, 1893. Tests J. B. Johnson.)

llowing is a condensed table of the range of results of mechanical over 2000 specimens, from 26 trees from four different sites in ; reduced to 15 per cent moisture:

	Butt Logs.	Middle Logs.	Top Logs.	Av'g of all Butt Logs.
ravity	0.449 to 1.039	0.575 to 0.859	0.484 to 0.907	0.767
se strength, $\frac{3\,WL}{2\,bh^2}$	4,762 to 16,200	7,640 to 17,128	4,268 to 15,554	12,614
o. at elast. limit	4,930 to 13,110	5,540 to 11,790	2,553 to 11,950	9,460
last., thous. lbs.	1,119 to 3,117	1,136 to 2,982	842 to 2,697	1,926
elast. resilience, unds per cub. in.	0.23 to 4.69	1.34 to 4.21	^ 09 to 4.65	2.98
endwise, str. per lbs.	4,781 to 9,850	5,030 to 9,300	4,587 to 9,100	7,452
across grain, h per sq. in., lbs.	675 to 2,094	656 to 1,445	584 to 1,766	1,598
rength per sq. in.	8,600 to 31,890	6,330 to 29,500	4,170 to 23,280	17,359
strength (with mean per sq. in.	464 to 1,299	539 to 1,230	484 to 1156	866

f the deductions from the tests were as follows:

the exception of tensile strength a reduction of moisture is ac- d by an increase in strength, stiffness, and toughness.

ation in strength goes generally hand-in-hand with specific gravity.

he first 20 or 30 feet in height the values remain constant; then decrease of strength which amounts at 70 feet to 20 to 40 per cent f the butt-log.

hearing parallel with the grain and crushing across and parallel grain, practically no difference was found.

e beams appear 10 to 20 per cent weaker than small pieces.

pression tests endwise seem to furnish the best average statement ue of wood, and if one test only *can* be made, this is the safest, as recognized by *Bauschinger*.

timber is in no respect inferior to unbled timber.

The figures for crushing across the grain represent the load r cause a compression of 15 per cent. The relative elastic resilienc pounds per cubic inch of the material, is obtained by measurin of the plotted-strain diagram of the transverse test from the or point in the curve at which the rate of deflection is 50 per cent gr the rate in the earlier part of the test where the diagram is a str This point is arbitrarily chosen since there is no definite "elastic timber as there is in iron. The "strength at the elastic limi strength taken at this same point. Timber is not perfectly elast load if left on any great length of time.

The long-leaf pine is found in all the Southern coast states fr Carolina to Texas. Prof. Johnson says it is probably the strong in large sizes to be had in the United States. In small selected s other species, as oak and hickory, may exceed it in strength a ness. The other Southern yellow pines, viz., the Cuban, shor the loblolly pines are inferior to the long-leaf about in the ratio specific gravities; the long-leaf being the heaviest of all the averages (kiln-dried) 48 pounds per cubic foot, the Cuban 47, the 40, and the loblolly 34 pounds.

Strength of Spruce Timber.—The modulus of rupture is given as follows by different authors: Hatfield, 9900 lbs. per sq Rankine, 11,100; Laslett, 9045; Trautwine, 8100; Rodman, 61 wine advises for use to deduct one-third in the case of knotty timber.

Prof. Lanza, in 25 tests of large spruce beams, found a m rupture from 2995 to 5666 lbs.; the average being 4613 lbs. T average beams, ordered from dealers of good repute. Two selected stock, seasoned four years, gave 7562 and 8748 lbs. Th of elasticity ranged from 897,000 to 1,588,000, averaging 1,294,000.

Time tests show much smaller values for both modulus of ru modulus of elasticity. A beam tested to 5800 lbs. in a screw ma left over night, and the resistance was found next morning to hav to about 3000, and it broke at 3500.

Prof. Lanza remarks that while it was necessary to use larger safety, when the moduli of rupture were determined from tests wi pieces, it will be sufficient for most timber constructions, except in to use a factor of four. For breaking strains of beams, he state better engineering to determine as the safe load of a timber bea that will not deflect it more than a certain fraction of its span, 1/300 to 1/400 of its length.

Properties of Timber.

(N. J. Steel & Iron Co.'s Book.)

Description.	Weight per cubic foot, in lbs.	Tensile Strength per sq. inch, in lbs.	Crushing Strength per sq. inch, in lbs.	Relative Strength for Cross Breaking. White Pine = 100
Ash	43 to 55.8	11,000 to 17,207	4,400 to 9,363	130 to 180
Beech	43 to 53.4	11,500 to 18,000	5,800 to 9,363	100 to 144
Cedar	50 to 56.8	10,300 to 11,400	5,600 to 6,000	55 to 63
Cherry	....	....	....	130
Chestnut	33	10,500	5,350 to 5,600	96 to 123
Elm	34 to 36.7	13,400 to 13,489	6,831 to 10,331	96
Hemlock	....	8,700	5,700	88 to 95
Hickory	....	12,800 to 18,000	8,925	150 to 210
Locust	44	20,500 to 24,800	9,113 to 11,700	132 to 227
Maple	49	10,500 to 10,584	8,150	122 to 220
Oak, White	45 to 54.5	10,253 to 19,500	4,684 to 9,509	130 to 177
Oak, Live	70	....	6,850	155 to 189
Pine, White	30	10,000 to 12,000	5,000 to 6,650	100
Pine, Yellow	28.8 to 33	12,600 to 19,200	5,400 to 9,500	98 to 170
Spruce	....	10,000 to 19,500	5,050 to 7,850	86 to 110
Walnut, Black	42	9,286 to 16,000	7,500	....

table should be taken with caution. The range of variation in apt to be much greater than the figures indicate. See Johnson's -leaf pine, and Lanza's on spruce, above. The weight of yellow ble is much less than that given by Johnson. (W. K.)

ssive Strengths of American Woods, *when slowly y seasoned.*—Approximate averages, deduced from many exper- e with the U. S. Government testing-machine at Watertown, . S. P. Sharpless, for the Census of 1880. *Seasoned woods* resist ch better than green ones; in many cases, twice as well. Differ- s of the same wood vary greatly. The strengths may readily h as one-third part more or less from the average.

	End-wise,* lbs. per sq. in.	Side-wise,† lbs. per sq. in.	
		.01	.1
white	6800	1300	3000
....	4400	800	1400
....	7000	1100	1900
......	8000	1300	2600
....	4400	600	1400
....	5400	700	1600
more)	6000	1300	2600
....	6000	700	1000
arbor-			
.....	4400	500	900
.bean)	5000	700	1300
......	8000	1700	2600
.... .	5300	900	1600
Ky...	5200	1300	2600
l.	6000	500	1200
white	6800	1300	2600
... ...	7700	1300	2600
. .....	5300	600	1100
....	8000	2000	4000
.....	10000	1600	13000
rican.	5000	500	900
ellow.	9800	1900	4400
....	7000	1600	2600
... ...	9000	1700	5300
d, Ore.	5300	1400	2600

	End-wise,* lbs. per sq. in.	Side-wise,† lbs. per sq. in.	
		.01	.1
Maple:			
sugar and black..	8000	1900	4300
white and red....	6800	1300	2900
Oak:			
white, post (or iron), swamp white, red, and black...	7000	1600	4000
scrub and basket.	6000	1700	4200
chestnut and live	7500	1600	4500
pin..............	6500	1300	3000
Pine:			
white.	5400	600	1200
red or Norway....	6300	600	1400
pitch and Jersey scrub...........	5000	1000	2000
Georgia....	8500	1300	2600
Poplar.............	5000	600	1100
Sassafras...	5000	1300	2100
Spruce, black... ...	5700	700	1300
" white... ...	4500	600	1200
Sycamore (button-wood)...........	6000	1300	2600
Walnut:			
black.	8000	1300	2600
white (butternut).	5400	700	1600
Willow	4400	700	1400

s 1.57 ins. square × 12.6 ins. long.
s 1.57 ins. square × 6.3 ins. long. Pressure applied at mid-length covering one-fourth of the length. The first column gives the ing an indentation of .01 inch, the second those producing an f .1 inch. (See also page 306).

n of Timber Due to the Absorption of Water.

(De Volson Wood, A. S. M. E., vol. x.)

5 in., of pine, oak, and chestnut, were dried thoroughly, and ed in water for 37 days.
per cent of elongation and lateral expansion were:

	Pine.	Oak.	Chestnut.
tion, per cent...........	0.065	0.085	0.165
expansion, per cent...	2.6	3.5	3.65

on of Wood by Heat.—Trautwine gives for the expan *for 1 degree Fahr. 1 part in 440,530*, or for 180 degrees 1 p *one-third of the expansion of iron.*

Shearing Strength of American Woods, adapt Pins or Treenails.

J. C. Trautwine (*Jour. Franklin Inst.*). (Shearing across the

	per sq. in.	
Ash	6280	Hickory
Beech	5223	"
Birch	5595	Maple
Cedar (white)	1372	Oak
" "	1519	Oak (live)
Cedar (Central American)	3410	Pine (white)
Cherry	2945	Pine (Northern yellow
Chestnut	1536	Pine (Southern yellow)
Dogwood	6510	Pine (very resinous yellov
Ebony	7750	Poplar
Gum	5890	Spruce
Hemlock	2750	Walnut (black)
Locust	7176	Walnut (common)

THE STRENGTH OF BRICK, STONE, ET

A great advance has recently been made in the manufacture the direction of increasing their strength. Chas. P. Chase, in *E News*, says: "Taking the tests as given in standard engineering or ten years ago, we find in Trautwine the strength of brick give 4200 lbs. per sq. in. Now, taking recent tests in experiment Watertown Arsenal, the strength ran from 5000 to 22,000 lbs per the tests on Illinois paving-brick, by Prof. I. O. Baker, we find strength in hard paving brick of over 5000 lbs. per square inch. T crushing strength of ten varieties of paving-brick much used in find to be 7150 lbs. to the square inch."

A recent test of brick made by the dry-clay process at Waterlow according to *Paving*, showed an average compressive strength per sq. in. In one instance it reached 4973 lbs. per sq. in. A test at the same place on a "fancy pressed brick." The first crack at a pressure of 305,000 lbs., and the brick crushed at 364,300 lbs lbs. per sq. in. This indicates almost as great compressive st granite paving-blocks, which is from 12,000 to 20,000 lbs. per sq. in

The following notes on bricks are from Trautwine's *Enginee book:*

Strength of Brick.—40 to 300 tons per sq. ft., 622 to 4668 lbs A soft brick will crush under 450 to 600 lbs. per sq. in., or 30 to 4 square foot, but a first-rate machine-pressed brick will stand 200 per sq. ft. (3112 to 6224 lbs. per sq. in.).

Weight of Bricks.—Per cubic foot, best pressed brick, 150 pressed brick, 131 lbs.; common hard brick, 125 lbs.; good comm 118 lbs.; soft inferior brick, 100 lbs.

Absorption of Water.—A brick will in a few minutes ab ¾ lb. of water, the last being 1/7 of the weight of a hand-moulded of its bulk.

Tests of Bricks, full size, on flat side. (Tests made town Arsenal in 1883.)—The bricks were tested between flat steel Compressed surfaces (the largest surface) ground approximately bricks were all about 2 to 2.1 inches thick, 7.5 to 8.1 inches long, 3.76 inches wide. Crushing strength per square inch: One lot ra 11,056 to 16,734 lbs.; a second, 12,995 to 22,351; a third, 10,390 to 12,7 tests gave results from 5960 to 10,250 lbs. per sq. in.

Crushing Strength of Masonry Materials. (Fro "Retaining-Walls.")

	tons per sq. ft.		tons
Brick, best pressed	40 to 300	Limestones and marbles	[illegible]
Chalk	20 to 30	Sandstone	[illegible]
Granite	300 to 1200	Soapstone	[illegible]

Strength of Granite.—The crushing strength of granite is 0 to 15,000 lbs. per sq. in., when tested in two-inch t and toughest of the commonly used varietie ,000 lbs. Samples of granite from a quarry iver, tested at the Watertown Arsenal, have shown a strength of per sq. in. (*Engineering News*, Jan. 12, 1893).

th of Avondale, Pa., Limestone—(*Engineering News*, 3).—Crushing strength of 2 in. cubes: light stone 12,112, gray stone per sq. in.

rse test of lintels, tool-dressed, 42 in. between knife-edge bear- with knife-edge brought upon the middle between bearings:

e, section 6 in. wide × 10 in. high, broke under a load of	20,950 lbs.
dulus of rupture	2,200 "
e, section 8¼ in. wide × 10 in. high, broke under	14,720 "
dulus of rupture	1,170 "
a.—Gray stone	.051 of 1%
Light stone	.052 of 1%

Transverse Strength of Flagging.

(N. J. Steel & Iron Co.'s Book.)

EXPERIMENTS MADE BY R. G. HATFIELD AND OTHERS.

h of the stone in inches; d = its thickness in inches; l = distance arings in inches.

king loads in tons of 2000 lbs., for a weight placed at the *centre* e, will be as follows:

	$\frac{bd^2}{l} \times$		$\frac{bd^2}{l} \times$
flagging	.744	Dorchester freestone	.264
nite	.624	Aubigny freestone	.216
s freestone	.576	Caen freestone	.144
N. J., freestone	.480	Glass	1.000
other quarry)	.432	Slate	1.2 to 2.7
t freestone	.312		

lock of Quincy granite 80 inches wide and 6 inches thick, resting 6 inches in the clear, would be broken by a load resting midway

$$\text{e beams} = \frac{80 \times 36}{36} \times .624 = 49.92 \text{ tons.}$$

ENGTH OF LIME AND CEMENT MORTAR.

(*Engineering*, October 2, 1891.)

de at the University of Illinois on the effects of adding cement to r. In all the tests a good quality of ordinary fat lime was used. two days in an earthenware jar, adding two parts by weight of e of lime, the loss by evaporation being made up by fresh addi- ter. The cements used were a German Portland, Black Diamond), and Rosendale. As regards fineness of grinding, 85 per cent of d passed through a No. 100 sieve, as did 72 per cent of the Rosen- irly sharp sand, thoroughly washed and dried, passing through a e and caught on a No. 30, was used. The mortar in all cases con- wo volumes of sand to one of lime paste. The following results ned on adding various percentages of cement to the mortar:

Tensile Strength, pounds per square inch.

Age	4 Days.	7 Days.	14 Days.	21 Days.	28 Days.	50 Days.	84 Days.
ar	4	8	10	13	18	21	26
t Rosendale	5	8½	9½	12	17	17	18
Portland	5	8½	14	20	25	24	26
Rosendale	7	11	13	18½	21	22½	23
Portland	8	16	18	22	25	28	27
Rosendale	10	12	16½	21½	22½	24	36
Portland	27	39	38	43	47	59	57
Rosendale	9	*13*	*20*	*16*	22	22½	23
Portland	*45*	*58*	*55*	68	67	102	78
Rosendale	*12*	*18½*	22½	27	29	31½	33
Portland	*87*	*91*	*103*	124	94	210	145
Rosendale	*18*	*23*	26	31	34	46	48
Portland	*90*	*120*	*146*	152	181	205	202

MODULI OF ELASTICITY OF VARIOUS MATE

The modulus of elasticity determined from a tensile test of a b material is the quotient obtained by dividing the tensile stress in p square inch at any point of the test by the elongation per inch produced by that stress; or if P = pounds of stress applied, K = tional area, l = length of the portion of the bar in which the ment is made, and λ = the elongation in that length, the mo elasticity $E = \frac{P}{K} \div \frac{\lambda}{l} = \frac{Pl}{K\lambda}$. The modulus is generally measured w elastic limit only, in materials that have a well-defined elastic limit iron and steel, and when not otherwise stated the modulus is unde be the modulus within the elastic limit. Within this limit, for such the modulus is practically constant for any given bar, the elongat directly proportional to the stress. In other materials, such as which have no well-defined elastic limit, the elongations from the b of a test increase in a greater ratio than the stresses, and the m therefore at its maximum near the beginning of the test, and co decreases. The moduli of elasticity of various materials have alre given above in treating of these materials, but the following ta some additional values selected from different sources:

Brass, cast..	9,170,000
" wire.	14,230,000
Copper	15,000,000 to 18,000,000.
Lead.	1,000,000
Tin, cast	4,600,000
Iron, cast	12,000,000 to 27,000,000 (?)
Iron, wrought	22,000,000 to 29,000,000
Steel	26,000,000 to 32,000,000
Marble.	25,000,000
Slate.	14,500,000
Glass	8,000,000
Ash	1,600,000
Beech	1,300,000
Birch	1,250,000 to 1,500,000
Fir	869,000 to 2,191,000
Oak	974,000 to 2,283,000
Teak..	2,414,000
Walnut	306,000
Pine, long-leaf (butt-logs)	1,119,000 to 3,117,000 Avge. 1,9

The maximum figures given by many writers for iron and st 40,000,000 and 42,000,000, are undoubtedly erroneous.

Prof. J. B. Johnson, in his report on Long-leaf Pine, 1893, say modulus of elasticity is the most constant and reliable propert engineering materials. The wide range of value of the modulus of e of the various metals found in public records must be explained neous methods of testing."

In a tensile test of cast iron by the author (Van Nostrand's Scienc No. 41, page 45), in which the ultimate strength was 23,285 lbs. per the measurements of elongation were made to .0001 inch, and the of elasticity was found to decrease from the beginning of the follows: At 1000 lbs. per sq. in., 25,000,000; at 2000 lbs., 16,666,000; lbs., 15,384,000; at 6000 lbs., 13,636,000; at 8000 lbs., 12,500,000; at 1 11,250,000; at 15,000 lbs., 10,000,000; at 20,000 lbs., 8,000,000; at 2 6,140,000. The modulus of elasticity of steel (within the elastic remarkably constant, notwithstanding great variations in chemical temper, etc. It rarely is found below 28,000,000 or above 31,000,0 generally taken at 30,000,000 in engineering calculations.

FACTORS OF SAFETY.

A factor of safety is the ratio in which the load that is just suff overcome instantly the strength of a piece of material is greater t greatest safe ordinary working load. (Rankine.)

Rankine gives the following "examples of the values of thos which occur in machines":

	Dead Load.	Live Load, Greatest.	Live Load, Mean.
Iron and steel	3	6	from 6 to 4
[illegible]	4 to 5	8 to 10	....
[illegible]	4	8	....

factor of safety, 40, is for shafts in millwork which transmit
e efforts.

es the following "factors of safety which have been adopted in
s for different materials." They "include an allowance for
tingencies."

	Dead Load.	Live Load. In Temporary Structures.	Live Load. In Permanent Structures.	Live Load. In Structures subj. to Shocks.
n and steel.	3	4	4 to 5	10
....	3	4	5	10
....	....	4	10	....
....	....	....	6	....
....	20	....	20 to 30	....

s says that "these numbers fairly represent practice based on
many actual cases, but they are not very trustworthy."

d in his "Resistance of Materials" says: "In regard to the
should be left for safety, much depends upon the character of
If the load is simply a dead weight, the margin may be com-
nall; but if the structure is to be subjected to percussive forces
he margin should be comparatively large on account of the
e effect produced by the force. In machines which are sub-
onstant jar while in use, it is very difficult to determine the
in which is consistent with economy and safety. Indeed, in
economy as well as safety generally consists in making them
trong, as a single breakage may cost much more than the extra
essary to fully insure safety."

sion of the resistance of materials to repeated stresses and
pages 238 to 240.

using factors of safety it is becoming customary in designing
in number of pounds per square inch as the maximum stress
e allowed on a piece. Thus, in designing a boiler, instead of
ctor of safety of 6 for the plates and 10 for the stay-bolts, the
sile strength of the steel being from 50,000 to 60,000 lbs. per sq. in.,
working stress of 10,000 lbs. per sq. in. on the plates and 6000
in. on the stay-bolts may be specified instead. So also in
formula for columns (see page 260) the dimensions of a column
d after assuming a maximum allowable compressive stress per
on the concave side of the column.

s for masonry under dead load as given by Rankine and by Unwin,
show a remarkable difference, which may possibly be explained
If the actual crushing strength of a pier of masonry is known
experiment, then a factor of safety of 4 is sufficient for a pier of
e and quality under a steady load; but if the crushing strength
sumed from figures given by the authorities (such as the crush-
of pressed brick, quoted above from Howe's Retaining Walls, 40
per square foot, average 170 tons), then a factor of safety of 20
too great. In this case the factor of safety is really a "factor
."

on of the proper factor of safety or the proper maximum unit
y given case is a matter to be largely determined by the judg-
engineer and by experience. No definite rules can be given.
ary or advisable factors in many particular cases will be found
cases are considered throughout this book. In general the
rcumstances are to be taken into account in the selection of

he ultimate strength of the material is known within narrow
the case of structural steel when tests of samples have been
the load is entirely a steady one of a known amount, and there
to fear the deterioration of the metal by corrosion, the lowest
should be adopted is 3.

he circumstances of 1 are modified by a portion of the load being
in floors of warehouses, the factor should be not less than 4.

he whole load, or nearly the whole, is apt to be alternately put
n off, as in suspension rods of floors of bridges, the factor should

he stresses are reversed *in direction from* tension to compres-
ome *bridge diagonals and parts of* machines, the factor should
han 6.

5. When the piece is subjected to repeated shocks, the factor sh not less than 10.

6. When the piece is subject to deterioration from corrosion the should be sufficiently increased to allow for a definite amount of co before the piece be so far weakened by it as to require removal.

7. When the strength of the material, or the amount of the load, are uncertain, the factor should be increased by an allowance suffi cover the amount of the uncertainty.

8. When the strains are of a complex character and of uncertain such as those in the crank-shaft of a reversing engine, a very high f necessary, possibly even as high as 40, the figure given by Rankine fo in millwork.

THE MECHANICAL PROPERTIES OF CORK

Cork possesses qualities which distinguish it from all other solid bodies, namely, its power of altering its volume in a very marked de consequence of change of pressure. It consists, practically, of an a tion of minute air-vessels, having thin, water-tight, and very stron and hence, if compressed, the resistance to compression rises in a more like the resistance of gases than the resistance of an elastic so as a spring. In a spring the pressure increases in proportion to tance to which the spring is compressed, but with gases the pres creases in a much more rapid manner; that is, inversely as the which the gas is made to occupy. But from the permeability of air, it is evident that, if subjected to pressure in one direction only gradually part with its occluded air by effusion, that is, by its through the porous walls of the cells in which it is contained. The part of cork constitutes 53% of its bulk. Its elasticity has not only considerable range, but it is very persistent. Thus in the better kind used in bottling the corks expand the instant they escape from the This expansion may amount to an increase of volume of 75%, even a corks have been kept in a state of compression in the bottles for te If the cork be steeped in hot water, the volume continues to incr it attains nearly three times that which it occupied in the neck of the

When cork is subjected to pressure a certain amount of permanen mation or "permanent set" takes place very quickly. This pro common to all solid elastic substances when strained beyond their limits, but with cork the limits are comparatively low. Besides the nent set, there is a certain amount of sluggish elasticity—that is, being released from pressure springs back a certain amount at o the complete recovery takes an appreciable time.

Cork which had been compressed and released in water many t times had not changed its molecular structure in the least, and had ued perfectly serviceable. Cork which has been kept under a pre three atmospheres for many weeks appears to have shrunk to fro 85% of its original volume.—*Van Nostrand's Eng'g Mag.* 1886, xxx

TESTS OF VULCANIZED INDIA-RUBBER.

Lieutenant L. Vladomiroff, a Russian naval officer, has recently out a series of tests at the St. Petersburg Technical Institute with a establishing rules for estimating the quality of vulcanized india- The following, in brief, are the conclusions arrived at, recourse be to physical properties, since chemical analysis did not give any reli sult: 1. India-rubber should not give the least sign of superficial c when bent to an angle of 180 degrees after five hours of exposure in air-bath to a temperature of 125° C. The test-pieces should be 2 thick. 2. Rubber that does not contain more than half its weight o lic oxides should stretch to five times its length without breaking. ber free from all foreign matter, except the sulphur used in vulcan should stretch to at least seven times its length without rupture extension measured immediately after rupture should not exceed 1 original length, with given dimensions. 5. Suppleness may be dete by measuring the percentage of ash formed in incineration. This m the basis for deciding between different grades of rubber for certa poses. 6. Vulcanized rubber should not harden under cold. The have been adopted for the Russian navy.—*Iron Age*, June 15, 1893.

XYLOLITH, OR WOODSTONE

material invented in 1883, but only lately introduced to the trig & Co., of Pottschappel, near Dresden. It is made of lcined magnesite, mixed with sawdust and saturated with a lloride of calcium. This pasty mass is spread out into sheets l to a pressure of about 1000 lbs. to the square inch, and then n the air. Specific gravity 1.553. The fractured surface shows se grain of a yellow color. It has a tensional resistance when per square inch, and when wet about 66 lbs. When immersed 2 hours it takes up 2.1% of its weight, and 3.8% when immersed

ed for several days with hydrochloric acid it loses 2.3% in hows no loss of weight under boiling in water, brine, soda-lye, f sulphates of iron, of copper, and of ammonium. In hardness tands between feldspar and quartz, and as a non-conductor of between asbestos and cork.

e well, and at a red heat it is rendered brittle and crumbles at t retains its general form and cohesion. This xylolith is sup-ts from ¼ in. to 1½ in. thick, and up to one metre square. It used in Germany for floors in railway stations, hospitals, etc., of vessels. It can be sawed, bored, and shaped with ordinary tools. Putty in the joints and a good coat of paint make it r-proof. It is sold in Germany for flooring at about 7 cents per and the cost of laying adds about 4 cents more.—*Eng'g News*, and July 27, 1893.

IINUM—ITS PROPERTIES AND USES.

lfred E. Hunt, Pres't of the Pittsburgh Reduction Co.)

c gravity of pure aluminum in a cast state is 2.58; in rolled section it is 2 6; in very thin sheets subjected to high com-er chilled rolls, it is as much as 2.7. Taking the weight of a cast aluminum as 1, wrought iron is 2.90 times heavier; struc-95 times; copper, 3.60; ordinary high brass, 3.45. Most wood se in structures has about one third the weight of aluminum, 0.092 lb. to the cubic inch.

num is practically not acted upon by boiling water or steam. de or hydrogen sulphide does not act upon it at any tempera-0° F. It is not acted upon by most organic secretions.

ic acid is the best solvent for aluminum, and strong solutions alies readily dissolve it. Ammonia has a slight solvent action. rated sulphuric acid dissolves aluminum upon heating, with sulphurous acid gas. Dilute sulphuric acid acts but slowly on ough the presence of any chlorides in the solution allow rapid n. Nitric acid, either concentrated or dilute, has very little he metal, and sulphur has no action unless the metal is at a red ater has very little effect on aluminum. Strips of the metal sides of a wooden ship corroded less than 1/1000 inch after six sure to sea-water, corroding less than copper sheets similarly

ility pure aluminum is only exceeded by gold and silver. In tands seventh in the series, being exceeded by gold, silver, m, very soft steel, and copper. Sheets of aluminum have been to a thickness of 0.0005 inch, and beaten into leaf nearly as leaf. The metal is most malleable at a temperature of between F., and at this temperature it can be drawn down between rly as much draught upon it as with heated steel. It has also down into the very finest wire. By the Mannesmann process bes have been made in Germany.

stands very high in the series as an electro-positive metal, and other metals should be avoided, as it would establish a galvanic

cal conductivity of aluminum is only surpassed by pure copper, old. With silver taken at 100 the electrical conductivity of 54.20; that of gold on the same scale is 78; zinc is 29.90; iron is platinum 10.60. Pure aluminum has no polarity, and the market is absolutely non-magnetic.

ings can be made of aluminum in either dry or "green" sand n metal "chills." It must not be heated much beyond its t, and must be poured with care, owing to the ready absorption ases and air. *The shrinkage in cooling is 17/64 inch per fo-ore than ordinary brass. It should be melted in plum- the metal becomes molten at a temperature of 1120° fessor Roberts-Austen, or at 1300° F. according to Ric-*

Nos. 1a and 2 were full of blow-holes.

Tests Nos. 1 and 1a show the variation in cast copper due to va ditions of casting. In the crushing tests Nos. 12 to 20, inclusive, cr broke under the strain, but all the others bulged and flattened out cases the crushing strength is taken to be that which caused a d 10% in the length. The test-pieces were 2 in. long and 5/8 in. diam torsional tests were made in Thurston's torsion machine, on p diameter and 1 in. long between heads.

Specific Gravity of the Copper-tin Alloys.—Th gravity of copper, as found in these tests, is 8.874 (tested in tur the ingot, and reduced to 39.1° F.). The alloy of maximum sp contained 62.42 copper, 37.48 tin, and all the alloys containing les tin varied irregularly in sp. gr. between 8.65 and 8.93, the density not on the composition, but on the porosity of the casting. It i that the actual sp. gr. of all these alloys containing less than 37% t 8.95, and any smaller figure indicates porosity in the specimen.

From 37% to 100% tin, the sp. gr. decreases regularly from the m 8.956 to that of pure tin, 7.293.

Note on the Strength of the Copper-tin Allo

The bars containing from 2% to 24% tin, inclusive, have re strength, and all the rest are practically worthless for purposes strength is required. The dividing line between the strong and br is precisely that at which the color changes from golden yellow white, viz., at a composition containing between 24% and 30% of tin

It appears that the tensile and compressive strengths of these in no way related to each other, that the torsional strength is c portional to the tensile strength, and that the transverse streng pend in some degree upon the compressive strength, but it is n nearly related to the tensile strength. The modulus of rupture, a by the transverse tests, is, in general, a figure between those of t compressive strengths per square inch, but there are a few exc which it is larger than either.

The strengths of the alloys at the copper end of the series incre with the addition of tin till about 4% of tin is reached. The strength continues regularly to increase to the maximum, till the taining about 17½% of tin is reached, while the tensile and strengths also increase, but irregularly, to the same point. This ir is probably due to porosity of the metal, and might possibly be r any means which would make the castings more compact. The is reached at the alloy containing 82.70 copper, 17.34 tin, the strength, however, being very much greater at this point than t or torsional strength. From the point of maximum strength t drop rapidly to the alloys containing about 27.5% of tin, and then m to 37.5%, at which point the minimum (or nearly the minimum) st all three methods of test, is reached. The alloys of minimum st found from 37.5% tin to 52.5% tin. The absolute minimum is prob 45% of tin.

From 52.5% of tin to about 77.5% tin there is a rather slow and ir crease in strength. From 77.5% tin to the end of the series, or strengths slowly and somewhat irregularly decrease.

The results of these tests do not seem to corroborate the theor some writers, that peculiar properties are possessed by the al are compounded of simple multiples of their atomic weights or equivalents, and that these properties are lost as the composi more or less from this definite constitution. It does appear that percentage composition gives a maximum strength and anoth percentage a minimum, but neither of these compositions is repr simple multiples of the atomic weights.

There appears to be a regular law of decrease from the maxim minimum strength which does not seem to have any relation to t proportions, but only to the percentage compositions.

Hardness.—The pieces containing less than 24% of tin were the lathe without difficulty, a gradually increasing hardness bein the last named giving a very short chip, and requiring frequent of the tool.

[illegible] brittle alloys it was found impossible to turn the [illegible] smooth surface. No. 13 to No. 17 (26.85 to 34.47 [illegible] tool at all. Chips would fly off in advance of t th it, leaving a rough surface; or the tool would sometimes, apparently, off portions of the metal, grinding it to powder. Beyond 40% tin the ess decreased so that the bars could be easily turned.

LLOYS OF COPPER AND ZINC. (U. S. Test Board).

ean Composition by Analysis.		Tensile Strength, lbs. per sq. in.	Elastic Limit % of Breaking Load, lbs. per sq. in.	Elongation % in 5 inches.	Transverse Test Modulus of Rupture.	Deflection 1″ sq. bar 22″ long, in.	Crushing Str'gth per sq. in., lbs.	Torsional Tests.	
op-er.	Zinc.							Max. Tors. Moment ft.-lbs.	Angle of Torsion, deg.
7.83	1.88	27,240	....	....	....	.....	....	130	357
2.93	16.98	32,600	26.1	26.7	23,197	Bent	....	155	329
1.91	17.99	32,670	30.6	31.4	21,198	"	....	166	345
7.39	22.45	35,630	20.0	35.5	25,374	"	....	169	311
5.65	23.08	30,520	24.6	35.8	22,325	"	42,000	165	267
3.20	26.47	31,580	23.7	38.5	25,894	"	....	168	293
1.20	28.54	30,510	29.5	29.2	24,468	"	....	164	269
9.74	30.06	28,120	28.7	20.7	26,930	"	....	143	202
5.27	33.50	37,800	25.1	37.7	28,459	"	....	176	2[illegible]7
3.44	36.36	48,300	32.8	[illegible]1.7	43,216	"	....	202	230
0.94	38.65	41,065	40.1	20.7	38,068	"	75,000	194	202
8.49	41.10	50,450	54.4	10.1	63,[illegible]04	"	....	2[illegible]7	93
5.15	44.44	44,280	44.0	15.3	42,463	"	78,000	209	109
4.86	44.78	46,400	53.9	8.0	47,955	"	....	223	72
9.66	50.14	30,900	54.5	5.0	33,467	1.26	117,400	172	38
8.99	50.82	26,050	100.	0.8	40,189	0.61	....	176	16
7.56	52.28	24,150	100.	0.8	48,471	1.17	121,000	155	13
3.36	56.22	9,170	100.	....	17,691	0.10	....	88	2
.30	58.12	3,727	100.	....	7,761	0.04	....	18	2
2.94	66.23	1,774	100.	....	8,296	0.04	....	29	1
9.20	70.17	6,414	100.	....	16,579	0.04	....	40	2
.81	77.63	9,000	100.	0.2	22,972	0.13	52,152	65	1
2.12	86.67	12,413	100.	0.4	35,026	0.31	....	82	3
.35	94.59	18,065	100.	0.5	26,162	0.46	....	81	22
ast	Zinc.	5,400	75.	0.7	7,539	0.12	22,000	37	142

iation in Strength of Gun-bronze, and Means of oving the Strength.—The figures obtained for alloys of from 12.7% tin, viz., from 26,860 to 29,430 pounds, are much less than are given as the strength of gun-metal. Bronze guns are usually cast he pressure of a head of metal, which tends to increase the strength nsity. The strength of the upper part of a gun casting, or sinking s not greater than that of the small bars which have been tested in experiments. The following is an extract from the report of Major concerning the strength and density of gun-bronze (1850):—Extreme on of six samples from different parts of the same gun (a 32-pounder er): Specific gravity, 8.487 to 8.835; tenacity, 26,428 to 52,192. Extreme on of all the samples tested: Specific gravity, 8.308 to 8.850; tenacity, o 54,531. Extreme variation of all the samples from the gun heads: c gravity, 8.308 to 8.756; tenacity, 23,529 to 35,484.

r Wade says: The general results on the quality of bronze as it is in guns are mostly of a negative character. They expose defects in y and strength, develop the heterogeneous texture of the metal in dif- parts of the same gun, and show the irregularity and uncertainty of y which attend the casting of all guns, although made from similar als, treated in like manner.

y ordnance bronze containing 9 parts copper and 1 part tin, tested at ngton, D. C., in 1875–6, showed a variation in tensile strength from to 51,400 lbs. per square inch, in elongation from 3% to 58%, and in spe- ravity from 8.39 to 8.88.

t a *great improvement* may be made in the density and tenacity of *nze by compression has been shown by the experiments of* Mr. S. B. *Boston, Mass., in 1869, and by* those of General Uchatius in Austria *The former increased the* density of the metal next the bore of the *8.321 to 8.875,* and the tenacity from 27,238 to 41,471 pounds per

square inch. The latter, by a similar process, obtained the following fi[gures]
for tenacity:

	Pounds per sq. in.
Bronze with 10% tin	72,053
Bronze with 8% tin	73,958
Bronze with 6% tin	77,656

ALLOYS OF COPPER, TIN, AND ZINC.

(Report of U. S. Test Board, Vol. II, 1881.)

No. in Report.	Analysis, Original Mixture.			Transverse Strength.		Tensile Strength per square inch.		Elongation per cent in 5 inches
	Cu.	Sn.	Zn.	Modulus of Rupture	Deflection, ins.	*A.*	*B.*	*A.*
72	90	5	5	41,334	2.63	23,660	30,740	2.31
5	88.14	1.86	10	31,986	3.67	32,000	33,000	17.6
70	85	5	10	44,457	2.85	28,840	28,560	6.80
71	85	10	5	62,470	2.56	35,680	36,000	2.51
89	85	12.5	2.5	62,405	2.83	34,500	32,800	1.29
88	82.5	12.5	5	60,960	1.61	36,000	34,000	.86
77	82.5	15	2.5	69,045	1.09	33,600	33,800	....
67	80	5	15	42,618	3.88	37,560	32,300	11.6
68	80	10	10	67,117	2.45	32,830	31,950	1.57
69	80	15	5	54,476	.44	32,350	30,760	.55
86	77.5	10	12.5	63,849	1.19	35,500	36,000	1.00
87	77.5	12.5	10	61,705	.71	36,000	32,500	.72
63	75	5	20	55,355	2.91	33,140	34,960	2.50
85	75	7.5	17.5	62,607	1.39	33,700	39,300	1.56
64	75	10	15	58,345	.73	35,320	34,000	1.13
65	75	15	10	51,109	.31	35,440	28,000	.59
66	75	20	5	40,235	.21	23,140	27,660	.43
83	72.5	7.5	20	51,839	2.86	32,700	34,800	3.73
84	72.5	10	17.5	53,230	.74	30,000	30,000	.48
59	70	5	25	57,349	1.37	38,000	32,940	2.06
82	70	7.5	22.5	48,836	.36	38,000	22,400	.84
60	70	10	20	36,520	.18	33,140	26,300	.31
61	70	15	15	37,924	.20	33,440	27,800	.25
62	70	20	10	15,126	.08	17,000	12,900	.08
81	67.5	2.5	30	58,345	2.91	34,720	45,850	7.27
74	67.5	5	27.5	55,976	.49	34,000	34,460	1.06
75	67.5	7.5	25	46,875	.32	29,500	20,000	.36
80	65	2.5	32.5	56,949	2.36	41,350	38,300	3.26
55	65	5	30	51,369	.56	37,140	36,000	1.21
56	65	10	25	27,075	.14	25,720	22,500	.15
57	65	15	20	13,591	.07	6,820	7,231	....
58	65	20	15	11,932	.05	3,765	2,665	....
79	62.5	2.5	35	69,255	2.34	44,400	45,000	2.15
78	60	2.5	37.5	69,508	1.46	57,400	52,900	4.87
52	60	5	35	46,076	.28	41,160	38,330	.39
53	60	10	30	24,699	.13	21,780	21,240	.15
54	60	15	25	18,248	.09	18,020	12,400	....
12	58.22	2.30	39.48	95,623	1.99	66,500	67,600	3.13
3	58.75	8.75	32.5	35,752	.18	Broke	before t	est; very
4	57.5	21.25	21.25	2,752	.02	725	1,300	....
73	55	0.5	44.5	72,308	3.05	68,900	68,900	9.43
50	55	5	40	38,174	.22	27,400	30,500	.46
51	55	10	35	28,258	.14	25,460	18,500	.29
49	50	5	45	20,814	.11	23,000	31,300	.66

The transverse tests were made in bars 1 in. square, 22 in. betwe[en]
...s. The tensile tests were made on bars 0.798 in. diam., turned fr[om]
...lves of the transverse-test bar, one half being marked *A* a[nd]

t **Bronzes.**—The usual composition of ancient bronze was the
t of modern gun-metal—90 copper, 10 tin; but the proportion of
rom 5% to 15%, and in some cases lead has been found. Some an-
ian tools contained 88 copper, 12 tin.

h **of the Copper-zinc Alloys.**—The alloys containing less
zinc by original mixture were generally defective. The bars
blow-holes, and the metal showed signs of oxidation. To insure
gs it appears that copper-zinc alloys should contain more than

. 2 to No. 8 inclusive, 16.98 to 30.06% zinc the bars show a remark-
rity in all their properties. They have all nearly the same
d ductility, the latter decreasing slightly as zinc increases, and
alike in color and appearance. Between Nos. 8 and 10, 30.06 and
the strength by all methods of test rapidly increases. Between
No. 15, 36.36 and 50.14% zinc, there is another group, distinguished
ength and diminished ductility. The alloy of maximum tensile,
and torsional strength contains about 41% of zinc.

s containing less than 55% of zinc are all yellow metals. Beyond
r changes to white, and the alloy becomes weak and brittle. Be-
nd pure zinc the color is bluish gray, the brittleness decreases
ngth increases, but not to such a degree as to make them useful
ctive purposes.

nce between Composition by Mixture and by
.—There is in every case a smaller percentage of zinc in the
alysis than in the original mixture, and a larger percentage of
e loss of zinc is variable, but in general averages from 1 to 2%.

on or Separation of the Metals.—In several of the
iderable amount of liquation took place, analysis showing a
n composition of the two ends of the bar. In such cases the
omposition was gradual from one end of the bar to the other,
nd in general containing the higher percentage of copper. A
ance was bar No. 13, in the above table, turnings from the upper
ing 40.36% of zinc, and from the lower end 48.52%.

Gravity.—The specific gravity follows a definite law, varying
mposition, and decreasing with the addition of zinc. From the
ve of specific gravities the following mean values are taken:

c.........	0	10	20	30	40	50	60	70	80	90	100.
vity.......	8.80	8.72	8.60	8.40	8.36	8.20	8.00	7.72	7.40	7.20	7.14.

Representation of the Law of Variation of of Copper-Tin-Zinc Alloys.—In an equilateral triangle
he perpendicular distances from any point within it to the three
l to the altitude. Such a triangle can therefore be used to
cally the percentage composition of any compound of three
s a triple alloy. Let one side represent 0 copper, a second
e third 0 zinc, the vertex opposite each of these sides repre-
of each element respectively. On points in a triangle of wood
different alloys tested, wires were erected of lengths propor-
tensile strengths, and the triangle then built up with plaster to
f the wires. The surface thus formed has a characteristic
representing the variations of strength with variations of
. The cut shows the surface thus made. The vertical section
presents the law of tensile strength of the copper-tin alloys,
he right that of tin-zinc alloys, and the one at the rear that of
inc alloys. The high point represents the strongest possible
three metals. Its composition is copper 55, zinc 43, tin 2, and its
ut 70,000 lbs. The high ridge from this point to the point of
eight of the section on the left is the line of the strongest alloys,
by the formula zinc + (3 × tin) = 55.

ying to the rear of the ridge, containing more copper and less
e alloys of greater ductility than those on the line of maximum
l are the valuable commercial alloys; those in front on the decliv-
e central valley are brittle, and those in the valley are both brit-
. Passing from the valley toward the section at the right the
eir brittleness and become soft, the maximum softness being
but they remain weak, as is shown by the low elevation of
s model was planned and constructed by Prof. Thurst
rans. A. S. C. E. 1881, Report of the U. S. Board appoin

test Iron, Steel, etc., vol. ii., Washington, 1881, and Thurston's M[illegible] of Engineering, vol. iii.)

The best alloy obtained in Thurston's research for the U. S. Testi[illegible] has the composition, Copper 55, Tin 0.5, Zinc 44.5. The tensile stre[illegible] cast bar was 68,900 lbs. per sq. in., two specimens giving the same re[illegible] elongation was 47 to 51 per cent in 5 inches. Thurston's formula for tin-zinc alloys of maximum strength (Trans. A. S. C. E., 1881) is $z +$ [illegible]

FIG. 77.

in which z is the percentage of zinc and t that of tin. Alloys prop[illegible] according to this formula should have a strength of about 40,[illegible] per sq. in. + 500z. The formula fails with alloys containing less th[illegible] cent of tin.

The following would be the percentage composition of a number [illegible] made according to this formula, and their corresponding tensile str[illegible] castings:

Tin.	Zinc.	Copper.	Tensile Strength, lbs. per sq. in.	Tin.	Zinc.	Copper.	T[illegible]
1	52	47	66,000	8	31	61	[illegible]
2	49	49	64,500	9	28	63	[illegible]
3	46	51	63,000	10	25	65	[illegible]
4	43	53	61,500	12	19	69	[illegible]
5	40	55	60,000	14	13	73	[illegible]
6	37	57	58,500	16	7	77	[illegible]
7	34	59	57,000	18	1	81	[illegible]

These alloys, while possessing maximum tensile strength, would in[illegible] be too hard for easy working by machine tools. Another series [illegible] the formula $z + 4t = 50$ would have greater ductility, together w[illegible] [illegible], as follows, the strength being calculated as

[illegible]bs. per sq. in. = 40,000 + 500z.

Tin.	Zinc.	Copper.	Tensile Strength, lbs. per sq. in.	Tin.	Zinc.	Copper.	Tensile Strength, lbs. per sq. in.
1	46	53	63,000	7	22	71	51,000
2	42	56	61,000	8	18	74	49,000
3	38	59	59,000	9	14	77	47,000
4	34	62	57,000	10	10	80	45,000
5	30	65	55,000	11	6	83	43,000
6	26	68	53,000	12	2	86	41,000

Composition of Alloys in Every-day Use in Brass Foundries. (*American Machinist.*)

	Copper.	Zinc.	Tin.	Lead.	
	lbs.	lbs.	lbs.	lbs.	
Admiralty metal	87	5	8	......	For parts of engines on board naval vessels.
Bell metal....	16	......	4	......	Bells for ships and factories.
Brass (yellow).....	16	8	......	½	For plumbers, ship and house brass work.
Bush metal......	64	8	4	4	For bearing bushes for shafting
Gun metal.........	32	1	3	......	For pumps and other hydraulic purposes.
Steam metal.......	20	1	1½	1	Castings subjected to steam pressure.
Hard gun metal...	16	......	2½	......	For heavy bearings.
Muntz metal.......	60	40	......	......	Metal from which bolts and nuts are forged, valve spindles, etc.
Phosphor bronze..	92	......	8 phos. tin		For valves, pumps and general work.
" " ..	90	.. ...	10 "	"	For cog and worm wheels bushes, axle bearings, slide valves, etc.
Brazing metal......	16	3	......	......	Flanges for copper pipes.
" solder....	50	50	......	......	Solder for the above flanges.

Gurley's Bronze.—16 parts copper, 1 tin, 1 zinc, ½ lead, used by W. & L. E. Gurley of Troy for the framework of their engineer's transits. Tensile strength 41,114 lbs. per sq. in., elongation 27% in 1 inch, sp. gr. 8.696 (W. J. Keep, Trans. A. I. M. E. 1890.)

Useful Alloys of Copper, Tin, and Zinc.

(Selected from numerous sources.)

	Copper.	Tin.	Zinc.	
U. S. Navy Dept. journal boxes and guide-gibs =	6	1	¼	parts.
	82.8	13.8	3.4	per cent.
Tobin bronze......................	58.22	2.30	39.48	" "
Naval brass......................	62	1	37	" "
Composition, U. S. Navy............	88	10	2	" "
Brass bearings (J. Rose)............	64	8	1	parts.
	87.7	11.0	1.3	per cent.
Gun metal.........................	92.5	5	2.5	" "
" "	91	7	2	" "
" "	87.75	9.75	2.5	" "
" "	85	5	10	" "
" "	83	2	15	" "
Tough brass for engines............	13	2	2	parts.
	76.5	11.8	11.7	per cent.
Bronze for rod-boxes (Lafond).......	82	16	2	slightly malleable
" " pieces subject to shock..	83	15	1.50	0.50 lead.
Red brass .. .. parts	20	1	1	1 "
" ....per cent	87	4.4	4.3	4.3 "
Bronze for pump casings (Lafond)...	88	10	2	
" " eccentric straps. "	84	14	2	
" " *shrill whistles*............	*80*	*18*	....	2.0 antimony.
" " *low-toned whistles*.......	*81*	*17*	....	2.0 "

	Copper.	Tin.	Zinc.	
Art bronze, dull red fracture........	97	2	1	
Gold bronze........................	89.5	2.1	5.6	2.8 lead.
Bearing metal	89	8	3	
" "	89	2½	8½	
" "	86	14	...	
" "	85¼	12¾	2	
" "	80	18	2	
" "	79	18	2½	½ lead.
" "	74	9½	9½	7 lead.
English brass of A.D. 1504............	64	3	29½	3½ lead.

Copper-Nickel Alloys, German Silver.

	Copper.	Nickel.	Tin.	Z
German silver......	51.6	25.8	22.6	2
" "	50.2	14.8	3.1	..
" "	51.1	13.8	3.2	
" "	52 to 55	18 to 25	......	20
Nickel "	75 to 66	25 to 33	......	..

A refined copper-nickel alloy containing 50% copper and 49% nickel, very small amounts of iron, silicon and carbon, is produced direct Bessemer matte in the Sudbury (Canada) Nickel Works. German manufacturers purchase a ready-made alloy, which melts at a low hea requires simple addition of zinc, instead of buying the nickel and co separately. This alloy, "50-50" as it is called, is almost indistinguis from pure nickel. Its cost is less than nickel, its melting point much l it can be cast solid in any form desired, and furnishes a casting which easily in the lathe or planer, yielding a silvery white surface unchange air or moisture. For bullet casings now used in various British and nental rifles, a special alloy of 80% copper and 20% nickel is made.

Special Alloys. (*Engineer*, March 24, 1893.)

JAPANESE ALLOYS for art work:

	Copper.	Silver.	Gold.	Lead.	Zinc.	Ir
Shaku-do......	94.50	1.55	3.73	0.11	trace.	tra
Shibu-ichi.	67.31	32.07	traces.	.52		

GILBERT'S ALLOY for *cera-perduta* process, for casting in plaster-of-p

Copper 91.4 Tin 5.7 Lead 2.9 Very fusible.

COPPER-ZINC-IRON ALLOYS.

(F. L. Garrison, *Jour. Frank. Inst.*, June and July, 1891.)

Delta Metal.—This alloy, which was formerly known as *sterro-m* is composed of about 60 copper, from 34 to 44 zinc, 2 to 4 iron, and 1 to The peculiarity of all these alloys is the content of iron, which appea have the property of increasing their strength to an unusual degree making delta metal the iron is previously alloyed with zinc in known definite proportions. When ordinary wrought-iron is introduced molten zinc, the latter readily dissolves or absorbs the former, and will it up to the extent of about 5% or more. By adding the zinc-iron alloy obtained to the requisite amount of copper, it is possible to introduce definite quantity of iron up to 5% into the copper alloy. Garrison give following as the range of composition of copper-zinc-iron, and copper tin-iron alloys:

I.	Per cent.	II.	Per
Iron....................	0.1 to 5	Iron..	0.1
Copper....	50 to 65	Tin.......	0.1
Zinc.....................	49.9 to 30	Zinc.	1
		Copper..................	[illegible]

The advantages claimed for delta metal are great strength and tough It produces sound castings of close grain. It can be rolled and forged and can [illegible] certain amount of drawing and hammering when cold [illegible] and when exposed to the atmosphere tarnishes less

When cast in sand delta metal has a tensile strength of about 45,000 pounds er square inch, and about 10% elongation; when rolled, tensile strength of 000 to 75,000 pounds per square inch, elongation from 9% to 17% on bars 1.128 ch in diameter and 1 inch area.

Wallace gives the ultimate tensile strength 33,600 to 51,520 pounds per uare inch, with from 10% to 20% elongation.

elta metal can be forged, stamped and rolled hot. It must be forged at dark cherry-red heat, and care taken to avoid striking when at a black at.

According to Lloyd's Proving House tests, made at Cardiff, December 20, , a half-inch delta metal-rolled bar gave a tensile strength of 88,400 unds per square inch, with an elongation of 30% in three inches.

Tobin Bronze.—This alloy is practically a sterro or delta metal with e addition of a small amount of lead, which tends to render copper softer d more ductile.

The following analyses of Tobin bronze were made by Dr. Chas. B. Dudley:

	Pig Metal, per cent.	Test Bar (Rolled). per cent.
Copper	59.00	61.20
Zinc	38.40	37.14
Tin	2.16	0.90
Iron	0.11	0.18
Lead	0.31	0.35

Dr. Dudley writes, "We tested the test bars and found 78,500 tensile rength with 15% elongation in two inches, and 40½% in eight inches. This gh tensile strength can only be obtained when the metal is manipulated. ch high results could hardly be expected with cast metal."

The original Tobin bronze in 1875, as described by Thurston, Trans. . S. C. E 1881, had, composition of copper 58.22, tin 2.30, zinc 39.48. As st it had a tenacity of 66,000 lbs. per sq. in., and as rolled 79,000 lbs.; cold lled it gave 104,000 lbs.

A circular of Ansonia Brass & Copper Co. gives the following:—The tensile rength of six Tobin bronze one-inch round rolled rods, turned down to a ameter of ⅝ of an inch, tested by Fairbanks, averaged 79,600 lbs. per sq. ., and the elastic limit obtained on three specimens averaged 54,257 lbs. per . in.

At a cherry-red heat Tobin bronze can be forged and stamped as readily steel. Bolts and nuts can be forged from it, either by hand or by ma- inery, with a marked degree of economy. Its great tensile strength, and sistance to the corrosive action of sea-water, render it a most suitable etal for condenser plates, steam-launch shafting, ship sheathing and stenings, nails, hull plates for steam yachts, torpedo and life boats, and ip deck fittings.

The Navy Department has specified its use for certain purposes in the achinery of the new cruisers. Its specific gravity is 8.071. The weight of cubic inch is .291 lb.

PHOSPHOR-BRONZE AND OTHER SPECIAL BRONZES.

Phosphor-bronze.—In the year 1868, Montefiore & Kunzel of Liège, lgium, found by adding small proportions of phosphorus or "phosphoret tin or copper" to copper that the oxides of that metal, nearly always resent as an impurity, more or less, were deoxidized and the copper much proved in strength and ductility, the grain of the fracture became finer, e color brighter, and a greater fluidity was attained.

Three samples of phosphor-bronze tested by Kirkaldy gave:

Elastic limit, lbs. per sq. in	23,800	24,700	16,100
Tensile strength, lbs. per sq. in.	52,625	46,100	44,448
Elongation, per cent.	8.40	1.50	33.40

The strength of phosphor-bronze varies like that of ordinary bronze cording to the percentages of copper, tin, zinc, lead, etc., in the alloy.

Deoxidized Bronze.—This alloy resembles phosphor bronze some- what in composition and also delta metal, in containing zinc and iron. The llowing analysis gives its average composition:

Copper	82.67	Iron	0.10
Tin	12.40	Silver	0.07
Zinc	3.23	Phosphorus	0.005
Lead	2.14		
			100.615

Comparison of Copper, Silicon-bronze, and Pho bronze Wires.

(*Engineering*, Nov. 23, 1883.)

Description of Wire.	Tensile Strength per square inch in		Rel Condu
	Tons.	Lbs.	
Pure copper..........................	17.78	39,827	100 pe
Silicon bronze (telegraph)...........	18.27	41,696	96
" " (telephone),...........	48.25	108,080	34
Phosphor Bronze (telephone).........	45.71	102,390	26

ALUMINUM ALLOYS.

(**Aluminum Bronze.** Cowles Electric Smelting and Al. Co.'s

The standard A No. 2 grade of aluminum bronze, containing 11% num and 90% of copper, has many remarkable characteristics w tinguish it from all other metals.

The tenacity of castings of A No. 2 grade metal varies betwe and 90,000 lbs. to the square inch, with from 4% to 14% elongation.

Increasing the proportion of aluminum in bronze beyond 110 pr brittle alloy; therefore nothing higher than the A No. 1, which cont is made.

The B, C, D, and E grades, containing 7½%, 5%, 2½%, and 1¼% of al respectively, decrease in tenacity in the order named, that of the being about 65,000 pounds, while the latter is 25,000 pounds. While also a proportionate decrease in transverse and torsional strength limit, and resistance to compression as the percentage of aluminum ered and that of copper raised, the ductility on the other hand incr the same proportion. The specific gravity of the A No. 1 grade is 7

Bell Bros., Newcastle, gave the specific gravity of the aluminum as below:

3% aluminum	....	8.691
4% "	....	8.621
5% "	....	8.369
10% "	....	7.689

Casting.—The melting point of aluminum bronze varies sligh the amount of aluminum contained, the higher grades melting at what lower temperature than the lower grades. The A No. 1 grad at about 1700° F., a little higher than ordinary bronze or brass.

Aluminum bronze shrinks more than ordinary brass. As the meta fies rapidly it is necessary to pour it quickly and to make the feeder large, so that there will be no "freezing" in them before the ca properly fed. Baked-sand moulds are preferable to green sand, ex small castings, and when fine skin colors are desired in the casting paper by Thos. D. West, Trans. A. S. M. E. 1886, vol. viii.)

All grades of aluminum bronze can be rolled, swedged, spun, o cold except A 1 and A 2. They can all be worked at a bright red he

In rolling, swedging, or spinning cold, it should be annealed very of at a brighter red heat than is used for annealing brass.

Brazing.—Aluminum bronze will braze as well as any other using one quarter brass solder (zinc 500, copper 500 (and three borax, or, better, three quarters cryolite.

Soldering.—To solder aluminum bronze with ordinary soft solder: Cleanse well the parts to be joined free from grease and di place the parts to be soldered in a strong solution of sulphate of co place in the bath a rod of soft iron touching the parts to be joined a while a coppery-like surface will be seen on the metal. Remo bath, rinse quite clean, and brighten the surfaces. These surfaces be tinned by using a fluid consisting of zinc dissolved in hydrochlori the ordinary way, with common soft solder.

Mierzinski recommends ordinary hard solder, and says that Hu *alloy of the* usual half-and-half lead-tin solder, with 12.5%, 25% *amalgam.*

Tests of Aluminum Bronzes.

H. J. Dagger, in a paper read before the British Association, 1889.)

cent of inum.	Tensile Strength.		Elonga-tion, per cent.	Specific Gravity.
	Tons per square inch.	Pounds per square inch.		
....	40 to 45	89,600 to 100,800	8	7.23
....	33 " 40	73,920 " 89,600	14	7.69
....	25 " 30	56,000 " 67,200	40	8.00
....	15 " 18	33,600 " 40,320	40	8.37
....	13 " 15	29,120 " 33,600	50	8.69
....	11 " 13	24,640 " 29,120	55	...

ysical and chemical tests made of samples cut from various sec- ½%, 5%, 7½%, or 10% aluminized copper castings tend to prove that inum unites itself with each particle of copper with uniform pro- each case, so that we have a product that is free from liquation y homogeneous. (R. C. Cole, *Iron Age*, Jan. 16, 1890.)

inum-Brass (E. H. Cowles, Trans. A. I. M. E., vol. xviii.)— luminum-brass is made by fusing together equal weights of A 1 n-bronze, copper, and zinc. The copper and bronze are first thor- elted and mixed, and the zinc is finally added. The material is left urnace until small test-bars are taken from it and broken. When rs show a tensile strength of 80,000 pounds or over, with 2 or 3 per tility, the metal is ready to be poured. Tests of this brass, on small ve at times shown as high as 100,000 pounds tensile strength.

rew of the United States gunboat Petrel is cast from this brass, ith a trifle less zinc in order to increase its ductility.

Tests of Aluminum-Brass.

(Cowles E. S. & Al. Co.)

es (Castings.)	Diameter of Piece, Inch.	Area, sq. in.	Tensile Strength, lbs. per sq. in.	Elastic Limit, lbs. per sq. in.	Elonga-tion, per ct.	Remarks.
ade Bronze. .... per..........	.465	.1698	41,225	17,668	41½	These test pieces were all 6" long between the shoulders.
Bronze.... inc.......... opper......	.465	.1698	78,327	......	2½	
Bronze.... inc.......... opper......	.460	.1661	72,246	....	2½	

st brass on the above list is an extremely tough metal with low mit, made purposely so as to "upset" easily. The other, which is uminum-brass No 2, is very hard.

e not in this country or in England any official standard by which of the physical characteristics of cast metals. There are two con- at are absolutely necessary to be known before we can make a parison of different materials: namely, whether the casting was dry or green sand or in a chill, and whether it was attached to a sting or cast by itself. It has also been found that chill-castings er results than sand-castings, and that bars cast by themselves y for testing almost invariably run higher than test-bars attached gs. It is also a fact that bars cut out from castings are generally han bars cast alone. (E. H. Cowles.)

on as to Reported Strength of Alloys.—The same in strength which has been found in tests of gun-metal (copper noted above, must be expected in tests of aluminum bronze and its l alloys. They are exceedingly subject to variation in density caused by *differences in method of molding and casting*, te *uring, size and shape of casting*, depth of "sinking head,"

Aluminum Hardened by Addition of Copper Rolle[d] Sheets .04 Inch Thick. (*The Engineer*, Jan 2, 1891.)

Al. Per cent.	Cu. Per cent.	Sp. Gr. Calculated.	Sp. Gr. Determined.	Tensile Stre[ss] in pounds [per] square inc[h]
100	..	....	2.67	26,535
98	2	2.78	2.71	43,563
96	4	2.90	2.77	44,130
94	6	3.02	2.82	54,773
92	8	3.14	2.85	50,374

Tests of Aluminum Alloys.

(Engineer Harris, U. S. N., Trans. A. I. M. E., vol. xviii.)

Composition.					Tensile Strength, per sq. in. lbs.	Elastic Limit, lbs. per sq. in.	Elongation, per ct.	Re[duc]tio[n of] Ar[ea] pe[r ct.]
Copper.	Aluminum.	Silicon.	Zinc.	Iron.				
91.50%	6.50%	1.75%	....	0.25%	60,700	18,000	23.2	3[illegible]
88.50	9.33	1.66	....	0.50	66,000	27,000	3.8	7[illegible]
91.50	6.50	1.75	....	0.25	67,600	24,000	13.	2[illegible]
90.00	9.00	1.00	....	....	72,830	33,000	2.40	[illegible]
63.00	3.33	0.33	33.33%	....	82,200	60,000	2.33	[illegible]
63.00	3.33	0.33	33.33	....	70,400	55,000	0.4	[illegible]
91.50	6.50	1.75	....	0.25	59,100	19,000	15.1	2[illegible]
93.00	6.50	0.50	....	....	53,000	19,000	6.2	1[illegible]
88.50	9.33	1.66	....	0.50	69,930	33,000	1.33	[illegible]
92.00	6.50	0.50	....	....	46,530	17,000	7.8	1[illegible]

For comparison with the above 6 tests of "Navy Yard Bronze," C[u] ... Sn 10, Zn 2, are given in which the T. S. ranges from 18,000 to 24,590, [E]... from 10,000 to 13,000, El. 2.5 to 5.8%, Red. 4.7 to 10.89.

Alloys of Aluminum, Silicon and Iron.

M. and E. Bernard have succeeded in obtaining through electrolysis[,] treating directly and without previous purification, the aluminum ea[rths] (red and white bauxites) the following:

Alloys such as ferro-aluminum, ferro-silicon-aluminum and silicon-alu[mi]num, where the proportion of silicon may exceed 10% which are emplo[yed] in the metallurgy of iron for refining steel and cast-iron.

Also silicon-aluminum, where the proportion of silicon does not exc[eed] 10%, which may be employed in mechanical constructions in a rolle[d or] hammered condition, in place of steel, on account of their great resista[nce,] especially where the lightness of the piece in construction constitutes [one] of the main conditions of success.

The following analyses are given:

1. Alloys applied to the metallurgy of iron, the refining of steel and [cast] iron:

Types.	Aluminum.	Iron.	Silicon.	Mangane[se]
No. 1......................	70%	25%	5%	0%
No. 2......................	70	20	10	0
No. 3......................	70	15	15	0
No. 4......................	70	10	20	0
No. 5......................	70	10	10	10
No. 6......................	70	trace	20	10

2. Mechanical alloys:

Types.	Aluminum.	Silicon.	Iro[n]
No. 1......................................	92%	6.75%	1.[illegible]
No. 2......................................	90	9.25	0.[illegible]
No. 3......................................	90	10.00	tra[ce]

Up to this time it has been thought that silicon was rather injurious w[hen] alloyed with aluminum. From numerous experiences it has been dem[on]strated [illegible] to aluminum some remarkable properties of resista[nce] [illegible] with alloys where the proportion of iron was very [illegible] silicon in the neighborhood of 10%. Above that

minum-Antimony Alloys.—Dr. C. R. Alder Wright describes aluminum-antimony alloys in a communication read before the Society emical Industry. The results of his researches do not disclose the nce of a commercially useful alloy of these two metals, and have r scientific than practical interest. A remarkable point is that the with the chemical composition *Al Sb* has a higher melting point than aluminum or antimony alone, and that when aluminum is added to antimony the melting-point goes up from that of antimony (450° C.) ertain temperature rather above that of silver (1000° C.).

ALLOYS OF MANGANESE AND COPPER.

rious Manganese Alloys.—E. H. Cowles, in Trans. A. I. M. E., viii, p. 495, states that as the result of numerous experiments on res of the several metals, copper, zinc, tin, lead, aluminum, iron, and anese, and the metalloid silicon, and experiments upon the same in aining tensile strength, ductility, color, etc., the most important binations appear to be about as follows:

hat pure metallic manganese exerts a bleaching effect upon copper radical in its action even than nickel. In other words, it was found 8½% of manganese present in copper produces as white a color in the ing alloy as 25% of nickel would do, this being the amount of each ed to remove the last trace of red.

hat upwards of 20% or 25% of manganese may be added to copper withducing its ductility, although doubling its tensile strength and chang- color.

hat manganese, copper, and zinc when melted together and poured moulds behave very much like the most "yeasty" German silver, ding an ingot which is a mass of blow-holes, and which swells up the mould before cooling.

hat the alloy of manganese and copper by itself is very easily ed.

hat the addition of 1.25% of aluminum to a manganese-copper alloy ts it from one of the most refractory of metals in the casting process metal of superior casting qualities, and the non-corrodibility of which any instances greater than that of either German or nickel silver.

ilver-bronze" alloy especially designed for rods, sheets, and wire e following composition: Manganese, 18; aluminum, 1.20; silicon, 0.5;

(4) Good bearing-metals should show small friction. It is true tha
is almost wholly a question of the lubricant used; but the metal of
ing has certainly some influence.

(5) Other things being equal, the best bearing-metal is that wh
slowest.

The principal constituents of bearing-metal alloys are copper,
zinc, antimony, iron, and aluminum. The following table gives the
ents of most of the prominent bearing-metals as analyzed at the
vania Railroad laboratory at Altoona.

Analyses of Bearing-metal Alloys.

Metal.	Copper.	Tin.	Lead.	Zinc.	Antimony
Camelia metal	70.20	4.25	14.75	10.20	......
Anti-friction metal	1.60	98.13	......	......	......
White metal	......	......	87.92	......	12.0
Car-brass lining	......	trace	84.87	......	15.1
Salgee anti-friction	4.01	9 91	1.15	85.57	......
Graphite bearing-metal	......	14.38	67.73	......	16.7
Antimonial lead	......	......	80.69	......	18.8
Carbon bronze	75.47	9.72	14.57	......	......
Cornish bronze	77.83	9.60	12.40	trace	......
Delta metal	92.39	2.37	5.10	......	......
*Magnolia metal	trace	......	83.55	trace	16.4
American anti-friction metal	......	......	78.44	0.98	19.6
Tobin bronze	59.00	2.16	0.31	38.40	......
Graney bronze	75.80	9.20	15.06	......	......
Damascus bronze	76.41	10.60	12.52	......	......
Manganese bronze	90.52	9.58	......	......	......
Ajax metal	81.24	10.98	7.27	......	......
Anti-friction metal	......	......	88.32	......	11.9
Harrington bronze	55.73	0.97	......	42.67	......
Car-box metal	......	......	84.33	trace	14.3
Hard lead	......	......	94.40	......	6.0
Phosphor-bronze	79.17	10.22	9.61	......	......
Ex. B. metal	76.80	8.00	15.00	......	......

Other constituents:
(1) No graphite.
(2) Possible trace of carbon.
(3) Trace of phosphorus.
(4) Possible trace of bismuth.
(5) No manganese.
(6) Phosphorus or arsenic, (
(7) Phosphorus, 0.94.
(8) Phosphorus, 0.20.

* Dr. H. C. Torrey says this analysis is erroneous and that
metal always contains tin.

As an example of the influence of minute changes in an alloy,
rington bronze, which consists of a minute proportion of iron in
zinc alloy, showed after rolling a tensile strength of 75,000 lbs. and
gation in 2 inches.

In experimenting on this subject on the Pennsylvania Railroad,
number of the bearings were made of a standard bearing-metal,
same number were made of the metal to be tested. These bear
placed on opposite ends of the same axle, one side of the car h
standard bearings, the other the experimental. Before going int
the bearings were carefully weighed, and after a sufficient time t
again weighed.

The standard bearing-metal used is the "S bearing-metal" of t
phor-bronze Smelting Co. It contains about 79.70% copper, 9.50%
tin, and 0.80% phosphorus. A large number of experiments have sh
the loss of weight of a bearing of this metal is 1 lb. to each 18,000
miles travelled. Besides the measurement of wear, observations w
on the frequency of "hot boxes" with the different metals.

The results of the tests for wear, so far as given, are condensed
following table.

Metal.	Composition.					Rate of Wear.
	Copper.	Tin.	Lead.	Phos.	Arsenic.	
rd............	79.70	10.00	9.50	0.80	......	100
-tin..........	87.50	12.50	....	....	......	148
-tin, second experiment, same metal						153
-tin, third experiment, same metal						147
-bronze	89.20	10.00	...	....	0.80	142
-bronze... ..	79.20	10.00	7.00	....	0.80	115
-bronze.... .	79.70	10.00	9.50	....	0.80	101
ronze	77.00	10.50	12.50	....	....	92
ronze, second experiment, same metal						92.7
"B"	77.00	8.00	15.00	....	....	86.5

ld copper-tin alloy of 7 to 1 has repeatedly proved its inferiority to the or-bronze metal. Many more of the copper-tin bearings heated the phosphor-bronze. The showing of these tests was so satisfac- at phosphor-bronze was adopted as the standard bearing-metal of nsylvania R.R., and was used for a long time.

xperiments, however, were continued. It was found that arsenic ally takes the place of phosphorus in a copper-tin alloy, and three ere made with arsenic-bronzes as noted above. As the proportion is increased to correspond with the standard, the durability increases In view of these results the "K" bronze was tried, in which neither orus nor arsenic were used, and in which the lead was increased he proportion in the standard phosphor-bronze. The result was that al wore 7.30% slower than the phosphor-bronze. No trouble from was experienced with the "K" bronze more than with the standard. lley continues:

out this time we began to find evidences that wear of bearing-metal aried in accordance with the following law: "That alloy which has test power of distortion without rupture (resilience), will best resist It was now attempted to design an alloy in accordance with this cing first the proportions of copper and tin, 9½ parts copper to 1 of settled on by experiment as the standard, although some evidence at time tends to show that 12 or possibly 15 parts copper to 1 of tin ave been better. The influence of lead on this copper-tin alloy seems uch the same as a still further diminution of tin. However, the y of the metal to yield under pressure increases as the amount of minished, and the amount of the lead increased, so a limit is set to of lead. A certain amount of tin is also necessary to keep the lead with the copper.

ngs were cast of the metal noted in the table as alloy "B," and it .5% slower than the standard phosphor-bronze. This metal is now ndard bearing-metal of the Pennsylvania Railroad, being slightly l in composition to allow the use of phosphor-bronze scrap. The adopted is: Copper, 105 lbs.; phosphor-bronze, 60 lbs.; tin, 9¾ lbs.; ¼ lbs. By using ordinary care in the foundry, keeping the metal ered with charcoal during the melting, no trouble is found in casting arings with this metal. The copper and the phosphor-bronze can be he pot before putting it in the melting-hole. The tin and lead should d after the pot is taken from the fire.

not known whether the use of a little zinc, or possibly some other ation, might not give still better results. For the present, however, y is considered to fulfil the various conditions required for good -metal better than any other alloy. The phosphor-bronze had an tensile strength of 30,000 lbs., with 6% elongation, whereas the alloy d 24,000 lbs. tensile strength and 11% elongation.

ther bearing-metals, see Alloys containing antimony, on next page.

ALLOYS CONTAINING ANTIMONY.

VARIOUS ANALYSES OF BABBITT METAL AND OTHER ALLOYS CONTAI ANTIMONY.

	Tin.	Copper	Antimony.	Zinc.	Lead.	Bism
Babbitt metal	50	1	5 parts	....	....	.....
for light duty	=89.3	1.8	8.9 per ct.	....	....	.....
Harder Babbitt	96	4	8 parts	....	....	.....
for bearings*	=88.9	3.7	7.4 per ct.	....	....	.....
Britannia......	85.7	1.0	10.1	2.9	....	.....
"	81.9	....	16.2	1.9	....	.....
"	81.0	2	16.	1.	....	.....
"	70.5	4	25.5	....	....	.....
"	22	10	62.	6.	....	.....
"Babbitt"....	45 5	1.5	13.	....	40.0	.....
Plate pewter..	89.3	1.8	7.1	....	... ..	
White metal...	85	5	10.	Bearings on Ger. locomo		

* It is mixed as follows: Twelve parts of copper are first melted an 36 parts of tin are added; 24 parts of antimony are put in, and then 3 of tin, the temperature being lowered as soon as the copper is me order not to oxidize the tin and antimony, the surface of the bath protected from contact with the air. The alloy thus made is subseq remelted in the proportion of 50 parts of alloy to 100 tin. (Joshua Ro

White-metal Alloys.—The following alloys are used as lining by the Eastern Railroad of France (1890):

Number.	Lead.	Antimony.	Tin.	Copp
1........	65	25	0	10
2............	0	11.12	83.33	5
3.	70	20	10	0
4............	80	8	12	0

No. 1 is used for lining cross-head slides, rod-brasses and axle-be No. 2 for lining axle-bearings and connecting-rod brasses of heavy e No. 3 for lining eccentric straps and for bronze slide-valves; and No metallic rod-packing.

Some of the best-known white-metal alloys are the following (C of Hoveler & Dieckhaus, London, 1893):

	Tin.	Antimony.	Lead.	Copper.
1. Parsons'.................	86	1	2	2
2. Richards'................	70	15	10½	4½
3. Babbitt's................	55	18	23½	3½
4. Fentons'.................	16	0	0	5
5. French Navy..........	7½	0	7	7
6. German Navy...........	85	7½	0	7½

"There are engineers who object to white metal containing lead This is, however, a prejudice quite unfounded, inasmuch as lead a often have properties of great use in white alloys."

It is a further fact that an "easy liquid" alloy must not contai than 18% of antimony, which is an invaluable ingredient of white m improving its hardness; but in no case must it exceed that margin, would reduce the plasticity of the compound and make it brittle.

Hardest alloy of tin and lead: 6 tin, 4 lead. Hardest of all tin alloy tin, 18 antimony, 8 copper.

Alloy for thin open-work, ornamental castings: Lead 2, antin White metal for patterns: Lead 10, bismuth 6, antimony 2, common tin 10.

Type-metal is made of various proportions of lead and antimon 17% to 20% antimony according to the hardness desired.

Babbitt Metals. (C. R. Tompkins, *Mechanical News*, Jan.

The practice of lining journal-boxes with a metal that is sufficien *ble to be* [illegible] *in a* common ladle is not always so much for the *of* [illegible] *ion* properties as for the convenience and cheap *forr* [illegible] *aring* in line with the shaft without the nec

Boxes that are bored, no matter how accurate, require great and attaching them to the frame or other parts of a machine. d practice, however, to use the shaft for the purpose of castgs, especially if the shaft be steel, for the reason that the hot to spring it; the better plan is to use a mandrel of the same larger for this purpose. For slow-running journals, where oderate, almost any metal that may be conveniently melted ree will answer the purpose. For wearing properties, with a ed, there is probably nothing superior to pure zinc, but when with some other metal it shrinks so much in cooling that it firmly in the recess, and soon works loose; and it lacks those roperties which are necessary in order to stand high speed. fting, and all work where the speed is not over 300 or 400 *r. p.* f 8 parts zinc and 2 parts block-tin will not only wear longer position of this class, but will successfully resist the force of The tin counteracts the shrinkage, so that the metal, if not ill firmly adhere to the box until it is worn out. But this not possess sufficient anti-friction properties to warrant its use g journals.

he soft metals in use there are none that possess greater antirties than pure lead; but lead alone is impracticable, for it is so nnot be retained in the recess. But when by any process lead ntly hardened to be retained in the boxes without materially ti-friction properties, there is no metal that will wear longer unning journals. With most of the best and most popular etals in use and sold under the name of the Babbitt metal, d.

ntimony have the property of combining with each other in s without impairing the anti-friction properties of either. The dens the lead, and when mixed in the proportion of 80 parts t with 20 parts antimony, no other known composition of ses greater anti-friction or wearing properties, or will stand a without heat or abrasion. It runs free in its melted state, has and is better adapted to light high-speeded machinery than wn metal. Care, however, should be manifested in using it, never be heated beyond a temperature that will scorch a dry

ent compositions are sold under the name of Babbitt metal. l, but more are worthless; while but very little genuine Babbitt hat is made strictly according to the original formula. Most sold under that name are the refuse of type-foundries and -works, melted and cast into fancy ingots with special brands, r the name of Babbitt metal.

at the present time to determine the exact formulas used by abbitt, the inventor of the recessed box, as a number of differ. are given for that composition. Tin, copper, and antimony dients, and from the best sources of information the original ere as follows:

		Another writer gives:
tin	= 89.3%	83.3%
copper	= 3.6%	8.3%
antimony	= 7.1%	8.3%

was first melted, and the antimony added first and then about pounds of tin, the whole kept at a dull-red heat and constantly the metals were thoroughly incorporated, after which the e tin was added, and after being thoroughly stirred again it t into ingots. When the copper is thoroughly melted, and timony is added, a handful of powdered charcoal should be he crucible to form a flux, in order to exclude the air and prenony from vaporizing; otherwise much of it will escape in the or and consequently be wasted. This metal, when carefully robably one of the best metals in use for lining boxes that are heavy weight and wear; but for light fast running journals nders it more susceptible to friction, and it is more liable to metal composed of lead and antimony in the p[illegible] just

SOLDERS.

Common solders, equal parts tin and lead; fine solder, 2 tin to 1 le... solder, 2 lead, 1 tin.

Fusing-point of tin-lead alloys:

Tin 1 to lead 25 .... 558° F.		Tin 1½ to lead 1	
" 1 " " 10 541		" 2 " " 1	
" 1 " " 5 511		" 3 " " 1	
" 1 " " 3 482		" 4 " " 1	
" 1 " " 2 441		" 5 " " 1	
" 1 " " 1 370		" 6 " " 1	

Common pewter contains 4 lead to 1 tin.

Gold solder: 14 parts gold, 6 silver, 4 copper. Gold solder fo... gold: 25 parts gold, 25 silver, 12½ brass, 1 zinc.

Silver solder: Yellow brass 70 parts, zinc 7, tin 11½. Another: ... parts, brass (3 copper, 1 zinc) 73, zinc 4.

German-silver solder: Copper 38, zinc 54, nickel 8.

Novel's solders for aluminum:

Tin 100 parts, lead 5;	melts at 536° to 53...
" 100 " zinc 5;	"	536 to 62...
" 1000 " copper 10 to 15;	"	662 to 842
" 1000 " nickel 10 to 15;	"	662 to 842

Novel's solder for aluminum bronze: Tin 900 parts, copper 100, ... to 3. It is claimed that this solder is also suitable for joining alu... copper, brass, zinc, iron, or nickel.

ROPES AND CABLES.

STRENGTH OF ROPES.

(A. S. Newell & Co., Birkenhead. Klein's Translation of Weisbac... part 1, sec. 2.)

Hemp.		Iron.		Steel.	
Girth.	Weight per Fathom.	Girth.	Weight per Fathom.	Girth.	Weight per Fathom.
Inches.	Pounds.	Inches.	Pounds.	Inches.	Pounds.
2¾	2	1	1		
		1⅛	1½	1	1
3¾	4	1⅝	2		
		1¾	2½	1⅛	1½
4½	5	1⅞	3		
		2	3½	1⅝	2
5½	7	2⅛	4	1¾	2½
		2¼	4½		
6	9	2⅜	5	1⅞	3
		2½	5½		
6½	10	2⅝	6	2	3½
		2¾	6½	2⅛	4
7	12	2⅞	7	2¼	4½
		3	7½		
7½	14	3⅛	8	2⅜	5
		3¼	8½		
8	16	3⅜	9	2½	5½
		3½	10	2⅝	6
8½	18	3⅝	11	2¾	6½
		3¾	12		
9½	22	3⅞	13	3¼	8
10	26	4	14		
11	30	4¼	15	3⅜	9
		4⅜	16		
		4½	18	3½	10
		4⅝	20	3¾	12

Flat Ropes.

mp.	Iron.		Steel.		Tensile Strength.
Weight per Fathom.	Girth.	Weight. per Fathom.	Girth.	Weight per Fathom.	
Pounds.	Inches.	Pounds.	Inches.	Pounds.	Gross tons.
20	2¼ × ½	11			20
24	2½ × ½	13			23
26	2¾ × ⅝	15			27
28	3 × ⅝	16	2 × ½	10	28
30	3¼ × ⅝	18	2¼ × ½	11	32
36	3½ × ⅝	20	2¼ × ½	12	36
40	3¾ × 11/16	22	2½ × ½	13	40
45	4 × 11/16	25	2¾ × ⅜	15	45
50	4¼ × ¾	28	3 × ¾	16	50
55	4½ × ¾	32	3¼ × ⅜	18	56
60	4⅝ × ¾	34	3½ × ⅜	20	60

ng Load, Diameter, and Weight of Ropes and Chains. (Klein's Weisbach, vol. iii, part 1, sec. 2, p. 561.)

ropes: d = diam. of rope. Wire rope: d = diam. of wire, n = of wires, G = weight per running foot, k = permissible load in er square inch of section, P = permissible load on rope or chain. hains: d = diam. of iron used; inside dimensions of oval $1.5d$ and ch link is a piece of chain $2.6d$ long. G_0 = weight of a single link = .; G = weight per running foot = $9.73d^2$ lbs.

	Hempen Rope.		Wire Rope.
	Dry and Untarred.	Wet or Tarred.	
=	1420	1160	17000
=	$0.03\sqrt{P}$	$0.033\sqrt{P}$	$0.0087\sqrt{\frac{P}{n}}$
=	$1120d^2 = 2855G$	$916d^2 = 1975G$	$13350nd^2 = 4590G$
=	$1.28d^2 = 0.00035P$	$1.54d^2 = 0.0005P$	$2.91nd^2 = 0.000218P$

	Open-link Chain.	Stud-link Chain.
(lbs.) =	8500	11400
(ins.) =	$0.0087\sqrt{P}$	$0.0076\sqrt{P}$
(lbs.) =	$13350d^2 = 1360G$	$17800d^2 = 1660G$
(lbs.) =	$9.73d^2 = 0.000737P$	$10.65d^2 = 0.0006P$

Chains 4/3 times as strong as open-link variety. [This is contrary to ements of Capt. Beardslee, U. S. N., in the report of the U. S. Test He holds that the open link is stronger than the studded link. See nte].

STRENGTH AND WEIGHT OF WIRE ROPE, HEMPEN ROPE, A
CHAIN CABLES. (Klein's Weisbach.)

Breaking Load in tons of 2240 lbs.	Kind of Cable.	Girth of Wire Rope and of Hemp Rope Diameter of Iron of Chain, inches.	Weight of O Foot in lengt Pounds.
1 Ton............	Wire Rope	1.0	0.125
	Hemp Rope	2.0	0.177
	Chain	¼	0.500
8 Tons..........	Wire Rope	2.0	0.438
	Hemp Rope	5.0	0.978
	Chain	½	2.667
12 Tons.........	Wire Rope	2.5	0.753
	Hemp Rope	7.0	2.036
	Chain	11/16	4.502
16 Tons..........	Wire Rope	3.0	1.136
	Hemp Rope	8.0	2.363
	Chain	13/16	6.169
20 Tons...... ...	Wire Rope	3.5	1.546
	Hemp Rope	9.0	3.225
	Chain	29/32	7.674
24 Tons..........	Wire Rope	4.0	2.043
	Hemp Rope	10.0	4.106
	Chain	31/32	8.836
30 Tons.....	Wire Rope	4.5	2.725
	Hemp Rope	11.0	5.000
	Chain	1.1/16	10.335
36 Tons..........	Wire Rope	5.0	3.723
	Hemp Rope	12.5	5.940
	Chain	1.3/16	13.01
44 Tons....	Wire Rope	5.5	4.50
	Hemp Rope	14.0	6.94
	Chain	1.5/16	16.00
54 Tons..........	Wire Rope	6.0	5.67
	Hemp Rope	15.0	7.92
	Chain	1.7/16	19.16

Length sufficient to provide the maximum working stress:

Hempen rope, dry and untarred..................	2855	feet.
" " wet or tarred.....................	1975	"
Wire rope.................................... .. .	4590	"
Open-link chain....	1360	"
Stud chain..	1660	"

Sometimes, when the depths are very great, ropes are given approxima the form of a body of uniform strength, by making them of separate pie whose diameters diminish towards the lower end. It is evident that by means the tensions in the fibres caused by the rope's own weight can considerably diminished.

Rope for Hoisting or Transmission. Manila Ro (C. W. Hunt Company, New York.)—Rope used for hoisting or for tra mission of power is subjected to a very severe test. Ordinary rope cha and grinds to powder in the centre, while the exterior may look as thou it was little worn.

In bending a rope over a sheave, the strands and the yarns of these stra slide a small distance upon each other, causing friction, and wear the r internally.

The "Stevedore" rope used by the C. W. Hunt Co. is made by lubricati the fibres with plumbago, mixed with sufficient tallow to hold it in positi This lubricates the yarns of the rope, and prevents internal chafing a wear. After running a short time the exterior of the rope gets compre and coated with the lubricant.

I- ...cturing rope, the fibres are first spun into a yarn, this ya ... a direction called "right hand." From 20 to 80 of th ... on the size of the rope, are then put together and twis ... irection, or "left hand," into a strand. Three of the for a 3-strand, or four for a 4-strand rope, are then twisted , the twist being again in the "right hand" direction. When the s twisted, it untwists each of the threads, and when the three are twisted together into rope, it untwists the strands, but again p the threads. It is this opposite twist that keeps the rope in its orm. When a weight is hung on the end of a rope, the tendency is rope to untwist, and become longer. In untwisting the rope, it ist the threads up, and the weight will revolve until the strain of isting strands just equals the strain of the threads being twisted In making a rope it is impossible to make these strains exactly each other. It is this fact that makes it necessary to take out the in a new rope, that is, untwist it when it is put at work. The rist that should be put in the threads has been ascertained approx- by experience.

mount of work that the rope will do varies greatly. It depends not the quality of the fibre and the method of laying up the rope, but the kind of weather when the rope is used, the blocks or sheaves ch it is run, and the strain in proportion to the strain put upon the he principal wear comes in practice from defective or badly set from excess of load and exposure to storms.

ds put upon the rope should not exceed those given in the tables, ost economical wear. The indications of excessive load will be the ning out of the rope, or one of the strands slipping out of its proper A certain amount of twist comes out in using it the first day or after that the rope should remain substantially the same. If it , the load is too great for the durability of the rope. If the rope the outside, and is good on the inside, it shows that it has been running over the pulleys or sheaves. If the blocks are very small, crease the sliding of the strands and threads, and result in a more ernal wear. Rope made for hoisting and for rope transmission is nade with four strands, as experience has shown this to be the most ble.

ength and weight of "stevedore" rope is estimated as follows:

reaking strength in pounds = 720 (circumference in inches)^2;
Veight in pounds per foot = .032 (circumference in inches)^2.

Technical Words relating to Cordage most frequently e:

–Fibres twisted together.
D.—Two or more *small yarns* twisted together.
—The same as a thread but a little larger *yarns.*
.—Two or more *large yarns* twisted together.
-Several threads twisted together.
-Several *strands* twisted together.
R.—A rope of three *strands.*
D-LAID.—A rope of four *strands.*
—Three hawsers twisted together.
are laid up left-handed into *strands.*
ds are laid up right-handed into rope.
rs are laid up left-handed into a cable.
:
y twisting strands together in making the rope.
b by joining to another rope by interweaving the strands.
ED.—By winding a string around the end to prevent untwisting.
.—When covered by winding a yarn continuously and tightly
.
ED.—By wrapping with canvas.
.—When two parts are bound together by a yarn, thread or string.
—When painted, tarred or greased to resist wet.
—To pull on a rope.
-Drawn tight or strained.

ing of Ropes.—The splice in a transmission rope is not only the part of the rope but is the first part to fail when the rope is worn the rope is larger at the splice, the projecting part will wear on the and the rope fail from the cutting off of the strands. The following s are given for splicing a *4-strand rope.*

ngravings show each successive operation in splicing a ope. *Each engraving was made from* a full-size specimen

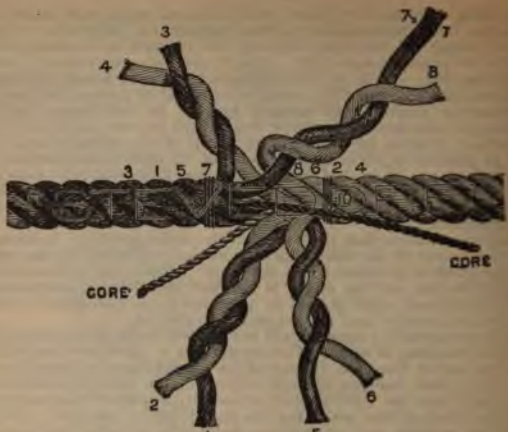


FIG. 78.

FIG. 79.

FIG. 80.

FIG. 81.
SPLICING OF ROPES.

of twine, 9 and 10, around the rope to be spliced, about 6 feet
d. Then unlay the strands of each end back to the twine.
opes together and twist each corresponding pair of strands
ep them from being tangled, as shown in Fig. 78.
9 is now cut, and the strand 8 unlaid and strand 7 carefully laid
r a distance of four and a half feet from the junction.
6 is next unlaid about one and a half feet and strand 5 laid in

the cores are now cut off so they just meet.
d 1 four and a half feet, laying strand 2 in its place.
d 3 one and a half feet, laying in strand 4.
trands off to a length of about twenty inches, for convenience
on.
w assumes the form shown in Fig. 79 with the meeting points
three feet apart.
f strands is successively subjected to the following operation:
oint of meeting of the strands 8 and 7, unlay each one three
oth the strand 8 and the strand 7 in halves as far back as they
id and "whip" the end of each half strand with a small

the strand 7 is now laid in three turns and the half of 8 also
turns. The half strands now meet and are tied in a simple
80, making the rope at this point its original size.
now opened with a marlin spike and the half strand of 7
d the half strand of 8 by passing the end of the half strand 7
ope, as shown in the engraving, drawn taut and again worked
alf strand until it reaches the half strand 13 that was not laid
strand 13 is now split, and the half strand 7 drawn through
us made, and then tucked under the two adjacent strands, as
81. The other half of the strand 8 is now wound around the
nd 7 in the same manner. After each pair of strands has
n this manner, the ends are cut off at 12, leaving them about
g. After a few days' wear they will draw into the body of the
ff, so that the locality of the splice can scarcely be detected.
sting. (C. W. Hunt Co.).—The amount of coal that can be
rope varies greatly. Under the ordinary conditions of use
from 5000 to 8000 tons. Where the circumstances are more
amounts run up frequently to 12,000 or 15,000 tons, occasion-
nd in one case 32,400 tons to a single fall.
ting rope is first put in use, it is likely from the strain put upon
when the block is loosened from the tub. This occurs in the
two only. The rope should then be taken down and the
out of the rope. When put up again the rope should give
uble until worn out.
ry that the rope should be much larger than is needed to bear
n the load.
perience for many years has substantially settled the most
e of rope to be used which is given in the table below.
es are not spliced, as it is difficult to make a splice that will
hile running over the sheaves, and the increased wear to be
is way is very small.
lly hoisted with what is commonly called a "double whip;"
running block that is attached to the tub which reduces the
ope to approximately one half the weight of the load hoisted.
table gives the usual sizes of hoisting rope and the proper
:

Stevedore Hoisting-rope.

C. W. Hunt Co.

e of ns.	Proper Working Strain on the Rope in lbs.	Nominal size of Coal tubs. Double whip.	Approximate Weight of a Coil, in lbs.
	350	1/6 to 1/5 tons.	360
	500	1/5 " ¼ "	480
	650	¼ " ⅓ "	[illegible]
	800	½ " ¾ "	[illegible]
	1000	¾ " 1 "	[illegible]

is ordered by circumference, transmissio [illegible] *ter.*

Weight and Strength of Manila Cordage.

Dodge Manufacturing Co.

Size. Diameter in inches.	Weight of 100 Fathoms Manila in lbs.	Strain Borne by New Rope pounds.	Feet in a pound.	Size. Diameter in inches.	Weight of 100 Fathoms Manila in lbs.	Strain Borne by New Rope pounds.
3/16	12	540	50′	1 5/16	310	16,000
1/4	18	780	33′ 4″	1 3/8	346	18,062
5/16	24	1,000	25	1 1/2	390	20,250
3/8	30	1,280	20	1 9/16	435	22,500
7/16	37	1,562	17 8	1 5/8	480	25,000
1/2	46	2,250	13	1 3/4	581	30,250
9/16	65	3,062	9 3	2	678	36,000
5/8	80	4,000	7 6	2 1/8	797	42,250
3/4	98	5,000	6	2 1/4	920	49,000
13/16	120	6,250	5	2 1/2	1,106	56,250
7/8	142	7,500	4 3	2 5/8	1,265	64,000
1	170	9,000	3 6	2 7/8	1,420	72,250
1 1/16	200	10,500	3	3	1,572	81,000
1 1/8	230	12,250	2 7	3 1/8	1,700	90,250
1 1/4	271	14,000	2 3	3 3/8	1,951	100,000

T. Spencer Miller (*Eng'g News*, Dec. 6, 1890) gives the following tab breaking strength of manila rope, which he considers more reliable the strength computed by Mr. Hunt's formula, Breaking strength = (circumference in inches)^2. Mr. Miller's formula is: Breaking weight l circumference^2 × a coefficient which varies from 900 for 1/2″ to 700 f diameter rope, as shown in the table.

Diam. in.	Circumference. in.	Ultimate Strength. lbs.	Coefficient.	Diam. in.	Circumference. in.	Ultimate Strength. lbs.
1/2	1 1/2	2,000	900	1 1/4	3 3/4	10,000
5/8	2	3,250	845	1 3/8	4 1/4	13,000
3/4	2 1/4	4,000	820	1 1/2	4 1/2	15,000
7/8	2 3/4	6,000	790	1 5/8	5	18,200
1	3	7,000	780	1 7/8	5 1/2	21,750
1 1/8	3 1/2	9,350	765	2	6	25,000

For rope-driving Mr. Hunt recommends that the working strain sh not exceed 1/20 of the ultimate breaking strain. For further data on see "Rope-driving."

Knots.—A great number of knots have been devised of which only are illustrated, but those selected are the most frequently used the cuts, Fig. 82, they are shown open, or before being drawn taut, in to show the position of the parts. The names usually given to them a

A. Bight of a rope.
B. Simple or Overhand knot.
C. Figure 8 knot.
D. Double knot.
E. Boat knot.
F. Bowline, first step.
G. Bowline, second step.
H. Bowline completed.
I. Square or reef knot.
J. Sheet bend or weaver's knot.
K. Sheet bend with a toggle.
L. Carrick bend.
M. Stevedore knot completed.
N. Stevedore knot commenced.
O. Slip knot.
P. Flemish loop.
Q. Chain knot with toggle.
R. Half-hitch.
S. Timber-hitch.
T. Clove hitch.
U. Rolling-hitch.
V. Timber-hitch and half-hi
W. Blackwall-hitch.
X. Fisherman's bend.
Y. Round turn and half-hi
Z. Wall knot commenced.
A A. " " completed.
B B. Wall knot crown comm
C C. " " " compl

principle of a knot is that no two parts, which would move in the direction if the rope were to slip, should lay along side of and touch ach other.
bowline is one of the most useful knots, it will not slip, and after strained is easily untied. Commence by making a bight in the rope, put the end through the bight and under the standing part as shown in en pass the end again through the bight, and haul tight.
e square or reef knot must not be mistaken for the "granny" knot slips under a strain. Knots *H*, *K* and *M* are easily untied after being r strain. The knot *M* is useful when the rope passes through an eye is held by the knot, as it will not slip and is easily untied after being ned.

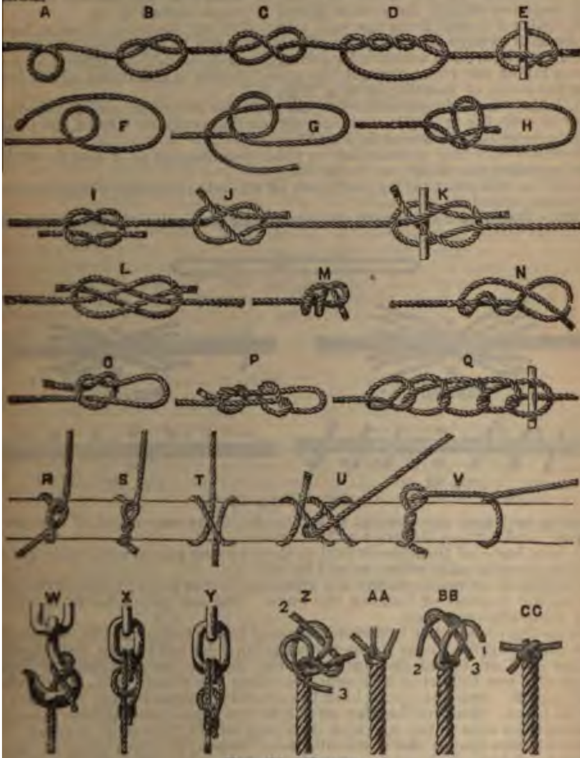


FIG. 82.—KNOTS.

timber hitch *S* looks as though it would give way, but it will not; the er the strain the tighter it will hold. The wall knot looks complicated, easily made by proceeding as follows: Form a bight with strand 1 the strand 2 around the end of it, and the strand 3 round the end and then through the bight of 1 as shown in the cut *Z*. Haul th when the appearance is as shown in *AA*. The end of the s laid over the centre of the knot, *strand 2 laid over 1 and 3 o* d of 3 is passed through the bight of 1 as shown in *BB*. is taut as shown in *CC*.

To Splice a Wire Rope.—The tools required will be a smal spike, nipping cutters, and either clamps or a small hemp-rope s which to wrap around and untwist the rope. If a bench-vise is a it will be found convenient.

In splicing rope, a certain length is used up in making the sp allowance of not less than 16 feet for ½ inch rope, and propor longer for larger sizes, must be added to the length of an endles ordering.

Having measured, carefully, the length the rope should be af ing, and marked the points *M* and *M′*, Fig. 83, unlay the strands f end *E* and *E′* to *M* and *M′* and cut off the centre at *M* and *M′*, and

(1). Interlock the six unlaid strands of each end alternately a them together so that the points *M* and *M′* meet, as in Fig. 84.

(2). Unlay a strand from one end, and following the unlay closely the seam or groove it opens, the strand opposite it belonging to end of the rope, until within a length equal to three or four times t of one lay of the rope, and cut the other strand to about the san from the point of meeting as at *A*, Fig. 85.

(3). Unlay the adjacent strand in the opposite direction, and foll unlay closely, lay in its place the corresponding opposite strand, c ends as described before at *B*, Fig. 85.

There are now four strands laid in place terminating at *A* and *B* eight remaining at *M M′*, as in Fig. 85.

It will be well after laying each pair of strands to tie them temp the points *A* and *B*.

Pursue the same course with the remaining four pairs of opposit

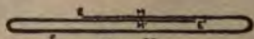


FIG. 83.

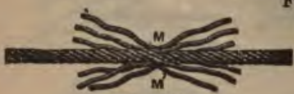


FIG. 84.

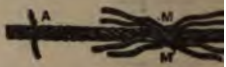


FIG. 85.

FIG. 86.

FIG. 87.

SPLICING WIRE ROPE.

stopping each pair about eight or ten turns of the rope short of t ing pair, and cutting the ends as before.

We now have all the strands laid in their proper places with the ive ends passing each other, as in Fig. 86.

All methods of rope-splicing are identical to this point; their va sists in the method of tucking the ends. The one given below i most generally practiced.

Clamp the rope either in a vise at a point to the left of *A*, Fig. 86 hand-clamp applied near *A*, open up the rope by untwisting suff cut the core at *A*, and seizing it with the nippers, let an assista out slowly, you following it closely, crowding the strand in its pla is all laid in. Cut the core where the strand ends, and push the into its place. Remove the clamps and let the rope close together Draw out the core in the opposite direction and lay the other stra centre of the rope, in the same manner. Repeat the operation a remaining points, and hammer the rope lightly at the points wher pass each other at *A*, *A*, *B*, *B*, etc., with small wooden mallets splice is complete, as shown in Fig. 87.

If a clamp and vise are not obtainable, two rope slings and sho levers may be used to untwist and open up the rope.

A rope spliced as above will be nearly as strong as the original smooth everywhere. After running a few days, the splice, if w cannot be found except by close examination.

The above instructions have been adopted by the leading rope

ture

SPRINGS.

efinitions.—A spiral spring is one which is wound around a fixed t or centre, and continually receding from it like a watch spring. A cal spring is one which is wound around an arbor, and at the same time ancing like the thread of a screw. An elliptical or laminated spring is e of flat bars, plates, or "leaves," of regularly varying lengths, super-d one upon the other.

aminated Steel Springs.—Clark (Rules, Tables and Data) gives following from his work on *Railway Machinery*, 1855:

$$\Delta = \frac{1.66L^3}{bt^3n}; \qquad s = \frac{bt^2n}{11.3L}; \qquad n = \frac{1.66L^3}{\Delta bt^3};$$

= elasticity, or deflection, in sixteenths of an inch per ton of load,
= working strength, or load, in tons (2240 lbs.),
= span, when loaded, in inches,
= breadth of plates, in inches, taken as uniform,
= thickness of plates, in sixteenths of an inch,
= number of plates.

OTE.—The span and the elasticity are those due to the spring when hted.

When extra thick back and short plates are used, they must be replaced n equivalent number of plates of the ruling thickness, prior to the em-ment of the first two formulæ. This is found by multiplying the num-of extra thick plates by the cube of their thickness, and dividing by the e of the ruling thickness. Conversely, the number of plates of the ruling kness given by the third formula, required to be deducted and replaced given number of extra thick plates, are found by the same calculation. It is assumed that the plates are similarly and regularly formed, and they are of uniform breadth, and but slightly taper at the ends.

euleaux's Constructor gives for semi-elliptic springs:

$$P = \frac{Snbh^2}{6l} \quad \text{and} \quad f = \frac{6Pl^3}{Enbh^3};$$

= max. direct fibre-strain in plate; b = width of plates;
= number of plates in spring; h = thickness of plates;
= one half length of spring; f = deflection of end of spring;
= load on one end of spring; E = modulus of direct elasticity.

he above formula for deflection can be relied upon where all the plates he spring are regularly shortened; but in semi-elliptic springs, as used, e are generally several plates extending the full length of the spring, the proportion of these long plates to the whole number is usually about fourth. In such cases $f = \frac{5.5Pl^3}{Enbh^3}$. (G. R. Henderson, Trans. A. S. M. E., xvi.)

order to compare the formulæ of Reuleaux and Clark we may make following substitutions in the latter: s in tons $= P$ in lbs. $\div$ 1120; $\Delta s =$ $L = 2l$; $t = 16h$; then

$$\Delta s = 16f = \frac{1.66 \times 8l^3 \times P}{4096 \times 1120 \times nbh^3}, \quad \text{whence} \quad f = \frac{Pl^3}{5,527,133},$$

ch corresponds with Reuleaux's formula for deflection if in the latter we $E = 33,162,800$.

lso $$s = \frac{P}{1120} = \frac{256nbh^2}{11.3 \times 2l}, \quad \text{whence} \quad P = \frac{12,687nbh^2}{l},$$

ch corresponds with Reuleaux's formula for working load when S in the er is taken at 76,120.

he value of E is usually taken at 30,000,000 and S at 80,000, in which case leaux's formulæ become

$$P = \frac{13,333nbh^2}{l} \quad \text{and} \quad f = \frac{Pl^3}{5,000,000nbh^3}.$$

Helical Steel Springs.—Clark quotes the following from the report Safety Valves (Trans. Inst. Engrs. and Shipbuilders in Scotland, 1874-5):

$$E = \frac{d^3 \times w}{D^4 \times C}.$$

E = compression or extension of one coil in inches,
d = diameter from centre to centre of steel bar constituting the sp
in inches,
w = weight applied, in pounds,
D = diameter, or side of the square, of the steel bar, in sixteenths o
inch,
C = a constant, which may be taken as 22 for round steel and
square steel.

NOTE.—The deflection E for one coil is to be multiplied by the numb free coils, to obtain the total deflection for a given spring.

The relation between the safe load, size of steel, and diameter of coil, be taken for practical purposes as follows:

$$D = \sqrt[3]{\frac{wd}{3}}, \text{ for round steel;}$$

$$D = \sqrt[3]{\frac{wd}{4.29}}, \text{ for square steel.}$$

Rankine's Machinery and Millwork, p. 390, gives the following:

$$\frac{W}{v} = \frac{cd^4}{64nr^3}; \qquad W_1 = \frac{.196fd^3}{r}; \qquad v_1 = \frac{12.566nfr^2}{cd};$$

$$\frac{W_1}{2} = \text{greatest safe sudden load.}$$

In which d is the diameter of wire in inches; c a co-efficient of trans elasticity of wire, say 10,500,000 to 12,000,000 for charcoal iron wire and r radius to centre of wire in coil; n effective number of coils; f greatest shearing stress, say 30,000; W any load not exceeding greatest safe v corresponding extension or compression; W_1 greatest safe load; a greatest safe steady extension or compression.

If the wire is square, of the dimensions $d \times d$, the load for a given d tion is greater than for a round wire of the diameter d in the ratio of 1.96 or of 1.43 to 1, or of 10 to 7, nearly.

Wilson Hartnell (Proc. Inst. M. E., 1882, p. 426), says: The size of a spring may be calculated from the formula on page 304 of "Rankine's ful Rules and Tables"; but the experience with Salter's springs has that the safe limit of stress is more than twice as great as there namely 60,000 to 70,000 lbs. per square inch of section with ⅜ inch wir about 50,000 with ½ inch wire. Hence the work that can be do springs of wire is four or five times as great as Rankine allows.

For ⅜ inch wire and under,

$$\text{Maximum load in lbs.} = \frac{12{,}000 \times (\text{diam. of wire})^3}{\text{Mean radius of springs}};$$

$$\text{Weight in lbs. to deflect spring 1 in.} = \frac{180{,}000 \times (\text{diam.})^4}{\text{Number of coils} \times (\text{rad.})^3}$$

The work in foot-pounds that can be stored up in a spiral spring lift it above 50 ft.

In a few rough experiments made with Salter's springs the coeffici rigidity was noticed to be 12,600,000 to 13,700,000 with ¼ inch wire; 11,0 for 11/32 inch; and 10,600,000 to 10,900,000 for ⅜ inch wire.

Helical Springs.—J. Begtrup, in the *American Machinist* of 18, 1892, gives formulas for the deflection and carrying capacity of springs of round and square steel, as follow:

$$\left.\begin{aligned} W &= .3927\frac{Sd^3}{D-d}, \\ F &= 8\frac{P(D-d)^3}{Ed^4}, \end{aligned}\right\} \text{ for round steel.}$$

$$\left.\begin{aligned} W &= .471\frac{Sd^3}{D-d}, \\ F &= 4.712\frac{P(D-d)^3}{Ed^4}, \end{aligned}\right\} \text{ for square steel.}$$

W = carrying capacity in pounds,
S = greatest tensile stress per square inch of material,
d = diameter of steel,
D = outside diameter of coil,
F = deflection of one coil,
E = torsional modulus of elasticity,
P = load in pounds.

these formulas the following table has been calculated by Mr. Beg- A spring being made of an elastic material, and of such shape as to great amount of deflection, will not be affected by sudden shocks or o the same extent as a rigid body, and a factor of safety very much an for rigid constructions may be used.

HOW TO USE THE TABLE.

designing a spring for continuous work, as a car spring, use a factor of safety than in the table; for intermittent working, as in n-engine governor or safety valve, use figures given in table; for steel multiply line W by 1.2 and line F by .59.

ple 1.—How much will a spring of ⅜″ round steel and 3″ outside er carry with safety? In the line headed D we find 3, and right un- th 473, which is the weight it will carry with safety. How many coils his spring have so as to deflect 3″ with a load of 400 pounds? Assum- odulus of elasticity of 12 millions we find in the centre line headed figure .0610; this is deflection of one coil for a load of 100 pounds; re $.061 \times 4 = .244''$ is deflection of one coil for 400 pounds load, and 3 = 12⅓ is the number of coils wanted. This spring will therefore be g when closed, counting working coils only, and stretch to 7¾″.

ple 2.—A spring 3¼″ outside diameter of 7/16″ steel is wound close; ich can it be extended without exceeding the limit of safety? We ximum safe load for this spring to be 702 pounds, and deflection of for 100 pounds load .0405 inches; therefore $7.02 \times .0405 = .284''$ is the t admissible opening between coils. We may thus, without know- load, ascertain whether a spring is overloaded or not.

ying Capacity and Deflection of Helical Springs of Round Steel.

iameter of steel. D = outside diameter of coil. W = safe working pounds—tensile stress not exceeding 60,000 pounds per square inch. flection by a load of 100 pounds of one coil, and a modulus of elasti- 10, 12 and 14 millions respectively. The ultimate carrying capacity about twice the safe load.

.25	.50	.75	1.00	1.25	1.50	1.75	2.00		
35	15	9	7	5	4.5	3.8	3.3		
.0276	.3588	1.433	3.562	7.250	12.88	20.85	31.57		
.0236	.3075	1.228	3.053	6.214	11.04	17.87	27.06		
.0197	.2562	1.023	2.544	5.178	9.200	14.89	22.55		
.50	.75	1.00	1.25	1.50	1.75	2.00	2.25	2.50	
107	65	46	36	29	25	22	19	17	
.0206	.0937	.2556	.5412	.9856	1.624	2.492	3.625	5.056	
.0176	.0804	.2191	.4639	.8448	1.392	2.136	3.107	4.334	
.0147	.0670	.182	.3866	.7010	1.160	1.780	2.589	3.612	
75	1.00	1.25	1.50	1.75	2.00	2.25	2.50	2.75	3.00
241	167	128	104	88	75	66	59	53	49
.0137	.0408	.0907	.1703	.2866	.4466	.6571	.9249	1.256	1.660
.0118	.0350	.0778	.1460	.2457	.3828	.5632	.7928	1.077	1.423
.0098	.0292	.0648	.1217	.2048	.3190	.4693	.6607	.8975	1.186
1.25	1.50	1.75	2.00	2.25	2.50	2.75	3.00	3.25	3.50
368	291	245	210	184	164	147	134	123	113
.0199	.0389	.0672	.1067	.1593	.2270	.3100	.4139	.5375	[illegible]
.0171	.0333	.0576	.0914	.1365	.1944	.2665	.3548	[illegible]	[illegible]
.0142	.0278	.0480	.0762	.1137	.1610	.2221	.2957	[illegible]	[illegible]

Carrying Capacity and Deflection of Helical Spri
Round Steel.—(*Continued*).

$d = 5/16''$	D	1.50	1.75	2.00	2.25	2.50	2.75	3.00	3.25	3.50
	W	605	500	426	371	329	295	267	245	226
	F	.0136	.0242	.0392	.0593	.0854	.1187	.1583	.2066	.2640
		.0117	.0207	.0336	.0508	.0732	.1012	.1357	.1771	.2263
		.0097	.0173	.0280	.0424	.0610	.0853	.1131	.1476	.1886
$d = 3/8''$	D	2.00	2.25	2.50	2.75	3.00	3.25	3.50	3.75	4.00
	W	765	663	589	523	473	433	398	368	343
	F	.0169	.0259	.0377	.0528	.0711	.0935	.1200	.1513	.1874
		.0145	.0222	.0323	.0452	.0610	.0801	.1029	.1297	.1606
		.0120	.0185	.0269	.0376	.0508	.0668	.0858	.1081	.1338
$d = 7/16''$	D	2.00	2.25	2.50	2.75	3.00	3.25	3.50	3.75	4.00
	W	1263	1089	957	853	770	702	644	596	544
	F	.0081	.0126	.0186	.0262	.0357	.0472	.0617	.0772	.0960
		.0069	.0108	.0160	.0225	.0306	.0405	.0529	.0661	.0823
		.0058	.0090	.0133	.0187	.0255	.0337	.0441	.0551	.0686
$d = 1/2''$	D	2.00	2.25	2.50	2.75	3.00	3.25	3.50	3.75	4.00
	W	1963	1683	1472	1309	1178	1071	982	906	841
	F	.0042	.0067	.0099	.0141	.0194	.0259	.0336	.0427	.0534
		.0036	.0057	.0085	.0121	.0167	.0222	.0288	.0360	.0457
		.0030	.0048	.0071	.0101	.0139	.0185	.0240	.0305	.0381
$d = 9/16''$	D	2.50	2.75	3.00	3.25	3.50	3.75	4.00	4.25	4.50
	W	2163	1916	1720	1560	1427	1315	1220	1137	1065
	F	.0056	.0081	.0112	.0151	.0197	.0252	.0316	.0390	.0474
		.0048	.0070	.0096	.0129	.0169	.0216	.0271	.0334	.0406
		.0040	.0058	.0080	.0108	.0141	.0180	.0225	.0278	.0339
$d = 5/8''$	D	2.50	2.75	3.00	3.25	3.50	3.75	4.00	4.25	4.50
	W	3068	2707	2422	2191	2001	1841	1704	1587	1484
	F	.0034	.0049	.0068	.0092	.0121	.0155	.0196	.0243	.0297
		.0029	.0042	.0058	.0079	.0104	.0133	.0168	.0208	.0254
		.0024	.0035	.0049	.0066	.0086	.0111	.0140	.0173	.0212
$d = 11/16''$	D	3.00	3.25	3.50	3.75	4.00	4.25	4.50	4.75	5.00
	W	3311	2988	2723	2500	2311	2151	2009	1885	1776
	F	.0043	.0058	.0077	.0100	.0127	.0157	.0193	.0233	.0279
		.0037	.0050	.0066	.0086	.0108	.0135	.0165	.0200	.0239
		.0030	.0042	.0055	.0071	.0090	.0112	.0138	.0167	.0199
$d = 3/4''$	D	3.00	3.25	3.50	3.75	4.00	4.25	4.50	4.75	5.00
	W	4418	3976	3615	3313	3058	2840	2651	2485	2339
	F	.0028	.0038	.0051	.0066	.0084	.0105	.0129	.0157	.0189
		.0024	.0033	.0044	.0057	.0072	.0090	.0111	.0135	.0162
		.0020	.0027	.0036	.0047	.0060	.0075	.0093	.0113	.0135
$d = 7/8''$	D	3.50	3.75	4.00	4.25	4.50	4.75	5.00	5.25	5.50
	W	6013	5490	5051	4676	4354	4073	3826	3607	3413
	F	.0021	.0027	.0035	.0045	.0055	.0067	.0081	.0097	.0115
		.0018	.0024	.0030	.0038	.0047	.0058	.0070	.0083	.0098
		.0015	.0020	.0025	.0032	.0039	.0048	.0058	.0069	.0082
$d = 1''$	D	3.50	3.75	4.00	4.25	4.50	4.75	5.00	5.25	5.50
	W	9425	8568	7854	7250	6732	6283	5890	5544	5236
	F	.0012	.0016	.0021	.0026	.0033	.0041	.0049	.0059	.0071
		.0010	.0014	.0018	.0023	.0028	.0035	.0043	.0051	.0061
		.0008	.0011	.0015	.0019	.0023	.0029	.0035	.0043	.0051

The formulæ for deflection or compression given by Clark, Ha
Begtrup, although very different in form, show a substantial
when ... he same form. Let d = diameter of wire in in
mea... oil, n the number of coils, w the applied
... icient, then

Compression or extension of one coil $= \dfrac{wD_1^3}{Cd^4}$;

ht in pounds to cause comp. or ext. of 1 in. $= \dfrac{Cd^4}{nD_1^3}$.

fficient C reduced from Hartnell's formula is $8 \times 180,000 = 1,440,000$; to Clark, $16^4 \times 22 = 1,441,792$, and according to Begtrup (using for the torsional modulus of elasticity) $= 12,000,000 \div 8 = 1,500,000$.

's formula for greatest safe extension, $v_1 = \dfrac{12.566nfr^2}{cd}$ may take

$v_1 = \dfrac{.7854nD_1^2}{100d}$ if we use 30,000 and 12,000,000 as the values for f ectively.

eral formulæ for safe load given above may be thus compared, = diameter of wire, and D_1 = mean diameter of coil, Rankine, $\dfrac{fd^3}{\ }$; Clark, $W = \dfrac{3(d \times 16)^3}{D_1}$; Begtrup, $W = \dfrac{.3927Sd^3}{D_1}$; Hartnell, $\dfrac{)0d^3}{r}$. Substituting for f the value 30,000 given by Rankine, and for as given by Begtrup, we have $W = 11,760 \dfrac{d^3}{D_1}$ Rankine; $12,288 \dfrac{d^3}{D_1}$

$562 \dfrac{d^3}{D_1}$ Begtrup; $24,000 \dfrac{d^3}{D_1}$ Hartnell.

from the Pennsylvania Railroad specifications the capacity when the following springs, in which d = diameter of wire, D diameter f coil, $D_1 = D - d$, c capacity, H height when free, and h height ed, all in inches.

d	D	D_1	c	H	h
$d = \frac{1}{4}$	$D = 1\frac{1}{2}$	$D_1 = 1\frac{1}{4}$	$c = 400$	$H = 9$	$h = 6$
$\frac{1}{2}$	3	$2\frac{1}{2}$	1,900	8	5
$\frac{3}{4}$	$5\frac{3}{4}$	5	2,100	7	$4\frac{1}{4}$
1	5	4	8,100	$10\frac{1}{2}$	8
$1\frac{1}{4}$	8	$6\frac{3}{4}$	10,000	9	$5\frac{3}{4}$
$1\frac{1}{8}$	$4\frac{7}{8}$	$3\frac{3}{4}$	16,000	$4\frac{3}{8}$	$3\frac{3}{8}$

ituting the values of c in the formula $c = W = x \dfrac{d^3}{D_1}$ we find x, the t of $\dfrac{d^3}{D_1}$ to be respectively 32,000; 38,000; 32,400; 24,888; 34,560; erage 34,000.

12,000 as the coefficient of $\dfrac{d^3}{D_1}$ according to Rankine and Clark for , and 24,000 as the coefficient according to Begtrup and Hartnell, for the safe load on these springs, as we take one or the other co-

	T.	*S.*	*K.*	*D.*	*I.*	*C.*
and Clark..............	150	600	1,012	3,000	3,750	5,400 lbs.
....	300	1,200	2,024	6,000	7,500	10,800 "
when closed, as above	400	1,900	2,100	8,100	10,000	16,000 "

loud (Trans. A. S. M. E., v. 173) gives the following:

$$P = \frac{S\pi d^3}{16R} \quad \text{and} \quad f = \frac{32PR^2l}{G\pi d^4};$$

P = load on spring;
S = maximum shearing fibre-strain in bar;
d = diameter of steel of which spring is made;
R = radius of centre of coil;
l = length of bar before coiling;
G = modulus of shearing elasticity;
f = deflection of spring under load.

nd takes $S = 80,000$ and $G = 12,600,000$.
ess in a helical spring is almost wholly one of torsion. For meth g the formulæ for springs from torsional formula see Mr. C *ove quoted.*

ELLIPTICAL SPRINGS, SIZES, AND PROOF TEST

Pennsylvania Railroad Specifications, 1889.

Class.	Length betw'n centres, in.	Width over all, inches.	Bands, inches.	Width of Plates, inches.	Tests. To stand ins. High.	With Lo of lbs.
A, Triple.......	40	11¾	3 × ⅝	3	3¾ between bands.	4800
					3 " "	5500
					2 " "	A. p. t.*
C, Quadruple..	40	15½	3 × ⅝	3	3¾ " "	6650
					3 " "	8000
					2 " "	A. p. t.*
D, Triple.... ..	36	11¾	3 × ⅝	3	4 " "	6000
					3 " "	8000
E, Single.......	40	sin.	3 × ⅝	3 × 11/32	5 bet. centre of eye and top of leaf.	When fre
					3 " "	2350
F, Triple.......	36	11¾	3 × ⅝	3 × 11/32	2½ between bands.	11,800
G, Double.	32	7½	3 × ⅝	3	6¼ " "	When fre
					3 " "	8000
H, Double......	36	9½	3 × ⅝	4	3½ " "	5400
					3 " "	6000
K, Double, 6 plates	22	10⅝	3½ × ⅝	4½ × 11/32	13/16 " "	13,800
L, Double, 7 plates	22	10⅝	3½ × ⅝	4½ × 11/32	13/16 " "	15,600
M, Quadruple..	40	15½	3 × ⅝	3	4 " "	8000
					3 " "	10,000
					2 " "	A. p. t.*

* A. p. t., auxiliary plates touching.

PHOSPHOR-BRONZE SPRINGS.

Wilfred Lewis (Engineers' Club, Philadelphia, 1887) made some tests phosphor-bronze wire, .12 in. diameter, coiled in the form of a spiral sp 1¼ in. diameter from centre to centre, making 52 coils.

This spring was loaded gradually up to a tension of 30 lbs., but as the was removed it became evident that a permanent set had taken p Such a spring of steel, according to the practice of the P. R. R., mig used for 40 lbs. A weight of 21 lbs. was then suspended so as to all small amount of vibration, and the length measured from day to day. hours the spring lengthened from 20⅝ inches to 21⅛ inches, and in 200 h to 21¼ inches. It was concluded that 21 lbs. was too great for durability, that probably 10 lbs. was as much as could be depended upon with safe

For a given load the extension of the bronze spring was just doubl extension of a similar steel spring, that is, for the same extension the spring is twice as strong.

SPRINGS TO RESIST TORSIONAL FORCE.

(Reuleaux's Constructor.)

Flat spiral or helical spring... $P = \frac{S}{6}\frac{bh^2}{R}$; $f = R\vartheta = 12\frac{PlR^2}{Ebh^3}$

Round helical spring ....$P = \frac{S\pi}{32}\frac{d^3}{R}$; $f = R\vartheta = \frac{64}{\pi}\frac{Pl}{E}\frac{R^2}{d^4}$

Round bar, in torsion.......... $P = \frac{S\pi}{16}\frac{d^3}{R}$; $f = R\vartheta = \frac{32}{\pi}\frac{P}{G}\frac{R^2l}{d^4}$.

Flat bar, in torsion............ $P = \frac{S}{3R}\frac{b^2h^2}{\sqrt{b^2+h^2}}$; $f = R\vartheta = \frac{3PR^2l}{G}\frac{b^2+}{b^3}$

P = force applied at end of radius or lever-arm R; ϑ = angular mot end of radius R; S = permissible maximum stress, = 4/5 of permi stress in flexure; E = modulus of elasticity in tension; G = torsional m = 2/5 E; l = developed length of spiral, or length of bar; d = dia re; b = breadth of flat bar; h = thickness.

... Specifications of Penna. R. R. Co., ... Arranged in Order of Strength.

P. R. R. Co's Class.	Bar. Diameter.	Bar. Length.	Bar. Length after Tapering.	Spring. Weight. Normal.	Spring. Weight. Minimum	Spring. Diam. Outside of Coil.	Spring. Height. Free.	Spring. Height. Closed.	Spring. Capacity.	Spring. Capacity, partly closed.
	inches.	inches.	inches.	lbs.	lbs.	inches.	inches.	inches.	lbs.	lbs. at height.
Z	9/64	57 1/2	....	1/4	7/32	1	5 3/4	3	130	110 at 3 1/4 in.
E	1/4	25 1/4	....	1/8	....	2 1/4	2 1/4	1 1/8	240	....
R	11/64	75	....	1/2	7/16	1	8	5	270	170 at 6
T	1/4	94 7/8	....	1 1/4	1 3/16	1 1/2	9	6	400	300 at 6 3/4
A	1/4	45 1/2	....	5/8	9/16	1 1/4	5 1/8	3 3/8	500	....
S	1/2	78 1/2	82 3/8	4 5/8	4 1/8	3	8	5	1,900	1,200 at 6
K	3/4	88 1/2	96 3/8	11	10 5/8	5 3/4	7	4 1/4	2,100	1,230 at 5 3/8
P	15/16	85	92 7/8	16 1/2	16	6	8	5	6,000	3,600 at 6
J	1	85	92	18 1/2	18	5 1/2	8 1/2	6	7,500	4,500 at 7
D	1	100 1/2	106 3/4	22	21 3/8	5	10 1/2	8	8,100	....
I	1 1/4	97 1/2	108	33 1/2	32 1/2	8	9	5 3/4	10,000	4,000 at 7 1/2
N	1 5/8	52 3/4 55	60 5/8 60 1/8	16 1/2	16	6 3 7/8	5 1/8	3 3/8	10,000	6,000 at 4
H	1 5/16	96	106 1/2	36 1/2	35 1/2	8	9	6	11,000	6,000 at 7 1/4
O	1 1/8 5/8	56 62	63 5/8 66 3/4	20 3/4	20	6 3 5/8	5 3/4	4 1/8	13,000	6,000 at 4 15/16
Q	1 5/16 13/16	96 99 3/4	106 1/2 106 1/2	51	48 1/2	8 5 1/8	9	6	14,000	6,800 at 7 1/2
C	1 1/8	35 3/8	41 1/4	9 7/8	9 1/2	4 7/8	4 3/8	3 3/8	16,000	....
L	1 3/8 7/8	87 85	97 3/8 91 1/2	50 1/2	49	8 5	8	5 3/4	16,000	7,000 at 7
X	1 1/4 3/4	75 1/4 72 1/4	83 1/8 76 3/4	35	34	6 1/4 3 5/8	8	6	19,000	7,000 at 7 1/4
Y	1 1/4 3/4	56 1/2 54 1/4	64 5/8 58 3/4	26 1/4	25 1/2	6 1/4 3 5/8	6	4 1/2	19,000	7,000 at 5 7/16
B	4 bars 15/16	46	52 3/8	35 1/2	34 1/2	5	5 1/8	3 7/8	28,000	13,000 at 4 1/2
U	4 " 15/16	56	62 3/8	43 1/4	42	5	6 1/8	4 5/8	28,000	13,500 at 5 3/8
V	4 " 1 1/16	51 3/4	58 3/8	51 1/2	50	5 5/16	6 1/8	4 5/8	42,000	16,700 at 5 1/2
	4 " 15/16 3 " 19/32	72 1/2 40 1/2	79 43	73	71	....	6 7/8 5 11/16 } 7 1/8	over plates 5 3/8	42,000	21,000 at 5 27/32
	" 1 1/16	61	67 1/2	60 1/2	59	5 5/16	7 1/8	5 3/8	42,000	18,500 at 6 5/16

Springs N, O, Q, L, X, and Y are made of two coils, one inside of the other.
Springs B, U, G, V, and W are made of four equal coils placed near together and joined by top and bottom cap-pieces.

RIVETED JOINTS.

Fairbairn's Experiments. (From Report of Committee o[n] Riveted Joints, *Proc. Inst. M. E.*, April. 1881.)

The earliest published experiments on riveted joints are contained i[n a] memoir by Sir W. Fairbairn in the Transactions of the Royal Society. [...] ing certain empirical allowances, he adopted the following ratios a[s ex]pressing the relative strength of riveted joints:

Solid plate	100
Double-riveted joint	70
Single-riveted joint	56

These well-known ratios are quoted in most treatises on riveting, an[d] still sometimes referred to as having a considerable authority. It is sing[ular] however, that Sir W. Fairbairn does not appear to have been aware tha[t the] proportion of metal punched out in the line of fracture ought to be diff[erent] in properly designed double and single riveted joints. These celeb[rated] ratios would therefore appear to rest on a very unsatisfactory analy[sis of] the experiments on which they were based.

Loss of Strength in Punched Plates.—A report by M[r.] Parker and Mr. John, made in 1878 to Lloyd's Committee, on the eff[ect of] punching and drilling, showed that thin steel plates lost comparatively [little] from punching, but that in thick plates the loss was very consider[able.] The following table gives the results for plates punched and not ann[ealed] or reamed:

Thickness of Plates.	Material of Plates.	Loss of Tenacity, per cent.
¼	Steel	8
⅜	"	18
½	"	26
¾	"	33
¾	Iron	18 to 23

The effect of increasing the size of the hole in the die-block is sho[wn in] the following table:

Total Taper of Hole in Plate, inches.	Material of Plates.	Loss of Tenacity du[e to] Punching, per cen[t.]
1-16	Steel	17.8
⅛	"	12.3
¼	"	(Hole ragged) 24.5

The plates were from 0.675 to 0.712 inch thick. When ⅝-in. punche[d holes] were reamed out to 1⅛ in. diameter, the loss of tenacity disappear[ed, and] the plates carried as high a stress as drilled plates. Annealing also r[estored] to punched plates their original tenacity.

Strength of Perforated Plates.

(P. D. Bennett, *Eng'g*, Feb. 12, 1886, p. 155.)

Tests were made to determine the relative effect produced upon [the] strength of a flat bar of iron or steel: 1. By a ¾-inch hole drilled to [re]quired size; 2, by a hole punched ⅛ inch smaller and then drilled [to the] size of the first hole; and, 3, by a hole punched in the bar to the size [of the] drilled bar. The relative results in strength per square inch of origin[al area] were as follows:

	1. Iron.	2. Iron.	3. Steel.	[4.] St[eel.]
Unperforated bar	1.000	1.000	1.000	1.[illegible]
Perforated by drilling	1.029	1.012	1.068	1.[illegible]
" " punching and drilling.	1.030	1.008	1.059	1.[illegible]
" " punching only	0.795	0.894	0.935	0.[illegible]

In tests 2 and 4 the holes were filled with rivets driven by hydraulic [pres]sure. T[illegible] strength per square inch caused by drilling is [illegible] re to that of the increased strength of a groov[ed] [illegible] r of sectional area equal to the smallest sect[ion] [illegible] nett's tests on an iron bar 0.84 in. diameter g, and a similar bar turned to 0.84 in. diameter at one point only, showed t the relative strength of the latter to the former was 1.323 to 1.000.

Riveted Joints.—Drilling versus Punching of Holes.

he Report of the Research Committee of the Institution of Mechanical gineers, on Riveted Joints (1881), and records of investigations by Prof. B. W. Kennedy (1881, 1882, and 1885), summarize the existing information arding the comparative effects of punching and drilling upon iron and el plates. From an examination of the voluminous tables given in Prosor Unwin's Report, the results of the greatest number of the experints made on iron and steel plates lead to the general conclusion that, ile thin plates, even of steel, do not suffer very much from punching, yet hose of ¼-inch thickness and upwards the loss of tenacity due to punchranges from 10% to 23% in iron plates, and from 11% to 33% in the case of d steel. In drilled plates there is no appreciable loss of strength. It is sible to remove the bad effects of punching by subsequent reaming or ealing; but the speed at which work is turned out in these days is not orable to multiplied operations, and such additional treatment is seldom ctised. The introduction of a practicable method of drilling the plating hips and other structures, after it has been bent and shaped, is a matter reat importance. If even a portion of the deterioration of tenacity can prevented, a much stronger structure results from the same material and same scantling. This has been fully recognized in the modern English ctice (1887) of the construction of steam-boilers with steel plates; punchin such cases being almost entirely abolished, and all rivet-holes being led after the plates have been bent to the desired form.

omparative Efficiency of Riveting done by Different Methods.

he Reports of Professors Unwin and Kennedy to the Institution of Meical Engineers (*Proc.* 1881, 1882, and 1885) tend to establish the four foling points:

That the shearing resistance of rivets is not highest in joints riveted by ans of the greatest pressure;

That the ultimate strength of joints is not affected to an appreciable nt by the mode of riveting; and, therefore,

That very great pressure upon the rivets in riveting is not the indispenle requirement that it has been sometimes supposed to be;

That the most serious defect of hand-riveted as compared with machineted work consists in the fact that in hand-riveted joints visible slip mences at a comparatively small load, thus giving such joints a low e as regards tightness, and possibly also rendering them liable to failure er sudden strains after slip has once commenced.

he following figures of mean results, taken from Prof. Kennedy's tables *oceedings* 1885, pp. 218–225), give a comparative view of hand and hylic riveting, as regards their ultimate strengths in joints, and the periods which in both cases visible slip commenced.

Total Breaking Load.		Load at which Visible Slip began.	
Hand-riveting.	Hydraulic Riveting.	Hand-riveting.	Hydraulic Riveting.
Tons.	Tons.	Tons.	Tons.
86.01	85.75	21.7	47.5
......	77.00	....	35.0
82.16	82.70	25.0	53.7
......	78.58	...	54.0
149.2	145.5	31.7	49.7
......	140.2	....	46.7
193.6	183.1	25.0	56.0
......	183.7	....	....

n these figures hand-riveting appears to be rather better than hydraulic ting, as far as regards ultimate *strength* of joint; but is very much inr to hydraulic *work, in view of the small proportion of load* be*efore visible slip commenced.*

Some of the Conclusions of the Committee of Re on Riveted Joints.

(*Proc. Inst. M. E.*, Apl. 1885.)

The conclusions all refer to joints made in soft steel plate rivets, the holes all drilled, and the plates in their natural state (un In every case the rivet or shearing area has been assumed to be t holes, not the nominal (or real) area of the rivets themselves. strength of the metal in the joint has been compared with that cut from the same plates, and not merely with nominally similar

The metal between the rivet-holes has a considerably greater sistance per square inch than the unperforated metal. This exces amounted to more than 20%, both in 3/8-inch and 3/4-inch plates, pitch of the rivet was about 1.9 diameters. In other cases 3/8-inch an excess of 15% at fracture with a pitch of 2 diameters, of 10% wi of 3.6 diameters, and of 6.6%, with a pitch of 3.9 diameters; and 3/4- gave 7.8% excess with a pitch of 2.8 diameters.

In single-riveted joints it may be taken that about 22 tons per sq is the shearing resistance of rivet steel, when the pressure on the r not exceed about 40 tons per square inch. In double-riveted jo rivets of about 3/4 inch diameter, most of the experiments gave abo per square inch as the shearing resistance, but the joints in one se at 22 tons.

The ratio of shearing resistance to tenacity is not constant, but d very markedly and not very irregularly as the tenacity increases.

The size of the rivet heads and ends plays a most important p strength of the joints—at any rate in the case of single-riveted jo increase of about one third in the weight of the rivets (all this in course, going to the heads and ends) was found to add about 8 resistance of the joint, the plates remaining unbroken at the full resistance of 22 tons per square inch, instead of tearing at a shear of only a little over 20 tons. The additional strength is probably prevention of the distortion of the plates by the great tensile str rivets.

The intensity of bearing pressure on the rivet exercises, with join tioned in the ordinary way, a very important influence on their So long as it does not exceed 40 tons per square inch (measured o jected area of the rivets), it does not seem to affect their strength; sures of 50 to 55 tons per square inch seem to cause the rivets to most cases at stresses varying from 16 to 18 tons per square inch dinary joints, which are to be made equally strong in plate and the bearing pressure should therefore probably not exceed 42 or 4 square inch. For double-riveted butt-joints perhaps, as will be n a higher pressure may be allowed, as the shearing stress may pro be more than 16 or 18 tons per square inch when the plate tears.

A margin (or net distance from outside of holes to edge of plate) eq diameter of the drilled hole has been found sufficient in all cases hith

To attain the maximum strength of a joint, the breadth of la such as to prevent it from breaking zigzag. It has been found th metal measured zigzag should be from 30% to 35% in excess of that straight across, in order to insure a straight fracture. This corre a diagonal pitch of $2/3\,p + d/3$, if p be the straight pitch and d eter of the rivet-hole.

Visible slip or "give" occurs always in a riveted joint at a p much below its breaking load, and by no means proportional to A collation of the results obtained in measuring the slip indicates pends upon the number and size of the rivets in the joint, rather t anything else; and that it is tolerably constant for a given size of given type of joint. The loads per rivet at which a joint will com slip visibly are approximately as follows:

Diameter of Rivet.	Type of Joint.	Riveting.	Slipping Riv
3/4 inch	Single-riveted	Hand	2.5 to
3/4 "	Double-riveted	Hand	3.0 to 3
3/4 "	Double-riveted	Machine	7 tons
1 inch	Single-riveted	Hand	3.2 to
1 "	Double-riveted	Hand	4.3 to
1	Double-riveted	Machine	8 to 10

f the probable load at which a joint of any breadth will commence nultiply the number of rivets in the given breadth by the proper ken from the last column of the table above. It will be understood above figures are not given as exact; but they represent very well lts of the experiments.

periments point to simple rules for the proportioning of joints of m strength. Assuming that a bearing pressure of 43 tons per square be allowed on the rivet, and that the excess tenacity of the plate its original strength, the following table gives the values of the ratios ter d of hole to thickness t of plate $(d \div t)$, and of pitch p to diamole $(p \div d)$ in joints of maximum strength in ⅜-inch plate.

For Single-riveted Plates.

l Tenacity of Plate.		Shearing Resistance of Rivets.		Ratio. $d \div t$	Ratio. $p \div d$	Ratio. Plate Area / Rivet Area
er	Lbs. per sq. in.	Tons per sq. in.	Lbs. per sq. in.			
	67,200	22	49,200	2.48	2.30	0.667
	62,720	22	49,200	2.48	2.40	0.785
	67,200	24	53,760	2.28	2.27	0.713
	62,720	24	53,760	2.28	2.36	0.690

ble shows that the diameter of the hole (not the diameter of the ould be 2⅓ times the thickness of the plate, and the pitch of the times the diameter of the hole. Also, it makes the mean plate area rivet area.

aller rivet be used than that here specified, the joint will not be of and therefore not of maximum, strength; but with any other size the best result will be got by use of the pitch obtained from the rmula

$$p = a\frac{d^2}{t} + d,$$

before, d is the diameter of the hole.

ue of the constant a in this equation is as follows:

For 30-ton plate and 22-ton rivets, a = 0.524
" 28 " 22 " " 0.558
" 30 " 24 " " 0.570
" 28 " 24 " " 0.606

mean, the pitch $p = 0.56\frac{d^2}{t} + d$.

ld be noticed that with too small rivets this gives pitches often consmaller in proportion than 2¼ times the diameter.

ouble-riveted lap-joints a similar calculation to that given ut with a somewhat smaller allowance for excess tenacity, on f the large distance between the rivet-holes, shows that for joints num strength the ratio of diameter to thickness should remain prein single-riveted joints; while the ratio of pitch to diameter of hole 3.64 for 30-ton plates and 22 or 24 ton rivets, and 3.82 for 28-ton th the same rivets.

till more than in the former case, it is likely that the prescribed vet may often be inconveniently large. In this case the diameter hould be taken as large as possible; and the strongest joint for a ckness of plate and diameter of hole can then be obtained by using given by the equation

$$p = a\frac{d^2}{t} + d,$$

e values of the constant a for different strengths of plates and y be taken as follows:

Table of Proportions of Double-riveted Lap-joints,

in which $p = a\dfrac{d^2}{t} + d.$

Thickness of Plate.	Original tenacity of Plate, Tons per sq. in.	Shearing Resistance of Rivets. Tons per sq. in.	Value of Constant. a
3/8 inch	30	24	1.15
3/8 "	28	24	1.22
3/8 "	30	22	1.05
3/8 "	28	22	1.12
3/4 "	30	24	1.17
3/4 "	28	24	1.25
3/4 "	30	22	1.07
3/4 "	28	22	1.14

Practically, having assumed the rivet diameter as large as possible, can fix the pitch as follows, for any thickness of plate from 3/8 to 3/4 inch

For 30-ton plate and 24-ton rivets } $p = 1.16\dfrac{d^2}{t} + d;$
" 28 " " " 22 " " }

" 30 " " " 22 " " $p = 1.06\dfrac{d^2}{t} + d;$

" 28 " " " 24 " " $p = 1.24\dfrac{d^2}{t} + d.$

In double-riveted butt-joints it is impossible to develop the shearing resistance of the joint without getting excessive bearing pres because the shearing area is doubled without increasing the area on w the pressure acts. Considering only the plate resistance and the bea pressure, and taking this latter as 45 tons per square inch, the best p would be about 4 times the diameter of the hole. We may probably with some certainty that a pressure of from 45 to 50 tons per square in the rivets will cause shearing to take place at from 16 to 18 tons per sq inch. Working out the equations as before, but allowing excess streng only 5% on account of the large pitch, we find that the proportions of do riveted butt-joints of maximum strength, under given conditions, are t of the following table:

Double-riveted Butt-joints.

Original Tenacity of Plate, Tons per sq. in.	Shearing Resistance of Rivets, Tons per sq. in.	Bearing Pressure, Tons per sq. in.	Ratio $\dfrac{d}{t}$	Ratio $\dfrac{p}{d}$
30	16	45	1.80	3.83
28	16	45	1.80	4.06
30	18	48	1.70	4.03
28	18	48	1.70	4.27
30	16	50	2.00	4.20
28	16	50	2.00	4.42

Practically, therefore, it may be said that we get a double-riveted butt of maximum strength by making the diameter of hole about 1.8 time thickness of the plate, and making the pitch 4.1 times the diameter of hole.

The proportions just given belong to joints of maximum strength. B a boiler the one part of the joint, the plate, is much more affected by than the other part, the rivets. It is therefore not unreasonable to esti the percentage by which the plates might be weakened by corrosion before the boiler would be unfit for use at its proper steam-pressure, an add correspondingly to the plate area. Probably the best thing to do i case is to proportion the joint, not for the actual thickness of plate, b a nominal thickness less than the actual by the assumed percentag this case the joint will be approximately one of uniform strength *time it has reached its final workable condition;* up to which time the *as a whole w[illegible]* have been weakened, the corrosion only grad *bringing t[illegible]* plates down to that of rivets.

Efficiencies of Joints.

erage results of experiments by the committee gave: For double-lap-joints in ⅜-inch plates, efficiencies ranging from 67.1% to 81.2%. ble-riveted butt-joints (in double shear) 61.4% to 71.3%. These low re-e probably due to the use of very soft steel in the rivets. For single-ap-joints of various dimensions the efficiencies varied from 54.8% to

periments showed that the shearing resistance of steel did not in-early so fast as its tensile resistance. With very soft steel, for of only 26 tons tenacity, the shearing resistance was about 80% of le resistance, whereas with very hard steel of 52 tons tenacity the resistance was only somewhere about 65% of the tensile resistance.

rtions of Pitch and Overlap of Plates to Diameter of Rivet-Hole and Thickness of Plate.

(Prof. A. B. W. Kennedy, *Proc. Inst. M. E.*, April, 1885.)

t = thickness of plate;
d = diameter of rivet (actual) in parallel hole;
p = pitch of rivets, centre to centre;
s = space between lines of rivets;
l = overlap of plate.

tch is as wide as is allowable without imparing the tightness of the ler steam.

gle-riveted lap-joints in the circular seams of boilers which have veted longitudinal lap joints,

$$d = t \times 2.25;$$
$$p = d \times 2.25 = t \times 5 \text{ (nearly)};$$
$$l = t \times 6.$$

uble-riveted lap-joints:

$$d = 2.25t;$$
$$p = 8t;$$
$$s = 4.5t;$$
$$l = 10.5t.$$

ngle-riveted Joints.			Double-riveted Joints.				
d	p	l	t	d	p	s	l
7-16	15-16	1⅛	3-16	7-16	1½	⅞	2
9-16	1¼	1½	¼	9-16	2	1 3-16	2¾
11-16	1 9-16	1⅞	5-16	11-16	2½	1⅛	3⅜
13-16	1⅞	2¼	⅜	13-16	3	1¾	4
1	2 3-16	2⅝	7-16	1	3½	2	4⅝
1⅛	2½	3	½	1⅛	4	2¼	5¼
1¼	2 13-16	3⅜	9-16	1¼	4½	2½	5⅞

hese proportions and good workmanship there need be no fear of of steam through the riveted joint.

t diagonal area, or area of plate, along a zigzag line of fracture ot be less than 30% in excess of the net area straight across the d 35% is better.

eodore Cooper (*R. R. Gazette*, Aug. 22, 1890) referring to Prof. Ken-atement quoted above, gives as a sufficiently approximate rule for er pitch between the rows in staggered riveting, one half of the the rivets in a row plus one quarter the diameter of a rivet-hole.

ent Excess in Strength of Perforated over Unperforated Plates. (Proc. Inst. M. E., October, 1888.)

erial between the rivet-holes has a considerably greater tensile re-per square inch than the unperforated metal. This excess tenacity d to more than 20%, both in ⅜-inch and ¾-inch plates, when the the rivets was about 1.9 diameters. In other cases ⅜-inch plate excess of 15% at fracture *with a pitch of* 2 diameters, of 10% with 1.8 diameters, *and of 6.6% with a pitch of* 3.9 diameters; and *re 7.8% excess with a pitch of 2.8 diameters.*

(1) The "excess strength due to perforation" is increased by a which tends to make the stress in the plate uniform, and to dimi effect of the narrow strip of metal at the edge of the specimen.

(2) It is diminished by increase in the ratio of p/d, of pitch to diam hole, so that in this respect it becomes less as the efficiency of t increases.

(3) It is diminished by any increase in hardness of the plate.

(4) For a given ratio p/d, of pitch to diameter of hole, it is also ap diminished as the thickness of the plate is increased. The ratio of thickness of plate does not seem to affect this matter directly, within the limits of the experiments.

Test of Double-riveted Lap and Butt Joints.

(Proc. Inst. M. E., October, 1888.)

Steel plates of 25 to 26 tons per square inch T. S., steel rivets of shearing-strength per square inch.

Kind of Joint.	Thickness of Plate.	Diameter of Rivet-holes.	Ratio of Pitch to Diameter.	Com Effic Jo
Lap........ ...	3/8″	0.8″	3.62	[illegible]
Butt...........	3/8	0.7	3.93	[illegible]
Lap......... ..	3/4	1.1	2.82	[illegible]
"	3/4	1.6	3.41	[illegible]
Butt	3/4	1.1	4.00	[illegible]
"	3/4	1.6	3.94	[illegible]
Lap........ ..	1	1.3	2.42	[illegible]
"	1	1.75	3.00	[illegible]
Butt	1	1.3	3.92	[illegible]

Some Rules which have been Proposed for the Dia of the Rivet in Single Shear. (*Iron*, June 18, 188

Browne................ .... $d = 2t$ (with double covers $1\frac{1}{4}t$)
Fairbairn.................. $d = 2t$ for plates less than 3/8 in.
" .... $d = 1\frac{1}{8}t$ for plates greater than 3/8 in.
Lemaitre................... $d = 1.5t + 0.16$
Antoine.................... $d = 1.1\sqrt{t}$
Pohlig...... .... $d = 2t$ for boiler riveting
" .... $d = 3t$ for extra strong riveting
Redtenbacher......... ...... $d = 1.5t$ to $2t$
Unwin....................... $d = \frac{3}{4}t + 5/16$ to $\frac{7}{8}t + \frac{3}{8}$
" .... $d = 1.2\sqrt{t}$

The following table contains some data of the sizes of rivets practice, and the corresponding sizes given by some of these rules

Diameter of Rivets for Different Thicknesses of P

Thickness of plate. Inches.	Diameter of Rivets, in inches.								
	Lloyd's Rules.	Liverpool Rules.	English Dock-yards.	French Veritas.	Browne Eq. (1).	Fairbairn (2 and (3).	Lemaitre (4).	Antoine (5).	Unwin (10).
5/16	5/8	5/8	1/2	......	5/8	5/8	5/8	5/8	11/16
3/8	5/8	5/8	5/8	5/8	3/4	3/4	23/32	11/16	3/4
7/16	5/8	3/4	3/4	5/8	7/8	21/32	13/16	3/4	13/16
1/2	3/4	13/16	3/4	......	1	3/4	15/16	3/4	7/8
9/16	3/4	13/16	7/8	3/4	1 1/8	27/32	1	13/16	7/8
5/8	3/4	7/8	7/8	......	1 1/4	15/16	1 1/8	7/8	15/16
11/16	7/8	7/8	7/8	13/16	...	1 1/32	1 3/16	15/16	1
3/4	7/8	15/16	1	7/8	....	1 1/8	1 1/4	15/16	1 1/16
13/16	7/8	1	[illegible]	......	....	1 7/32	1 3/8	1	1 3/32
7/8	1	[illegible]	[illegible]	[illegible]	..	......	......	1	1 1/8
15/16	1	[illegible]	[illegible]	[illegible]	...	......	......	1 1/16	1 3/16
1	1	[illegible]	[illegible]	1 1/16	....	......	......	1 1/8	1 1/4

ngth of Double-riveted Seams, Calculated. — W. B. s, Jr., in *Power* for June, 1890, gives tables of relative strength of ind parts of sheet between rivets in double-riveted seams, compared rength of shell, based on the assumption that the shearing strength ts and the tensile strength of steel are equal. The following figures ie sizes in his tables which show the nearest approximation to equal-trength of rivets and parts of plates between the rivets, together e percentage of each relative to the strength of the solid plate.

itch of vets, ches.	Size of Rivet-holes, inches.	Percentage of Strength of Plate.		Thickness of Plate, inches.	Pitch of Rivets, inches.	Size of Rivet-holes, inches.	Percentage of Strength of Plate.	
		Rivets.	Plate.				Rivets.	Plate.
2⅛	½	.739	.765	7/16	2¾	¾	.734	.728
2½	9/16	.795	.775	7/16	3⅛	13/16	.758	.740
3⅛	⅝	.785	.800	7/16	3⅝	⅞	.758	.759
3⅝	11/16	.819	.810	7/16	4⅛	15/16	.765	.773
2⅛	9/16	.749	.735	½	2½	¾	.707	.700
2⅝	⅝	.748	.762	½	2⅞	13/16	.721	.718
3⅛	11/16	.761	.780	½	3¼	⅞	.740	.731
3⅝	¾	.780	.793	½	3¾	15/16	.736	.750
2¼	⅝	.727	.722	½	4⅛	1	.761	.758
2⅝	11/16	.755	.738	9/16	2⅝	13/16	.701	.690
2⅞	¾	.754	.760	9/16	3	⅞	.714	.708
3⅝	13/16	.762	.776	9/16	3⅜	15/16	.727	.722
3⅛	⅞	.777	.788	9/16	3¾	1	.745	.733
3⅝	11/16	.714	.711	9/16	4¼	1 1/16	.742	.750

B. Parsons (*Am. Engr. & R. R. Jour.*, 1893) holds that it is an error to that the shearing strength of the rivet is equal to the tensile strength. 'erring to the apparent excess in strength of perforated over unper-plates, he claims that on account of the difficulty in properly match-holes, and of the stress caused by forcing, as is too often the case ice, this additional strength cannot be trusted much more than 'riction.

ing the sizes of iron rivets as generally used in American practice plates from ¼ to 1 inch thick: the tensile strength of the plates as s.; the shearing strength of the rivets as 40,000 for single-shear and r double-shear, Mr. Parsons calculates the following table of so that the strength of the rivets against shearing will be approxi-equal to that of the plate to tear between rivet-holes. The diameter vets has in all cases been taken at 1/16 in. larger than the nominal the rivet is assumed to fill the hole under the power riveter.

Riveted Joints.

P OR BUTT WITH SINGLE WELT—STEEL PLATES AND IRON RIVETS.

ess s.	Diameter of Rivets.	Pitch.		Efficiency.	
		Single.	Double.	Single.	Double.
	in.	in.	in.		
	½	1 3/16	1⅞	55 7%	70.0%
	¾	1 11/16	2 11/16	52.7	68.6
	¾	1⅞	2¾	49.0	65.9
	⅞	1 11/16	2 7/16	43.6	60.4
	1	1⅞	2⅝	42.0	59.5
	1	1¾	2 7/16	38.6	55.[illegible]
	1 1/8	2 3/16	2⅝	38.1	[illegible]

Calculated Efficiencies—Steel Plates and Steel R The differences between the calculated efficiencies given in the t above are notable. Those given by Mr. Ruggles are probably too hi he assumes the shearing strength of the rivets equal to the tensile of the plates. Those given by Mr. Parsons are probably lower tha obtained in practice, since the figure he adopts for shearing st rather low, and he makes no allowance for excess of strength of t rated over the unperforated plate. The following table has been c by th author on the assumptions that the excess strength of the p plate is 10%, and that the shearing strength of the rivets per squa four fifths of the tensile strength of the plate. If t = thickness d = diameter of rivet-hole, p = pitch, and T = tensile strength pe inch, then for single-riveted plates

$$(p - d)t \times 1.10T = \frac{\pi}{4}d^2 \times \frac{4}{5}T, \quad \text{whence } p = .571\frac{d^2}{t} + d.$$

$$\text{For double-riveted plates, } p = 1.142\frac{d^2}{t} + d.$$

The coefficients .571 and 1.142 agree closely with the averages given in the report of the committee of the Institution of Mecha gineers, quoted on pages 357 and 358, *ante*.

Thickness.	Diam. of Rivet-hole.	Pitch. Single Riveting.	Pitch. Double Riveting.	Efficiency. Single Riveting.	Efficiency. Double Riveting.
in.	in.	in.	in.	%	%
3/16	7/16	1.020	1.603	57.1	72.7
"	½	1.261	2.023	60.5	75.3
¼	½	1.071	1.642	53.3	69.6
"	9/16	1.285	2.008	56.2	72.0
5/16	9/16	1.137	1.712	50.5	67.1
"	⅝	1.339	2.053	53.3	69.5
"	11/16	1.551	2.415	55.7	71.5
⅜	⅝	1.218	1.810	48.7	65.5
"	¾	1.607	2.463	53.3	69.5
"	⅞	2.041	3.206	57.1	72.7
7/16	⅝	1.136	1.647	45.0	62.0
"	¾	1.484	2.218	49.5	66.2
"	⅞	1.869	2.864	53.2	69.4
"	1	2.305	3.610	56.6	72.3

Thickness.	Diam. of Rivet-hole.	Pitch. Single Riveting.	Pitch. Double Riveting.	Ef Single
in.	in.	in.	in.	
½	¾	1.292	[illegible]	46
"	⅞	1.749	2.624	50
"	1	2.142	3.284	53
"	1⅛	2.570	4.016	56
9/16	¾	1.321	1.892	43
"	⅞	1.652	2.429	47
"	1	2.015	3.030	50
"	1⅛	2.410	3.624	53
"	1¼	2.836	[illegible]	55
⅝	¾	1.264	1.778	40
"	⅞	1.575	2.274	41
"	1	1.914	[illegible]	47
"	1⅛	2.281	3.438	50
"	1¼	2.678	4.105	53

Riveting Pressure Required for Bridge and Bo Work.

(Wilfred Lewis, Engineers' Club of Philadelphia, Nov., 18

A number of ⅝-inch rivets were subjected to pressures between 60.000 lbs. At 10,000 lbs. the rivet swelled and filled the hole withou a head. At 20,000 lbs. the head was formed and the plates wer pinched. At 30,000 lbs. the rivet was well set. At 40,000 lbs. the m plate surrounding the rivet began to stretch, and the stretchin more and more apparent as the pressure was increased to 50,000 a lbs. From these experiments the conclusion might be drawn that sure required for cold riveting was about 300,000 lbs. per square inc section. In hot riveting, until recently there was never any call f sure exceeding 60,000 lbs., but now pressures as high as 150,000 lb *uncommon*, and even 200,000 lbs. have been contemplated as desir

t Shearing Resistance of Rivet Iron and Steel.

(*Proc. Inst. M. E.*, 1879, *Engineering*, Feb. 20, 1880.)

shearing resistance of the rivets cannot be ascertained from s on riveted joints (1), because the uniform distribution of the the rivets cannot be insured; (2) because of the friction of the h has the effect of increasing the apparent resistance to shear-lement uncertain in amount. Probably in the case of single-ts the shearing resistance is not much affected by the friction;

	Ultimate Shearing Stress Tons per sq in.	Lbs. per sq. in.	
shear (12 bars)..	24.15	54.096	Clarke.
e shear (8 bars)..	22.10	49.504	Clarke.
" ..	22.62	50.669	Barnaby.
" ..	22.30	49.952	Rankine.
rivets............	23.05 to 25.57	51.632 to 57.277	Riley.
rivets............	24.32 to 27.94	54.477 to 62.362	Riley.
value............	25.0	56.000	Riley.
rivets............	19.01	42.582	Greig and Eyth.
....	17 to 26	38.080 to 58.240	Parker.
eel, ¾-in. rivets..	31.67 to 33.69	70.941 to 75.466	Riley.
" ⅝-in. rivets...	30.45 to 35.73	68.208 to 80.035	Riley.
" mean value..	33.3	74.592	Riley.
eel................	22.18	49.683	Greig and Eyth.

's experiments show that a rivet is 6½% weaker in a drilled than ed hole. By rounding the edge of the rivet-hole the apparent esistance is increased 12%. Mr. Maynard found the rivets 4% drilled holes than in punched holes. But these results were ith riveted joints, and not by direct experiments on shearing. good deal of difficulty in determining the true diameter of a le, and it is doubtful whether in these experiments the diameter ccurately ascertained. Messrs. Greig and Eyth's experiments te a greater resistance of the rivets in punched holes than in s.

ears above, the apparent shearing resistance is less for double ngle shear, it is probably due to unequal distribution of the stress rivet sections.

ring resistance of a bar, when sheared in circumstances which ction, is usually less than the tenacity of the bar. The following w the decrease:

	Tenacity of Bar.	Shearing Resistance.	Ratio.
on.........	26.4	16.5	0.62
ron..........	25.4	20.2	0.79
Eyth, iron...	22.2	19.0	0.85
" steel..	28.8	22.1	0.77

r's researches (in 1870) the shearing strength of iron was found fifths of the tenacity. Later researches of Bauschinger confirm generally, but they show that for iron the ratio of the shearing and tenacity depends on the direction of the stress relatively to on of rolling. The above ratio is valid only if the shear is in a endicular to the direction of rolling, and if the tension is applied the direction of rolling. The shearing resistance in a plane the direction of rolling is different from that in a plane perpen-that direction, and again differs according as the plane of shear is ular or parallel to the breadth of the bar. In the former case the is 18 to 20% greater than in a plane perpendicular to the fibres, or the tenacity. In the latter case it is only half as great as in a endicular to the fibres.

IRON AND STEEL.

CLASSIFICATION OF IRON AND STEEL.

CLASSIFICATION OF IRON AND STEEL.

(W. Kent, *Railroad & Engineering Journal*, April, 1887.)

Generic Term.	IRON.				
How Obtained.	*CAST*, Or obtained from a fluid mass.			*WROUGHT*, Or welded from a pasty mass.	
Distinguishing Quality.	*Non-malleable.*	*Malleable.*		*Will Not Harden.*	*Will Harden.*
Species.	CAST IRON.		CAST STEEL.	(7) WROUGHT IRON.	(8†) WROUGHT STEEL.
Varieties.	(1) Ordinary castings.	(2) Malleable cast iron, obtained from No. 1 by annealing in oxides.	(3) Crucible, (4) Bessemer, and (5) Open-hearth steels. (6) Mitis.*	*a.* Obtained by direct process from ores, as Catalan, Chenot, and other process irons. *b.* Obtained by indirect process from cast iron, as finery-hearth and puddled irons.	Obtained by direct or indirect process, as German, shear, blister, and puddled steels.

* No. 6, Mitis is the name given to a new product (having the same general properties and produced by the same processes as soft cast steels) made by adding an alloy of aluminum to melted wrought iron or soft steel before pouring.

† No. 8, Wrought steel is almost an obsolete product, having been replaced in commerce by cast steel.

Sub-varieties of Nos. 3, 4, and 5, soft, mild, medium, and hard steels, according to percentage of carbon, the divisions

CAST IRON.

ng of Pig Iron.—Pig iron is commonly graded according to its the number of grades varying in different districts. In Eastern nia the principal grades recognized are known as No. 1 and 2 gray forge or No. 3, mottled or No. 4, and white or No. 5. Inter-rades are sometimes made, as No. 2 X, between No. 1 and No. 2, al names are given to irons more highly siliciзed than No. 1, as lver-gray, and soft. Charcoal foundry pig iron is graded by num-, but the quality is very different from the corresponding num-thracite and coke pig. Southern coke pig iron is graded into ten grades. Grading by fracture is a fairly satisfactory method of ons made from uniform ore mixtures and fuel, but is unreliable as f determining quality of irons produced in different sections or rent ores. Grading by chemical analysis, in the latter case, is the sfactory method. The following analyses of the five standard northern foundry and mill pig irons are given by J. M. Hartman *S. A.*, Feb., 1892):

	No. 1.	No. 2.	No. 3.	No. 4.	No. 4 B.	No. 5.
....	92.37	92.31	94.66	94.48	94.08	94.68
tic carbon...	3.52	2.99	2.50	2.02	2.02	.....
ned carbon...	.13	.37	1.52	1.98	1.43	3.83
....	2.44	2.52	.72	.56	.92	.41
orus......	1.25	1.08	.26	.19	.04	.04
r...........	.02	.02	trace	.08	.04	.02
nese.........	.28	.72	.34	.67	2.02	.98

CHARACTERISTICS OF THESE IRONS.

ray.—A large, dark, open-grain iron, softest of all the numbers xclusively in the foundry. Tensile strength low. Elastic limit ture rough. Turns soft and tough.

ray.—A mixed large and small dark grain, harder than No. 1 iron, exclusively in the foundry. Tensile strength and elastic limit an No. 1. Fracture less rough than No. 1. Turns harder, less more brittle than No. 1.

ray.—Small, gray, close grain, harder than No. 2 iron, used either ng-mill or foundry. Tensile strength and elastic limit higher than rns hard, less tough, and more brittle than No. 2.

ottled.—White background, dotted closely with small black spots e carbon; little or no grain. Used exclusively in the rolling-mill. ength and elastic limit lower than No. 3. Turns with difficulty; and more brittle than No. 3. The manganese in the B pig iron art of the combined carbon, making the iron harder and closing notwithstanding the lower combined carbon.

White.—Smooth, white fracture, no grain, used exclusively in the l. Tensile strength and elastic limit much lower than No. 4. Too rn and more brittle than No. 4.

pig irons are graded as follows, beginning with the highest in os. 1 and 2 silvery. Nos. 1 and 2 soft, all containing over 3% of s. 1, 2, and 3 foundry, respectively about 2 ¾%, 2 ¼% and 2% silicon; or "foundry forge;" No. 4 mill, or gray forge; mottled; white. arcoal chilling iron for car wheels contains, as a rule, 0.56 to 0.95 to 0.90 manganese, 0.65 to 0.75 phosphorus. The following is an f a remarkably strong car wheel: Si, 0.734; Mn, 0.498; P, 0.428, aphitic C, 2.068; Combined C, 1.247; Copper, 0.029. The chill was —½ in. deep at root of flange, ¼ in. deep on tread. A good iron analyzed: Si, 0.80; Graphitic C, 2.80; Combined C, 1.70; P, 3.35 (?). Its specific gravity was 7.22 and tenacity 31,734 lbs.

nce of Silicon, Phosphorus, Sulphur, and Man- upon Cast Iron.—W. J. Keep, of Detroit, in several papers I. M. E., 1889 to 1893), discusses the influence of various chemical n the quality of cast iron. From these the following notes have nsed:

—Pig iron contains all the carbon that it could absorb du[illegible] in the blast-furnace. Carbon exists in cast iron in tw[illegible] chemical union, as "combined" carbon, it cannot be [illegible] may increase the whiteness of the fracture, in so-ca[illegible] iron. Carbon mechanically mixed with the iron as graphite is v ing in color from gray to black, while the fracture of the iron ra light to a very dark gray.

Silicon will expel carbon, if the iron, when melted, contains bon that it can hold and a portion of silicon be added.

Prof. Turner concludes from his tests that the amount of silicon the maximum strength is about 1.80%. But this is only true wl base is used. If an iron is used as a base which will produce a so to begin with, each addition of silicon will decrease strength. Sili a weakening agent. Variations in the percentage of silicon adc iron will not insure a given strength or physical structure, but th will depend upon the physical properties of the original iron.

After enough silicon has been added to cause solid castings, addition and consequent increase of graphite weakens the ca softness and strength given to castings by a suitable addition is, by a further increase of silicon, changed to stiffness, brittl weakness.

As strength decreases from increase of graphite and decrease of carbon, deflection increases; or, in other words, bending is in graphite. When no more graphite can form and silicon still inc flection diminishes, showing that high silicon not only weakens makes it stiff. This stiffness is not the same strength-stiffnes caused by compact iron and combined carbon. It is a *brittle-stif*

In pig irons which received their silicon while in the blast-fu graphite more easily separates, and the shrinkage is less than i ture. As silicon increases, shrinkage also increases. Silicon o creases shrinkage, though by reason of its action upon the carbo nary practice it is truly said that silicon "takes the shrinkage ou iron." The slower a casting crystallizes, the greater will be th of graphite formed within it.

Silicon of itself, however small the quantity present, hardens but the decrease of hardness from the change of the combined graphite, caused by the silicon, is so much more rapid than the produced by the increase of silicon, that the total effect is to decr ness, until the silicon reaches from 3 to 5%.

As practical foundry-work does not call for more than 3% of s ordinary use of silicon does reduce the hardness of castings; but t duced through its influence on the carbon, and not its direct influe iron.

When the change from combined to graphite carbon has ceased ish hardness, say at from 2% to 5% of silicon, the hardening by i itself becomes more and more apparent as the silicon increases.

Shrinkage and hardness are almost exactly proportional. Wh varies, and other elements do not vary materially, castings with l age are soft; as shrinkage increases, the castings grow hard in not exactly, the same proportion. For ordinary foundry-practice of shrinkage may be made also the scale of hardness, provided va sulphur, and phosphorus especially, are not present to complica sult.

The term "chilling" irons is generally applied to such as, cool would be gray, but cooled suddenly, become white either to a de cient for practical utilization (*e.g.*, in car-wheels) or so far as to be tal. Many irons chill more or less in contact with the cold surfa mould in which they are cast, especially if they are thin. Somet is a valuable quality, but for general foundry purposes it is de have all parts of a casting an even gray.

Silicon exerts a powerful influence upon this property of irons, or entirely removing their capacity of chilling.

When silicon is mixed with irons previously low in silicon the increased.

It is not the percentage of silicon, but the state of the carbon action of silicon through other elements, which causes the iron to

Silicon irons have always had the reputation of imparting fluidit irons. This comes, no doubt, from the fact that up to 3% or 4% the the quantity of graphite in the resulting casting.

From the statement of Prof. Turner, that the maximum stren ... such a ... of silicon, and his statement that a fo iron, ... quality of iron that he may need, ing t... what he calls a typical foundry-l silicon of itself is not a softener or a lessener of shrinkage; ts influence on carbon, and only during a certain stage, does it effects.

s.—While phosphorus of itself, in whatever quantity present, -iron, yet in quantities less than 1.5% its influence is n t suffi- to overbalance other beneficial effects, which are exerted rcentage reaches 1%. Probably no element of itself weakens uch as phosphorus, especially when present in large quantities. s decreased when phosphorus is increased. All high-phosphorus e low shrinkage. Phosphorus does not ordinarily harden cast y for the reason that it does not increase combined carbon. of the metal is slightly increased by phosphorus, but not to t extent as has been ascribed to it.

y of remaining long in the fluid state must not be confounded for it is not the measure of its ability to make sharp castings, the very thin parts of a mould. Generally speaking, the state- fled that, to some extent, phosphorus prolongs the fluidity of it is filling the mould.

tch irons contained about 1% of phosphorus. The foundry-irons st sought for for small and thin castings in the Eastern States general thing, over 1% of phosphorus.

s which contain from 4% to 7% silicon have been so much used their ability to soften other irons that they have come to be ofteners" and as lesseners of shrinkage. These irons are valu- rs of silicon; but the irons which are sold most as softeners e-lesseners are those containing from 1% to 2% of phosphorus. refore ascribe the reputation of some of them largely to the nd not wholly to the silicon which they contain.

1% of phosphorus will do all that can be done in a beneficial above that amount weakens the iron, without corresponding s not necessary to search for phosphorus-irons. Most irons than is needed, and the care should be to keep it within limits. Only a small percentage of sulphur can be made to remain l iron, and it is difficult to introduce sulphur into gray cast iron arbonized iron, although gray cast iron often takes from the more sulphur as the iron originally contained. Percentages at could be retained by gray cast iron cannot materially injure pt through an increase of shrinkage. The higher the carbon, r the silicon, the smaller will be the influence exerted by

ce of sulphur on all cast iron is to drive out carbon and

tained than when no sulphur is present. Thus, in some tests on
quoted by R. Akerman, it is stated that in the foundry-iron fro
used in the manufacture of cannons, a percentage of 0.1% to 0.14
in the iron increased its strength to a considerable extent. The
of sulphur found originally in the iron put in the cupola is
further increased by part of the sulphur that is invariably found
used. It is seldom that a coke with a small percentage of sulph
whereas coke containing 1% of it and over is very common.
fuel in the cupola, if no special precautions are resorted to, the
of sulphur in the metal will in most cases be increased."

That the sulphur contents of pig iron may be increased by
contained in the coke used, is shown by some experiments in
reported by Mr. Nau. Seven consecutive heats were made.

The sulphur content of the coke was 1%, and 11.7% of fuel was
charge.

Before melting, the silicon ranged from 0.320 to 0.830 in the s
after melting, it was from 0.110 to 0.534, the loss in melting bei
to .375. The sulphur before melting was from .076 to .090, and a
from .132 to .174, a gain from .044 to .098.

From the results the following conclusions were drawn:

1. In all the charges, without exception, sulphur increased in
after its passage through the cupola. In some cases this inc
than doubled the original amount of sulphur found in the pig ir

2. The increase of the sulphur contents in the iron follows the
of a greater amount of silicon from that same iron. A larger
limestone added to these charges would have produced a more b
and undoubtedly less sulphur would have been incorporated in t

3. This coke contained 1% of sulphur, and if all its sulphur had
the iron there would have been an average increase of 0.12 of
the seven charges, while the real increase in the pig iron amou
0.081. This shows that two thirds of the sulphur of the coke wa
by the iron in its passage through the cupola.

MANGANESE.—Manganese is a nearly white metal, having abou
appearance when fractured as white cast iron. Its specific
about 8, while that of white cast iron, reasonably free from im
but a little above 7.5. As produced commercially, it is combine
and with small percentages of silicon, phosphorus, and sulphur.

It is generally produced in the blast-furnace. If the mangane
40%, with the remainder mostly iron, and silicon not over 0.50%,
called spiegeleisen, and the fracture will show flat reflecting sur
which it takes its name.

With manganese above 50%, the iron alloy is called ferro-mang

As manganese increases beyond 50%, the mass cracks in coolin
it approaches 98% the mass crumbles or falls in small pieces.

Manganese combines with iron in almost any proportion, bu
containing manganese is remelted, more or less of the manganese
by volatilization, and by oxidation with other elements present
If sulphur be present, some of the manganese will be likely to u
and escape, thus reducing the amount of both elements in the ca

Cast iron, when free from manganese, cannot hold more than
bon, and 3.50% is as much as is generally present; but as manganes
carbon also increases, until we often find it in spiegel as high as
ferro-manganese as high as 6%. This effect on capacity to hol
peculiar to manganese.

Manganese renders cast iron less plastic and more brittle.

Manganese increases the shrinkage of cast iron. An increase
the shrinkage 26%. Judging from some test records, mangane
influence chill at all; but other tests show that with a given pe
silicon the carbon may be a little more inclined to remain in th
form, and therefore the chill may be a little deeper. Hence, t
chill to be the same, it would seem that the percentage of silicon
a little higher with manganese than without it.

An increase of 1% of manganese increased the hardness 40%.
chill is required, manganese gives it by adding hardness to the wh

J. B. Nau (*Iron Age*, March 29, 1894), discussing the influence
nese on cast iron, says:

Manganese favors the combination between carbon and iron
ence, wh[illegible] sufficiently large quantities, is even great
only to [illegible] which would be naturally found in p

eases the capacity of iron to retain larger amounts of car- it all in the combined state.

n is often used for foundry purposes when some chill and ace is required in the casting. For the rolls of steel-rail put into the mixture a large amount of manganiferous iron, btained always presented the desired hardness of surface mottled structure on the outside. The inside, which al- h slower, was gray iron. One of the standard mixtures that good results was the following:

indry iron with 1.3% silicon and 1.5% manganese;
indry iron with 1% silicon and 1.5% manganese;
(rail ends) with about 0.35% to 0.40% carbon.

ng from this mixture contained about 1% of silicon and 1%

ıre, which differed but little from the preceding, was as

n with about 1.3% silicon and 1.5% manganese;
n with about 1% silicon and 1.5% manganese;
ottled iron with about 0.5% to 0.6% Si. and 1.2% Mn.
teel-rail ends with about 0.35% to 0.40% C. and 0.6% to 1% Mn.

sed in the preceding mixtures contained also invariably of phosphorus, so that the rolls obtained therefrom carried % of that element. The last mixture used produced rolls e average 0.8% to 1% of silicon and 1% of manganese. When- make those rolls from a mixture containing but 0.2% to 0.3% rolls were invariably of inferior quality, grayer, and con- . Manganese iron cannot be used indiscriminately for s. When greater softness is required in the castings man- avoided, but when hardness to a certain extent has to be nese iron can be used with advantage.

reases the magnetism of the iron. This characteristic in- percentage of manganese that enters into the composition iron loses all its magnetism when manganese reaches 25% ion. This peculiarity has been made use of by French draw a clear line between spiegel and ferro-manganese. ntains less than 25% of manganese it is classified as spiegel, tains more than 25 it% is classified as ferro-manganese. For ganese iron has to be avoided in castings of dynamo fields belonging to electric machinery, where magnetic conduc- the first considerations.

Distribution of Silicon in Pig Iron.—J. W. ge, Nov. 12, 1891) finds in analyzing samples taken from every ast of pig iron that the silicon varies considerably, the iron n the furnace having generally the highest percentage. In ts the silicon decreased from 2.040 to 1.713 from the first bed In another case the third bed had 1.260 Si., the seventh 1.718, 1.101. He also finds that the silicon varies in each pig, be- e point than at the butt. Some of his figures are: point of t of same 2.157; point of pig 1.834, butt of same 1.787.

of Cast Iron. (G. Lanza, *Trans. A. S. M. E.*, x., 187.)— ıalyses were as follows:

	Gun Iron, per cent.	Common Iron, per cent.
l carbon	3.51	.....
hite	2.80	.....
hur	0.133	0.173
phorus	0.155	0.413
on	1.140	1.89

mens were 26 inches long and square in section; those tested being very nearly one inch square, and those tested with d being cast nearly one and one quarter inches square, and ed down to one inch square.

		Tensile Strength.	Elastic Limit.	Modulus of Elasticity.
non.	20,200 to 23,000 T. S. Av.	= 22,066	6,500	13,194,233
n ...	20,300 to 26,800 " "	= 20,520	5,833	[illegible]
....	*27,000 to 28,775* " "	= 28,175	11,000	[illegible]
....	*29,500 to 31,000* " "	= 30,500	8,500	[illegible]

The el stic limit is not clearly defined in cast iron, the elongations i ing faster than the increase of the loads from the beginning of t The modulus of elasticity is therefore variable, decreasing as the lo crease. For example, the following results of a test of common ca reported by Prof. Lanza:

Lbs. per sq. in.	Elongation in 13.4 inches.	Sets, in.	Modulus of Elasticity.
1000	.0004	.....	18,217,400
2000	.0013	.....	16,777,700
3000	.0024	.....	14,085,400
4000	.0036	.....	13,101,200
5000	.0048	.....	12,809,200
6000	.0061	.0000	12,319,300
8000	.0088	.0001	11,600,800
10000	.0119	.0001	10,930 500
12000	.0162	.0007	9,714,200

CHEMISTRY OF FOUNDRY IRONS.

(C. A. Meissner, *Columbia College Q'ly*, 1890; *Iron Age*, 1890.)

Silicon is a very important element in foundry irons. Its tendency not above 2½% is to cause the carbon to separate out as graphite, givi casting the desired benefits of graphitic iron. Between 2½% and 3½% is best adapted for iron carrying a fair proportion of low silicon scra close iron, for ordinarily no mixture should run below 1½% silicon to ge castings.

From 3% to 5% silicon, as occurs in silvery iron, will carry heavy amou scrap. Castings are liable to be brittle, however, if not handled car as regards proportion of scrap used.

From 1½ %to 2% silicon is best adapted for machine work; will give clean castings if not much scrap is used with it.

Below 1% silicon seems suited for drills and castings that have to great variations in temperature.

Silicon has the effect of making castings fluid, strong, and open-gra also sound, by its tendency to separate the graphite from the total ca and consequent slight expansion of the iron on cooling, causing it to f thoroughly. Phosphorus, when high, has a tendency to make iron retain its heat longer, thereby helping to fill out all small spaces in ca It makes iron brittle, however, when above ¾% in castings. It is exc when high to use in a mixture of low-phosphorus irons, up to 1½% good results, but, as said before, the casting should be below ¾%. It strong tendency when above 1% in pig to make the iron less graphitic venting the separation of graphite.

Sulphur in open iron seldom bothers the founder, as it is seldom pr to any extent. The conditions causing open iron in the furnace caus sulphur. A little manganese is an excellent antidote against sulphur i furnace. Irons above 1% manganese seldom have any sulphur of any sequence.

Graphite is the all-important factor in foundry irons; unless this is pr in sufficient amount in the casting, the latter will be liable to be Graphite causes iron to slightly expand on cooling, makes it soft, toug fluid. (The statement as to expansion on cooling is denied by W. J. Ke

Relation of the Appearance of Fracture to the Chem Composition.—S. H. Chauvenet says when run [from the blas nace] the lower bed is almost always close grain, but shows practicall same analysis as the large grain in the rest of the cast. If the iron rapidly, the lower bed may have as large grain as any in the cast. I iron runs rapidly for, say, six beds and some obstruction in the tap causes the seventh bed to fill up slowly and sluggishly, this bed m close-grain, although the eighth bed, if the obstruction is removed, w open-grain. Neither the graphitic carbon nor the silicon seems to hav influence on the fracture in these cases, since by analysis the graphit silicon is the same in each. The question naturally arises whether it not be better to be guided by the analysis than by the fracture. The ture is a guide, but it is not an infallible guide. Should not the open *the close-*[illegible] m the same cast be numbered under the same *when th*[illegible] analysis?

Mr. [illegible] analyses made for the comparison of tw alysis, and unless the condition of furnace, whether the iron ran low, and from what part of pig bed the sample is taken, are known, ture is often very misleading. Take the following analyses:

	A.	B.	C.	D.	E.	F.
....	4.315	4.818	4.270	3.328	3.869	3.861
......	0.008	0.008	0.007	0.033	0.006	0.006
ic car..	3.010	2.757	2.680	2.243	3.070	3.100
arbon..	....	....	....	....	0.108	0.096

ry close-grain iron, dark color, by fracture, gray forge.
en-grain, dark color, by fracture, No. 1.
ry close-grain, by fracture, gray forge.
dium-grain, by fracture, No. 2, but much brighter and more open C, or F.
ry large, open-grain, dark color, by fracture, No. 1.
ry close-grain, by fracture, gray forge.
mparing analyses A and B, or E and F, it appears that the close-on is in each case the highest in graphitic carbon. Comparing A the graphite is about the same, but the close-grain is highest in

Analyses of Foundry Irons. (C. A. Meissner.)

SCOTCH IRONS.

Name.	Grade.	Silicon.	Phos-phorus.	Manga-nese.	Sul-phur.	Graph-ite.	Comb Car on.
rlee........	1	2.70	0.545	1.80	0.01	3.09	0.25
......	1	2.47	0.760	2.51	0.015		
......	1	3.44	1.000	1.70	0.015		
....	2	2.70	0.810	2.90	0.02	2.00	0.80
n..........	1	2.15	0 618	2.80	0.025	3.76	0.21
ss..........	1	2.59	0.840	1.70	0.010	3.75	3.75
oe.........	1	1.70	1.100	1.83	0.008	3.50	0.40
rnock.......	1	3.03	1.200	2.85			
rnock said rry ⅔ scrap	2	4.00	0.900	3.41	0.010	1.78	0.90

AMERICAN SCOTCH IRONS.

le	Silicon.	Phos-phorus.	Manganese	Sulphur.	No. Grade.	
	6.00	0.430	1.00	....	1	....
	1.67	1.920	1.90	....	....	casting
	2.40	1.000	1.70	....	2	....
	1.28	0.690	1.40	....	2	....
	3.50	0.613	2.51	....	1	....
	2.90	0.733	1.40	....	....	casting
	3.44	1.000	1.70	0.015	1	....
	3.35	1.300	1.50	0.012	1	....
	3.68	0.503	2.96	....	1	....

CRIPTION OF SAMPLES.—No. 1. Well known Ohio Scotch iro- ... y, but carries two-thirds scrap; made from part black-ba- ... ful brand. The high *silicon gives it its scrap-carrying* ... 2. Brier *Hill Scotch castings, made at scale works*; cast ... re *fluidity than strength.*

No. 3. Formerly a famous Ohio Scotch brand, not now in the market. Made mainly from black-band ore.

No. 4. A good Ohio Scotch, very soft and fluid; made from black-band ore-mixture.

Nos. 5*a* and 5*b*. Brier Hill Scotch iron and casting; made for stove purposes; 350 lbs. of iron used to 150 lbs. scrap gave very soft fluid iron; worked well.

No. 6*a*. Shows comparison between Summerlee (Scotch) (6*a*) and Brier Hill Scotch (6*b*). Drillings came from a Cleveland foundry, which found both irons closely alike in physical and working quality.

No. 7. One of the best southern brands, very hard to compete with, owing to its general qualities and great regularity of grade and general working.

MACHINE IRONS.

Sample No.	Silicon.	Phosphorus.	Manganese.	Sulphur.	Graphite.	Comb. Carbon.	Grade No.
8	2.80	0.492	0.61	0.015	....	....	1
9	1.30	0.262	0.70	0.030	....	....	3
10*a*	2.66	0.770	1.20	0.020	2.51	....	2
10*b*	3.63	0.411	1.25	0.014	3.05	....	1
11	2.10	0.415	0.60	0.050	....	....	2
12	1.37	0.294	1.51	0.080	2.31	0.78	2
13	3.10	0.124	trace	0.021	....	....	2
14	2.12	0.610	0.80	....	....	....	......
15	1.70	0.632	1.60	....	....	....	......
16*a*	1.45	0.470	1.25	0.009	....	....	2
16*b*	1.40	0.316	1.37	0.008	....	....	......
17	3.26	0.426	0.25	....	....	....	1
18	0.80	0.164	0.90	0.015	....	....	1

DESCRIPTION OF SAMPLES.—No. 8. A famous Southern brand noted for fi machine castings.

No. 9. Also a Southern brand, a very good machine iron.

Nos. 10*a* and 10*b*. Formerly one of the best known Ohio brands. Does not shrink; is very fluid and strong. Foundries having used this have reported very favorably on it.

No. 11. Iron from Brier Hill Co., made to imitate No. 3; was stronger than No. 3; did not pull castings; was fluid and soft.

No. 12. Copy of a very strong English machine iron.

No. 13. A Pennsylvania iron, very tough and soft. This is partially Bes-mer iron, which accounts for strength, while high silicon makes it soft.

No. 14. Castings made from Brier Hill Co.'s machine brand for scale works, very satisfactory, strong, soft and fluid.

No. 15. Castings made from Brier Hill Co.'s one half machine brand, one half Scotch brand, for scale works, castings desired to be of fair strength but very fluid and soft.

No. 16*a*. Brier Hill machine brand made to compete with No. 3.

No. 16*b*. Castings (clothes-hooks) from same, said to have worked badly, castings being white and irregular. Analysis proved that some other iron too high in manganese had been used, and probably not well mixed.

No. 17. A Pennsylvania iron, no shrinkage, excellent machine iron, soft and strong.

No. 18. A very good quality Northern charcoal iron.

"Standard Grades" of the Brier Hill Iron and Co Company.

Brier Hill Scotch Iron.—Standard Analysis, Grade Nos. 1 and 2.

Silicon	2.00 to 3.00
Phosphorus	0.50 to 0.75
Manganese	2.00 to 2.50

Used successfully for scales, mowing-machines, agricultural imple novelty hardware, sounding-boards, stoves, and heavy work requir special strength.

Brier Hill Silvery Iron.—Standard Analysis, Grade No. 1.

Silicon	3.50 to 5.50
Phosphorus	1.00 to 1.50
Manganese	2.00 to 2.25

Used successfully for hollow-ware, car-wheels, etc., stoves, bumpers, and ilar work, with heavy amounts of scrap in all cases. Should be mainly d where fluidity and no great strength is required, especially for heavy k. When used with scrap or close pig low in phosphorus, castings of siderable strength and great fluidity can be made

Fairly Heavy Machine Iron.—Standard Analysis, Grade No. 1.

Silicon	1.75 to 2.50
Phosphorus	0.50 to 0.60
Manganese	1.20 to 1.40

The best iron for machinery, wagon-boxes, agricultural implements, p-works, hardware specialties, lathes, stoves, etc., where no large ounts of scrap are to be carried, and where strength, combined with at fluidity and softness, are desired. Should not have much scrap with

Regular Machine Iron.—Standard Analysis, Grade Nos. 1 *and* 2.

Silicon	1.50 to 2.00
Phosphorus	0.30 to 0.50
Manganese	0.80 to 1.00

Used for hardware, lawn-mowers, mower and reaper works, oil-well chinery, drills, fine machinery, stoves, etc. Excellent for all small fine tings requiring fair fluidity, softness, and mainly strength. Cannot be ll used alone for large castings, but gives good results on same when used th above-mentioned heavy machine grade; also when used with the otch in right proportion. Will carry but little scrap, and should be used ne for good strong castings.

For Axles and Materials Requiring Great Strength, Grade No. 2.

Silicon	1.50
Phosphorus	0.200 and less.
Manganese	0.80

This gave excellent results.

A good neutral iron for guns, etc., will run about as follows:

Silicon	1.00
Phosphorus	0.25
Sulphur	0.20
Manganese	none.

It should be open No. 1 iron.

This gives a very tough, elastic metal. More sulphur would make tough t decrease elasticity.

For fine castings demanding elegance of design but no strength, phosorus to 3.00% is good. Can also stand 1.50% to 2.00% manganese. For work a hard, abrasive character manganese can run 2.00% in casting.

Analyses of Castings.

Sample No.	Silicon.	Phosphorus.	Manganese	Sulphur.	Graphite.	Comb. Carbon.
31	2.50	1.400	2.20	......	......	......
32	0.85	0.351	0.92	0.030	......	......
33	1.53	0.327	1.08	0.040	3.10	0.58
34a	1.84	0.577	1.04	......	......	......
34b	2.20	0.742	1.10	......	......	......
34c	2.50	1.208	1.16	......	......	......
35a	2.80	0.418	0.54	......	......	......
35b	3.10	1.280	1.14	......	......	......
35c	3.30	0.879	0.80	......	......	......
5d	2.88	*0.408*	*1.10*	......	......	......
[illegible]	*4.50*	*0.660*	*0.78*	......	......	......
	3.48	*1.439*	*0.90*	0.025	......	......
	2.65	*0.900*	*1.30*	......	......	......
	1.90	*0.960*	*1.20*	......	......	......

No. 31. Sewing-machine casting, said to be very fluid and good ca This is an odd analysis. I should say it would have been too hard an tle, yet no complaint was made.
No. 32. Very good machine casting, strong, soft, no shrinkage.
No. 33. Drillings from an annealer-box that stood the heat very well
No. 34*a*. Drillings from door-hinge, very strong and soft.
No. 34*b*. Drillings from clothes-hooks, tough and soft, stood severe mering.
No. 34*c*. Drillings from window-blind hinge, broke off suddenly at strain. Too high phosphorus.
No. 35*a*. Casting for heavy ladle support, very strong.
Nos. 35*b* and 35*c*. Broke after short usage. Phosphorus too high bumpers.
No. 35*d*. Elbow for steam heater, very tough and strong.
No. 36. Cog-wheels, very good, shows absolutely no shrinkage.
No. 37. Heater top network, requiring fluidity but no strength.
No. 37*a*. Gray part of above.
No. 37*b*. White, honeycombed part of above. Probably bad mixin got chilled suddenly.

STRENGTH OF CAST IRON.

Rankine gives the following figures:

Various qualities, T. S.	13,400 to	29,000,	average	16,5
Compressive strength	82,000 to	145,000,	"	112,0
Modulus of elasticity	14,000,000 to	22,900,000,	"	17,000,0

Specific Gravity and Strength. (Major Wade, 1856.)
Third-class guns: Sp. Gr. 7.087, T. S. 20,148. Another lot: least Sp. Gr. T. S. 22,402.
Second-class guns: Sp. Gr. 7.154, T. S. 24,767. Another lot : mean Sp 7.302, T. S. 27,232.
First class guns: Sp. Gr. 7.204, T. S. 28,805. Another lot: greatest Sp 7.402, T. S. 31,027.

Strength of Charcoal Pig Iron.—Pig iron made from Sali ores, in furnaces at Wassaic and Millerton, N. Y., has shown over 40,0 T. S. per square inch, one sample giving 42,281 lbs. Muirkirk, Md. tested at the Washington Navy Yard showed: average for No. 2 iron, lbs.; No. 3, 23,959 lbs.; No. 4, 41,329 lbs.; average density of No. 4, 7.336 I. W., v. p. 44.)

Nos. 3 and 4 charcoal pig iron from Chapinville, Conn., showed a t strength per square inch of from 34,761 lbs. to 41,882 lbs. Charcoal pi from Shelby, Ala. (tests made in August, 1891), showed a streng 34,800 lbs. for No. 3; No. 4, 39,675 lbs.; No. 5, 46,450 lbs.; and a mixtu equal parts of Nos. 2, 3, 4, and 5, 41,470 lbs. (*Bull. I. & S. A.*)

Variation of Density and Tenacity of Gun-irons.—A crease of density invariably follows the rapid cooling of cast iron, an general rule the tenacity is increased by the same means. The te generally increases quite uniformly with the density, until the latter as to some given point; after which an increased density is accompanie diminished tenacity.

The turning-point of density at which the best qualities of gun-iron their maximum tenacity appears to be about 7.30. At this point of de or near it, whether in proof-bars or gun-heads, the tenacity is greates

As the density of iron is increased its liquidity when melted is dimin This causes it to congeal quickly, and to form cavities in the interior casting. (Pamphlet of Builders' Iron Foundry, 1893.)

Specifications for Cast Iron for the World's Fair B ings, 1892.—Except where chilled iron is specified, all castings sh of tough gray iron, free from injurious cold-shuts or blow-holes, t pattern, and of a workmanlike finish. Sample pieces 1 in. square, cas the same heat of metal in sand moulds, shall be capable of sustainin clear span of 4 feet 6 inches a central load of 500 lbs. when tested rough bar.

Specifications for Tests of Cast Iron in 12" B. L. Mor (Pamphlet of Builders Iron Foundry, 1893.)—*Charcoal Gun Iron.*—The *strength of* [illegible] must average at each end at least 30,000 l *quare in*[illegible] to be over 37,000 lbs. per square inch; t *ecime*[illegible] may be as low as 28,000 lbs. per square inch be measured by the clean white portion to the point where gin to show in the white. The grades are to be by eighths of ¼, ⅜, ½, ⅝, ¾, ⅞, etc., until the iron is mottled; the lowest of an inch in depth of chill. The pigs of each cast are to b· the depth of chill shown by its test-piece, and each grade itself at the furnace and in forwarding.

f Cast Iron with Steel.—Car wheels are sometimes mixture of charcoal iron, anthracite iron, and Bessemer llowing shows the tensile strength of a number of tests of , the average tensile strength of the charcoal iron used being

	lbs. per sq. in.
on with 2½% steel	22,467
" " 3¾% steel	26,733
" " 6¼% steel and 6¼% anthracite	24,400
" " 7½% steel and 7½% anthracite	28,150
" " 2½% steel, 2½% wro't iron, and 6¼% anth.	25,550
" " 5 % steel, 5% wro't iron, and 10 % anth.	26,500

(*Jour. C. I. W.*, iii. p. 184.)

Partially Bessemerized.—Car wheels made of par- rized iron (blown in a Bessemer converter for 3½ minutes), ll-test mould over an inch deep, just as a test of cold-blast or car wheels would chill. Car wheels made of this blown 50,000 miles. (*Jour. C. I. W.*, vi. p. 77.)

Iron.—On October 15, 1891, the cast-iron fly-wheel of a large engines belonging to the Amoskeag Mfg. Co., of Manchester, l from centrifugal force. The fly-wheel was 30 feet diam- ches face, with one set of 12 arms, and weighed 116,000 lbs. ent, the rim castings, as well as the ends of the arms, were l of flaws, caused chiefly by the drawing and shrinking of the ens of the metal were tested for tensile strength, and varied per square inch in sound pieces to 1000 lbs. in spongy ones. flaws showed on the surface, and a rigid examination of the ey were erected failed to give any cause to suspect their true iments were carried on for some time after the accident in Company's foundry in attempting to duplicate the flaws, but s in approaching the badness of these castings.

MALLEABLE CAST IRON.

l cast iron, or malleable iron castings, are castings made t iron which have been subjected to a process of decarboni-

Rules for Use of Malleable Castings, by Committee of Ma Carbuilders' Ass'n, 1890.

1. Never run abruptly from a heavy to a light section.

2. As the strength of malleable cast iron lies in the skin, expose as m surface as possible. A star-shaped section is the strongest possible fr which a casting can be made. For brackets use a number of thin ribs inste of one thick one.

3. Avoid all round sections; practice has demonstrated this to be weakest form. Avoid sharp angles.

4. Shrinkage generally in castings will be 3/16 in. per foot.

Strength of Malleable Cast Iron.—Experiments on the stren of malleable cast iron, made in 1891 by a committee of the Master C builders' Association. The strength of this metal varies with the thickn as the following results on specimens from ¼ in. to 1½ in. in thickness sho

Dimensions.		Tensile Strength.	Elongation.	Elastic Limi
in.	in.	lb. per sq. in.	per cent in 4 in.	lb. per sq. in
1.52 by	.25	34,700	2	21,100
1.52 "	.39	33,700	2	15,200
1.53 "	.5	32,800	2	17,000
1.53 "	.64	32,100	2	19,400
2. "	.78	25,100	1½	15,400
1.51 "	.88	33,600	1½	19,300
1 06 "	1 02	30,600	1	17,000
1.28 "	1.3	27,400	1	
1.52 "	1 54	28,200	1½	

The low ductility of the metal is worthy of notice. The committee gi the following table of the comparative tensile resistance and ductility malleable cast iron, as compared with other materials:

	Ultimate Strength, lb. per sq. in	Comparative Strength; Cast Iron = 1.	Elongation Per Cent in 4 in.	Comparat Ductility Malleab Cast Iro = 1.
Cast iron	20,000	1	0.25	0.17
Malleable cast iron.	32,000	1.6	2.00	1
Wrought iron	50,000	2.5	20.00	10
Steel castings	60,000	3	10.00	5

Another series of tests, reported to the Association in 1892, gave following:

Thickness.	Width.	Area.	Elastic Limit.	Ultimate Strength.	Elongat in 8 in
in.	in.	sq in.	lb. per sq.	lb. per sq. in.	percen
.271	2.81	.7615	23,520	32,620	1.5
.293	2.78	.8145	22,650	28,100	.6
.39	2.82	1.698	20,595	32,060	1.5
.41	2.79	1.144	20,220	28,850	1.0
.529	2.76	1.46	19,520	27,875	1.1
.661	2.81	1.857	18,840	25,700	.7
.8	2.76	2.208	18,390	25,120	1.1
1.025	2.82	2.890	18,220	28,720	1.3
1.117	2.81	3.138	17,050	25,510	1.3
1.021	2.82	2.879	18,410	26,950	1.3

WROUGHT IRON.

fluence of Chemical Composition on the Properties Vrought Iron. (Beardslee on Wrought Iron and Chain Cables. Igement by W. Kent. Wiley & Sons, 1879.)—A series of 2000 tests of mens from 14 brands of wrought iron, most of them of high repute, made in 1877 by Capt. L. A. Beardslee, U.S.N., of the United States ng Board. Forty-two chemical analyses were made of these irons, a view to determine what influence the chemical composition had the strength, ductility, and welding power. From the report of these by A. L. Holley the following figures are taken :

nd.	Average Tensile Strength.	Chemical Composition. S.	P.	Si.	C.	Mn.	Slag.
	66,598	trace	{ 0.065	0.080	0.212	0.005	0.192
			{ 0.084	0.105	0.512	0.029	0.452
	54,363	{ 0.009	0.250	0.182	0.033	0.033	0.848
		{ 0.001	0.095	0.028	0.066	0.009	1.214
	52,764	0.008	0.231	0.156	0.015	0.017	....
	51,754	{ 0.003	0.140	0.182	0.027	trace	0.678
		{ 0.005	0.291	0.321	0.051	0.053	1.724
	51,134	{ 0.004	0.067	0.065	0.045	0.007	1.168
		{ 0.005	0.078	0.073	0.042	0.005	0.974
	50,765	0.007	0.169	0.154	0.042	0.021	....

ere two analyses are given they are the extremes of two or more ana- of the brand. Where one is given it is the only analysis. Brand L d be classed as a puddled steel.

ORDER OF QUALITIES GRADED FROM No. 1 TO No. 19.

and.	Tensile Strength.	Reduction of Area.	Elongation.	Welding Power.
L	1	18	19	most imperfect.
P	6	6	3	badly.
B	12	16	15	best.
J	16	19	18	rather badly.
O	18	1	4	very good.
C	19	12	16	——

reduction of area varied from 54.2 to 25.9 per cent, and the elonga- om 29.9 to 8.3 per cent.

nd O, the purest iron of the series, ranked No. 18 in tensile strength, as one of the most ductile; brand B, quite impure, was below the re both in strength and ductility, but was the best in welding power; quite impure, was one of the best in every respect except welding. L, the highest in strength, was not the most pure, it had the least ty, and its welding power was most imperfect. The evidence of the ice of chemical composition upon quality, therefore, is quite contra- y and confusing. The irons differing remarkably in their mechanical ties, it was found that a much more marked influence upon their es was caused by different treatment in rolling than by differences in sition.

egard to slag Mr. Holley says: "It appears that the smallest and vorked iron often has the most slag. It is hence reasonable to con- hat an iron may be dirty and yet thoroughly condensed."

s summary of "What is learned from chemical analysis," he says: r, it may appear that little of use to the makers or users of wrought as been learned. . . . The character of steel can be surely pred- on the analyses of the materials; that of wrought iron is altered by and unobserved causes."

uence of Reduction in Rolling from Pile to Bar on trength of Wrought Iron.—The tensile strength of the irons Beardslee's tests ranged from 46,000 to 62,700 lbs. per sq. in. ch was really a steel, not being considered. Some specim *ures as high as 70,000 lbs.* The amount of reduction of

area in rolling the bars has a notable influence on the strength and ela
limit; the greater the reduction from pile to bar the higher the streng
The following are a few figures from tests of one of the brands:

Size of bar, in. diam.:	4	3	2	1	½	
Area of pile, sq. in.:	80	80	72	25	9	
Bar per cent of pile:	15.7	8.83	4.36	3.14	2.17	1
Tensile strength, lb.:	46,322	47,761	48,280	51,128	52,275	59
Elastic limit, lb.:	23,430	26,400	31,892	36,467	39,126	—

Specifications for Wrought Iron (F. H. Lewis, Engineers' C of Philadelphia, 1891).—1. All wrought iron must be tough, ductile, fibr and of uniform quality for each class, straight, smooth, free from cin pockets, flaws, buckles, blisters, and injurious cracks along the edges, must have a workmanlike finish. No specific process or provision manufacture will be demanded, provided the material fulfils the requ ments of these specifications.

2. The tensile strength, limit of elasticity, and ductility shall be de mined from a standard test-piece not less than ¼ inch thick, cut from full-sized bar, and planed or turned parallel. The area of cross-section s not be less than ½ square inch. The elongation shall be measured a breaking on an original length of 8 inches.

3. The tests shall show not less than the following results:

	Ultimate Strength, lbs per sq. inch.	Limit of Elasticity, lbs. per sq. inch.	Elongatio 8 inches per cent
For bar iron in tension	50,000	26,000	18
For shape iron....................	48,000	26,000	15
For plates under 36 in. wide.....	48,000	26,000	12
For plates over 36 in. wide......	46,000	25,000	10

4. When full-sized tension members are tested to prove the strength their connections, a reduction in their ultimate strength of (500 × width bar) pounds per square inch will be allowed.

5. All iron shall bend, cold, 180 degrees around a curve whose diame is twice the thickness of piece for bar iron, and three times the thickn for plates and shapes.

6. Iron which is to be worked hot in the manufacture must be capa of bending sharply to a right angle at a working heat without sign fracture.

7. Specimens of tensile iron upon being nicked on one side and bent sl show a fracture nearly all fibrous.

8. All rivet iron must be tough and soft, and be capable of bending c until the sides are in close contact without sign of fracture on the con side of the curve.

Pennsylvania Railroad Specifications for Merchant B Iron or Steel.—Miscellaneous merchant bar iron or steel for which special specifications defining shapes and uses are issued, should hav tensile strength of 50,000 to 55,000 lbs. per square inch and an elongation 20% in a section originally 2 inches long.

No iron or steel will be accepted under this specification if tensile stren falls below 48,000 lbs. or goes above 60,000 lbs. per square inch, nor if el gation is less than 15% in 2 inches, nor if it shows a granular fracture cov ing more than 50% of the fractured surface, nor if it shows any difficulty welding.

In preparing test-pieces from round or rectangular bars, they will turned or shaped so that the tested sections may be the central portion the bar, in all sizes up to 1¾ inches in any diametrical or side measureme In larger sizes test-pieces will be made to fall about half-way from centre ircumference.

Bars of iron ½ in. thick or less, or tortured forms of iron, such as angle, *chann*[illegible] will be accepted if tensile strength is above 45,000 lbs. *ing*[illegible]; but the testing of such sizes and sections is optiona

ions for Wrought Iron for the World's Fair (*Eng'g News*, March 26, 1892.)—All iron to be used in the
rs of open trusses, laterals, pins and bolts, except plate iron
vide, and shaped iron, must show by the standard test-pieces
gth in lbs. per square inch of:

$$52{,}000 - \frac{7{,}000 \times \text{area of original bar in sq. in.}}{\text{circumference of original bar in inches}},$$

c limit not less than half the strength given by this formula,
tion of 20% in 8 in.

inches wide and under, and more than 8 inches wide, must
andard test-pieces a tensile strength of 48,000 lbs. per sq. in.
limit not less than 26,000 lbs. per square inch, and an elon-
ess than 12%. All plates over 24 inches in width must have a
h not less than 46,000 lbs., with an elastic limit not less than
square inch. Plates from 24 inches to 36 inches in width must
ation of not less than 10%; those from 36 inches to 48 inches in
r 48 inches in width, 5%.

on, flanges of beams and channels, and other iron not herein-
d, must show by the standard test-pieces a tensile strength in
inch of:

$$50{,}000 - \frac{7{,}000 \times \text{area of original bar}}{\text{circumference of original bar}},$$

limit of not less than half the strength given by this formula,
ion of 15% for bars 5/8 inch and less in thickness, and of 12% for
thickness. For webs of beams and channels, specifications
apply.

n must be tough and soft, and pieces of the full diameter of
be capable of bending cold, until the sides are in close contact,
f fracture on the convex side of the curve.

Iron.—Mr. Vauclain, of the Baldwin Locomotive Works,
of the American Railway Master Mechanics' Association, in
ny advocate the softest iron in the market as the best for
e believed in an iron as hard as was consistent with heading
. The higher the tensile strength of the iron, the more vibra-
and, for it is not so easily strained beyond the yield-point.
pecifications for stay-bolt iron call for a tensile strength of
lbs. per square inch, the upper figure being preferred, and
g insisted upon as the minimum.

ULÆ FOR UNIT STRAINS FOR IRON AND STEEL IN STRUCTURES.

H. Lewis, Engineers' Club of Philadelphia, 1891.)
ng formulæ for unit strains per square inch of net sectional
sed in determining the allowable working stress in each mem-
ucture. (For definitions of soft and medium steel see Specifi-
eel.)

Tension Members.

	Wrought Iron.	Soft Steel.	Medium Steel.
ngers or forged	Will not be used	Will not be used	7000
....	6000	" " "	7000
hangers s, riveted net sec-	5000	5500	7000
ams.....	8000	8000	Will not be used
members flanges net sec-	$7000\left(1+\frac{\text{min.}}{\text{max.}}\right)$	8% greater than iron	$9000\left(1+\frac{\text{min.}}{\text{max.}}\right)$
s.......	Will not be used	Will not be used	$9000\left(1+\frac{\text{min}}{\text{max.}}\right)$
ross-sec-	*15,000*	16,000	(For eyebars only, 17,0[illegible])

Shearing.

	Wrought Iron.	Soft Steel.	Medium Steel.
On pins and shop rivets	6000	6600	7200
On field rivets.........	4800	5200	Will not be used
In webs of girders.....	Will not be used	5000	6000

Bearing.

	Wrought Iron.	Soft Steel.	Medium Steel.
On projected semi-intrados of main-pin holes	12,000	13,200	14,500
On projected semi-intrados of rivet-holes*	12,000	13,200	14,500
On lateral pins	15,000	16,500	18,000
Of bed-plates on masonry	250 lbs. per sq. in.		

* Excepting that in pin-connected members taking alternate stresses, the bearing stress must not exceed 9000 lbs. for iron or steel.

Bending.

On extreme fibres of pins when centres of bearings are consider d as points of application of strains:

Wrought Iron, 15,000. Soft Steel, 16.000. Medium Steel, 17,000.

Compression Members.

	Wrought Iron.	Soft Steel.	Medium Steel.
Chord sections:			
Flat ends...	$7000\left(1+\frac{\text{min.}}{\text{max.}}\right)-30\frac{l}{r}$	10% greater than iron	20% greater than iron
One flat and one pin end..	$7000\left(1+\frac{\text{min.}}{\text{max.}}\right)-35\frac{l}{r}$	10% greater than iron	20% greater than iron
Chords with pin ends and all end-posts	$7000\left(1+\frac{\text{min.}}{\text{max.}}\right)-40\frac{l}{r}$	10% greater than iron	20% greater than iron
All trestle-posts...........	$7000\left(1+\frac{\text{min.}}{\text{max.}}\right)-35\frac{l}{r}$	10% greater than iron	20% greater than iron
Intermediate posts	$7500-40\frac{l}{r}$	10% greater than iron	20% greater than iron
Lateral struts, and compression in collision struts, stiff suspenders and stiff chords.........	$10{,}500-50\frac{l}{r}$	10% greater than iron	20% greater than iron

In which formulæ l = length of compression member in inches, and r = least radius of gyration of member in inches. No compression member shall have a length exceeding 45 times its least width, and no post should be used in which $l \div r$ exceeds 125.

Members Subject to Alternate Tension and Compression.

	Wrought Iron.	Soft Steel.	Medium Steel.
For compression only...	Use the formulæ above		
For the greatest stress..	$7000\left(1-\frac{\text{max. lesser}}{2\text{ max. greater}}\right)$	8% greater than iron	20% greater than iron

Use the formula giving the greatest area of section.

The compression flanges of beams and plate girders shall have the *cross-section* as the tension flanges.

Burr, discussing the formulæ proposed by Mr. Lewis, says: "Taking ts of experiments as a whole, I am constrained to believe that they at least 15% increase of resistance for soft-steel columns over those ght iron, with from 20% to 25% for medium steel, rather than 10% and ctively.

igh capacity of soft steel for enduring torture fits it eminently for and combined stresses, and for that reason I would give it 15% over iron, with about 22% for medium steel.

ing tests on steel seem to show that 15% and 22% increases, for the es respectively, are amply justified.

uld not hesitate to assign 15% and 22% increases over values for iron ng and bending of soft and medium steel as being within the safe experience. Provision should also be made for increasing pin-bending and bearing stresses for increasing ratios of fixed to mov-"

mum Permissible Stresses in Structural Materials Buildings. (Building Ordinances of the City of Chicago, 1893.) crushing stress: For plates, 15,000 lbs. per square inch; for lintels, or corbels, compression 13,500 lbs. per square inch, and tension per square inch. For girders, beams, corbels, brackets, and trusses, per square inch for steel and 12,000 lbs. for iron.

te girders:

$$\text{Flange area} = \frac{\text{maximum bending moment in ft.-lbs.}}{CD.}$$

stance between centre of gravity of flanges in feet.
3,500 for steel.
0,000 for iron.

$$\text{Web area} = \frac{\text{maximum shear}}{C}. \quad C = \begin{cases} 10,000 \text{ for steel,} \\ 6,000 \text{ for iron.} \end{cases}$$

ets in single shear per square inch of rivet area:

	Steel.	Iron.
shop-driven...	9000 lbs.	7500 lbs.
field-driven	7500 "	6000 "

ber girders:

$$S = \frac{cbd^2}{l}.$$

b = breadth of beam in inches.
d = depth of beam in inches.
l = length of beam in feet.
$c = \begin{cases} 160 \text{ for long-leaf yellow pine,} \\ 120 \text{ for oak,} \\ 100 \text{ for white or Norway pine.} \end{cases}$

ortioning of Materials in the Memphis Bridge (Geo. n, *Trans. A. S. C. E.*, 1893).—The entire superstructure of the Memge is of steel and it was all worked as steel, the rivet-holes being all principal members and punched and reamed in the lighter s.

sion members were proportioned on the basis of allowing the dead roduce a strain of 20,000 lbs. per square inch, and the live load a 10,000 lbs. per square inch. In the case of the central span, where load was twice the live load, this corresponded to 15,000 lbs. total r square inch, this being the greatest tensile strain.

mpression members were proportioned on a somewhat arbitrary o distinction was made between live and dead loads. A maximum 14,000 lbs. per square inch was allowed on the chords and other mpression members where the length did not exceed 16 times the nsverse dimension, this strain being reduced 750 lbs. for each addit of length. In long compression members the maximum length ted to 30 times the least transverse dimension, and the strains o 6,000 lbs. per square inch, this amount being increased by 200 lbs. unit by which the length is decreased.

ver reversals of strains occur the member was proportioned to resum of compression and tension on whichever basis (tension or ion) there would be the greatest strain per square inch; and, in the net section was proportioned to resist the maximum tension, cross section to resist the maximum compression.

or beams and girders were *calculated on the strain being limited to . per square inch in extreme fibres. Rivet-holes in cover-plates ere deducted.*

The rivets of steel in drilled or reamed holes were proportioned on th basis of a bearing strain of 15,000 lbs. per square inch and a shearing strai of 7500 lbs. per square inch, and special pains were taken to get the doubl shear in as many rivets as possible. This was the requirement for sho rivets. In the case of field rivets, the number was increased one-half.

The pins were proportioned on the basis of a bearing strain of 18,000 lbs per square inch and a bending strain of 20,000 lbs. per square inch in ex treme fibre, the diameters of the pins being never made more than one inch less than the width of the largest eye-bar attaching to them.

The weight on the rollers of the expansion joint on Pier II is 40,000 lbs per linear foot of roller, or 3,333 lbs. per linear inch, the rollers being 15 in in diameter.

As the sections of the superstructure were unusually heavy, and the strain from dead load greatly in excess of those from moving load, it was though best to use a slightly higher steel than is now generally used for lighte structures, and to work this steel without punching, all holes being drilled A somewhat softer steel was used in the floor-system and other lighte parts.

The principal requirements which were to be obtained as the results o tests on samples cut from finished material were as follows:

	Max. Ultimate Strength, lbs. per sq. inch.	Min. Ultimate Strength, lbs. per sq. inch.	Min. Elastic Limit, lbs. per sq. in.	Min. percentage of Elongation in 8 inches.	Min. Percentage of Reduction at Fractu
High-grade steel.	78,500	69,000	40,000	18	38
Eye-bar steel....	75,000	66,000	38,000	20	40
Medium steel....	72,500	64,000	37,000	22	44
Soft steel........	63,000	55,000	30,000	28	50

TENACITY OF METALS AT VARIOUS TEMPERATURES.

The British Admiralty made a series of experiments to ascertain what lo of strength and ductility takes place in gun-metal compositions when rais to high temperatures. It was found that all the varieties of gun-met suffer a gradual but not serious loss of strength and ductility up to a certa temperature, at which, within a few degrees, a great change takes plac the strength falls to about one half the original, and the ductility is whol gone. At temperatures above this point, up to 500, there is little, if an further loss of strength; the temperature at which this great change an loss of strength takes place, although uniform in the specimens cast fro the same pot, varies about 100° in the same composition cast at differe temperatures, or with some varying conditions in the foundry proce The temperature at which the change took place in No. 1 series was asce tained to be about 370°, and in that of No. 2, at a little over 250°. Whatev may be the cause of this important difference in the same composition, t fact stated may be taken as certain. Rolled Muntz metal and copper a satisfactory up to 500°, and may be used as securing-bolts with safet Wrought iron, Yorkshire and remanufactured, increase in strength up 500°, but lose slightly in ductility up to 300°, where an increase begins an continues up to 500°, where it is still less than at the ordinary temperatu of the atmosphere. The strength of Landore steel is not affected by tempe ature up to 500°, but its ductility is reduced more than one half. (*Iron*, Oc 6, 1877.)

Tensile Strength of Iron and Steel at High Tempera tures.—James E. Howard's tests (*Iron Age*, April 10, 1890), shows that th tensile strength of steel diminishes as the temperature increases from until a minimum is reached between 200° and 300° F., the total decrea being about 4000 lbs. per square inch in the softer steels, and from 6 8000 lbs. in steels of over 80,000 lbs. tensile strength. From this minimum the strength increases up to a temperature of 400° to 650° F., the maxi being r... earlier in the harder steels, the increase amounting to 2,00... per square inch above the minimum strength at from

m this maximum, the strength of all the steel decreases steadily proximating 10,000 lbs. decrease per 100° increase of tempera- ength of 20,000 lbs. per square inch is still shown by .10 C. steel 0° F., and by .60 to 1.00 C. steel at about 1600° F.
th of wrought iron increases with temperature from 0° up to a t from 400 to 600° F., the increase being from 8000 to 10,000 lbs. nch, and then decreases steadily till a strength of only 6000 lbs. nch is shown at 1500° F.
appears to maintain its strength, with a tendency to increase, reached, beyond which temperature the strength gradually Under the highest temperatures, 1500° to 1600° F., numerous e cylindrical surface of the specimen were developed prior to is remarkable that cast iron, so much inferior in strength to the mospheric temperature, under the highest temperatures has ame strength the high-temper steels then have.

h of Iron and Steel Boiler-plate at High Tem- (Chas. Huston, *Jour. F. I.*, 1877.)

AVERAGE OF THREE TESTS OF EACH.

ure F.	68°	575°	925°
ron plate, tensile strength, lbs.......	55.366	63,080	65,343
" " contr. of area %...........	26	23	21
hearth steel, tensile strength, lbs.....	54,600	66,083	64,350
" " contr. %	47	38	33
ble steel, tensile strength, lbs........	64,000	69,266	68,600
" contr. %.....................	36	30	21

h of Wrought Iron and Steel at High Temper- *Jour. F. I.*, cxii., 1881, p. 241.) Kollmann's experiments at Ober- nded tests of the tensile strength of iron and steel at tempera- g between 70° and 2000° F. Three kinds of metal were tested, iron having an ultimate tensile strength of 52,464 lbs., an elastic 38,280 lbs., and an elongation of 17.5%; fine-grained iron having e elements values of 56,892 lbs., 39,113 lbs., and 20%; and Bes- having values of 84,826 lbs., 55,029 lbs., and 14.5%. The mean sile strength of each material expressed in per cent of that at nospheric temperature is given in the following table, the fifth hich exhibits, for purposes of comparison, the results of experi- ed on by a committee of the Franklin Institute in the years

ure F.	Fibrous Wrought Iron, p. c.	Fine-grained Iron, per cent.	Bessemer Steel, per cent.	Franklin Institute, per cent.
	100.0	100.0	100.0	96.0
	100.0	100.0	100.0	102.0
	100.0	100.0	100.0	105.0
	97.0	100.0	100.0	106.0
	95.5	100.0	100.0	106.0
	92.5	98.5	98.5	104.0
	88.5	95.5	92.0	99.5
	81.5	90.0	68.0	92.5
	67.5	77.5	44.0	75.5
	44.5	51.5	36.5	53.5
	26.0	36.0	31.0	36.0
	20.0	30.5	26.5	.....
	18.0	28.0	22.0	.....
	16.5	23.0	18.0	.....
	13.5	19.0	15.0	.....
	10.0	15.5	12.0	.....
	7.0	12.5	10.0	.....
	5.5	10.5	8.5	.....
	4.5	8.5	7.5	.....
	3.5	7.0	6.5	.....
	3.5	5.0	5.0	.....

ect of Cold on the Strength of Iron and Steel.— g conclusions were arrived at by Mr. Styffe in 1865:
e absolute strength of iron and steel is not diminish t even at the *lowest temperature* which ever occurs in *s great as at the ordinary* temperature (about 60° F.).

(2) That neither in steel nor in iron is the extensibility less in sever
than at the ordinary temperature.

(3) That the limit of elasticity in both steel and iron lies higher in s
cold.

(4) That the modulus of elasticity in both steel and iron is increas
reduction of temperature, and diminished on elevation of temperature
that these variations never exceed 0.05 % for a change of temperature o
F., and therefore such variations, at least for ordinary purposes, are
special importance.

Mr. C. P. Sandberg made in 1867 a number of tests of iron rails at va
temperatures by means of a falling weight, since he was of opinion
although Mr. Styffe's conclusions were perfectly correct as regards te
strength, they might not apply to the resistance of iron to impact at
temperatures. Mr. Sandberg convinced himself that "the breaking str
of iron, such as was usually employed for rails, "as tested by sudden
or shocks, is considerably influenced by cold; such iron exhibiting at
only from one third to one fourth of the strength which it possess
84° F." Mr. J. J. Webster (Inst. C. E., 1880) gives reasons for dou
the accuracy of Mr. Sandberg's deductions, since the tests at the l
temperature were nearly all made with 21-ft. lengths of rail, while tho
the higher temperatures were made with short lengths, the suppor
every case being the same distance apart.

W. H. Barlow (Proc. Inst. C. E.) made experiments on bars of wro
iron, cast iron, malleable cast iron, Bessemer steel, and tool steel. The
were tested with tensile and transverse strains, and also by impact
half of them at a temperature of 50° F., and the other half at 5° F.
lower temperature was obtained by placing the bars in a freezing mix
care being taken to keep the bars covered with it during the whole tim
the experiments.

The results of the experiments were summarized as follows:

1. When bars of wrought iron or steel were submitted to a tensile s
and broken, their strength was not affected by severe cold (5° F.), but
ductility was increased about 1% in iron and 3% in steel.

2. When bars of cast iron were submitted to a transverse strain at a
temperature, their strength was diminished about 3% and their flexib
about 16%.

3. When bars of wrought iron, malleable cast iron, steel, and ordi
cast iron were subjected to impact at a temperature of 5° F., the forc
quired to break them, and the extent of their flexibility, were reduce
follows, viz.:

	Reduction of Force of Impact, per cent.	Reduction of F bility, per cen
Wrought iron, about	3	18
Steel (best cast tool), about	3½	17
Malleable cast iron, about	4½	15
Cast iron, about	21	not taken

The experience of railways in Russia, Canada, and other countries w
the winter is severe is that the breakages of rails and tires are far m
numerous in the cold weather than in the summer. On this accou
softer class of steel is employed in Russia for rails than is usual in t
temperate climates.

The evidence extant in relation to this matter leaves no doubt that
capability of wrought iron or steel to resist impact is reduced by cold.
the other hand, its static strength is not impaired by low temperatures.

Effect of Low Temperatures on Strength of Railr Axles. (Thos. Andrews, Proc. Inst. C. E., 1891.)—Axles 6 ft. 6 in.
between centres of journals, total length 7 ft. 3½ in., diameter at middl
in., at wheel-sets 5¼ in., journals 3¾ × 7 in. were tested by impact at tem
atures of 0° and 100° F. Between the blows each axle was half turned o
and was also replaced for 15 minutes in the water-bath.

The mean force of concussion resulting from each impact was ascerta
as follows:

Let h = height of free fall in feet, w = weight of test ball, hw =
"energy," or work in foot-tons, x = extent of deflections between bear

$$\text{then } F \text{ (mean force)} = \frac{W}{x} = \frac{hw}{x}.$$

lts of these experiments show that whereas at a temperature of tal average mean force of 1.79 tons was sufficient to cause the of the axles, at a temperature of 100° F. a total average mean 8 tons was requisite to produce fracture. In other words, the re- concussion of the axles at a temperature of 0° F. was only about t it was at a temperature of 100° F.

rage total deflection at a temperature of 0° F. was 6.48 in., as .06 in. with the axles at 100° F. under the conditions stated; this an ultimate reduction of flexibility, under the test of impact, of for the cold axles at 0° F., compared with the warm axles at

PANSION OF IRON AND STEEL BY HEAT.

. Howard, engineer in charge of the U. S. testing-machine at Wa- lass., gives the following results of tests made on bars 35 inches *Age*, April 10, 1890):

al.	Marks.	Chemical composition.				Coefficient of Expansion.
		C.	Mn.	Si.	Fe by difference.	Per degree F. per unit of length.
iron.....	......	......	......	......	......	.0000067302
....	1a	.09	.11	......	99.80	.0000067561
....	2a	.20	.45	......	99.35	.0000066259
....	3a	.31	.57	......	99.12	.0000065149
....	4a	.37	.70	......	98.93	.0000066597
....	5a	.51	.58	.02	98.89	.0000066202
....	6a	.57	.93	.07	98.43	.0000063891
....	7a	.71	.58	.08	98.63	.0000064716
....	8a	.81	.56	.17	98.46	.0000062167
....	9a	.89	.57	.19	98.35	.0000062335
....	10a	.97	.80	.28	97.95	.0000061700
) iron....	......	......	......	......	......	.0000059261
pper.....	......	......	......	......	......	.0000091286

URABILITY OF IRON, CORROSION, ETC.

ility of Cast Iron.—Frederick Graff, in an article on the ia water-supply, says that the first cast-iron pipe used there was). These pipes were made of charcoal iron, and were in constant years. They were uncoated, and the inside was well filled with In salt water good cast iron, even uncoated, will last for a cen- st; but it often becomes soft enough to be cut by a knife, as is ron cannon taken up from the bottom of harbors after long sub- Close-grained, hard white metal lasts the longest in sea water.— s, April 23, 1887, and March 26, 1892.

of Iron after Forty Years' Service.—A square link 12 ad, 1 inch thick and about 12 feet long was taken from the Kieff n 40 years old, and tested in comparison with a similar link which preserved in the stock-house since the bridge was built. The fol- the record of a mean of four longitudinal test-pieces, 1 × 1¼ × 8 ten from each link (*Stahl und Eisen*, 1890):

	Old Link taken from Bridge.	New Link from Store-house.
strength per square inch, tons	21.8	22.2
limit " "	11.1	11.9
ation, per cent.........................	14.05	13.42
action, per cent................	17.35	18.75

ility of Iron in Bridges. (G. Lindenthal, *Eng'g*, May 2,).—The Old Monongahela suspension bridge in Pittsburgh, built as taken down in 1882. *The wires of the cables were frequ half of their ultimate strength*, yet on testing them after 37 use they showed a tensile strength of from 72,700 to 100,000 lbs. pe inch. The elastic limit was from 67,100 to 78,600 lbs. per square in duction at point of fracture, 35% to 75%. Their diameter was 0.13 in

A new ordinary telegraph wire of same gauge tested for com showed: T. S., of 100,000 lbs.; E. L., 81,550 lbs.; reduction, 57%. I used as stays or suspenders showed: T. S., 43,770 to 49,720 lbs. per inch; E. L., 26,380 to 29,200. Mr. Lindenthal draws these conclusio his tests:

"The above tests indicate that iron highly strained for a long nu years, but still within the elastic limit, and exposed to slight vibra not deteriorate in quality.

"That if subjected to only one kind of strain it will not change its even if strained beyond its elastic limit, for many years. It will str behave much as in a testing-machine during a long test.

"That iron will change its texture only when exposed to alternat straining, as in bending in different directions. If the bending is sl very rapid, as in violent vibrations, the effect is the same."

Corrosion of Iron Bolts.—On bridges over the Thames in bolts exposed to the action of the atmosphere and rain-water we away in 25 years from a diameter of ⅞ in. to ½ in., and from ⅝ in. d to 5/16 inch.

Wire ropes exposed to drip in colliery shafts are very liable to cor

Corrosion of Iron and Steel.—Experiments made at the R Iron Works, Wheeling, W. Va., on the comparative liability to rus and soft Bessemer steel: A piece of iron plate and a similar piece both clean and bright, were placed in a mixture of yellow loam a with which had been thoroughly incorporated some carbonate of soda of soda, ammonium chloride, and chloride of magnesium. The prepared was kept moist. At the end of 33 days the pieces of me taken out, cleaned, and weighed, when the iron was found to have l of its weight and the steel 0.72%. The pieces were replaced and after weighed again, when the iron was found to have lost 2.06% of its weight and the steel 1.79%. (*Eng'g*, June 26, 1891.)

Corrosive Agents in the Atmosphere.—The experime Crace Calvert (*Chemical News*, March 3, 1871) show that carbonic the presence of moisture, is the agent which determines the oxid iron in the atmosphere. He subjected perfectly cleaned blades of steel to the action of different gases for a period of four mont results as follows:

Dry oxygen, dry carbonic acid, a mixture of both gases, dry an oxygen and ammonia: no oxidation. Damp oxygen: in three exp one blade only was slightly oxidized.

Damp carbonic acid: slight appearance of a white precipitate iron, found to be carbonate of iron. Damp carbonic acid and oxidation very rapid. Iron immersed in water containing carbo oxidized rapidly.

Iron immersed in distilled water deprived of its gases by boilin the iron in spots that were found to contain impurities.

Galvanic action is a most active agent of corrosion. It tak when two metals, one electro-negative to the other, are placed in and exposed to dampness.

Sulphurous acid (the product of the combustion of the sulphur in an exceedingly active corrosive agent, especially when the expose coated with soot. This accounts for the rapid corrosion of iron in bridges exposed to the smoke from locomotives. (See account of ments by the author on action of sulphurous acid in *Jour. Frank. Ins* 1875, p. 437.) An analysis of sooty iron rust from a railway bridge the presence of sulphurous, sulphuric, and carbonic acids, chlor ammonia. Bloxam states that ammonia is formed from the nitrog air during the process of rusting.

Rustless Coatings for Iron and Steel.—Tinning, enamel quering, galvanizing, electro-chemical painting, and other pres methods are discussed in two important papers by M. P. Wood, i A. S. M. E., vols. xv and xvi.

A Method of Producing an Inoxidizable Surf iron and steel by means of electricity has been developed by M. A. tens (*Engineering*.) The article to be protected is placed in a bath ary or distilled water, at a temperature of from 158° to 176° F. ectric cur ent through. The water is decomposed into its and hydrogen, and the oxygen is deposited on the metal, while the en appears at the other pole, which may either be the tank in which ration is conducted or a plate of carbon or metal. The current has fficient electromotive force to overcome the resistance of the circuit lecompose the water; for if it be stronger than this, the oxygen comith the iron to produce a pulverulent oxide, which has no adherence. onditions are as they should be, it is only a few minutes after the appears at the metal before the darkening of the surface shows e gas has united with the iron to form the magnetic oxide Fe_3O_4, is well known will resist the action of the air and protect the metal it. After the action has continued an hour or two the coating is itly solid to resist the scratch-brush, and it will then take a brilliant

iece of thickly rusted iron be placed in the bath, its sesquioxide is rapidly transformed into the magnetic oxide. This outer layer adhesion, but beneath it there will be found a coating which is a part of the metal itself.

early experiments M. de Meritens employed pieces of steel only. rought and cast iron he was not successful, for the coating came off slightest friction. He then placed the iron at the negative pol of aratus, after it had been already applied to the positive pole. Here e was reduced, and hydrogen was accumulated in the pores of the The specimens were then returned to the anode, when it was found oxide appeared quite readily and was very solid. But the result quite perfect, and it was not until the bath was filled with distilled place of that from the public supply, that a perfectly satisfactory as attained.

ganese Plating of Iron as a Protection from Rust. ding to the Italian *Progresso*, articles of iron can be protected rust by sinking them near the negative pole of an electric bath com-10 litres of water, 50 grammes of chloride of manganese, and 200 s of nitrate of ammonium. Under the influence of the current the osits on the articles a film of metallic manganese which prevents m rusting.

n-oxidizing Process of Annealing is described by H. in *Eng'g News*, Jan. 2, 1892. The ordinary process of annealing, s of which hard and brittle iron or steel is rendered soft and tough, in heating the metal to a good red-heat and then allowing it to cool y. While the metal is in a heated condition the surface becomes and although for many classes of work the scale of oxide is of no importance, yet in some cases it is very undesirable and even tes considerable expense in its removal.

w process uses a non-oxidizing gas, and is the invention of Mr. Jones, of Hartford, Conn. The principal feature of this process in keeping the annealing-retort in communication with the gas gas main during the entire process of heating and cooling, the gas ng allowed to expand back into the main, and being, therefore, practically constant pressure.

torts used are made from wrought-iron tubes. The gas used is ectly from the mains supplying the city with illuminating gas. It ed that if metal which had been blued or slightly oxidized was subthe annealing process it came out bright, the oxide being reduced tion of the gas. Practical use has been made of this fact in deoxital.

ative tests were made of specimens of metal annealed in illumis and of specimens annealed in nitrogen. The results of these tests pared with the results of tests of specimens annealed in an open ooled in ashes, and of specimens of the unannealed metal, and elative efficiency of the gas process was determined.

cimens were made from steel wire .188 in. in diameter and were wn to diameters of .156 and .150 in. Different lots of wire were order to secure average results. The elongations were in each red to an original length of 1.15 ins.

erence in total per cent of elongation and in breaking load between ens annealed in nitrogen and those annealed in illuminating gas ght. The average results were as follows:

Lot.	Gas used.	No. Test Pieces.	Breaking Load, lbs. per sq. in.	Elongation.	
				Total p. c.	p. c. gain
A	Nitrogen	4	62,140	29 12	22.0
B.......	Illuminating	4	63,140	28.08	20.8
C.......	Nitrogen	4	60.000	28.00	19.2
D.......	Illuminating	4	60,400	27.20	18.4
E... ...	Nitrogen	5	57,330	30.88	23.7
F.......	Illuminating	5	57.070	29.60	22.4
G.......	Open fire	8	63,090	26.76	19.6
H......	Unannealed	5	97,120	7.12	.....
I.......	Unannealed	5	80,790	8.80	.....

Painting Wood and Iron Structures. (E. H. Brown, En Club of Phila., *Engineering News*, April 20, 1893.)—A paint consists of portions—the pigment and the vehicle or binder. The pigment is a substance which is more or less finely ground, so as to be capable mixed with the vehicle) of being spread out in a thin layer or coating the surface to be painted. The vehicle or binder is the liquid in which pigment is mixed or ground, which serves to spread the pigment over surface to be painted, and which also holds it to that surface. For ordin painting the most generally used vehicle is linseed oil.

Linseed oil possesses the peculiar property of drying by uniting with oxygen of the air to form a tough, leather-like compound called linoxi

For painting on wood, zinc white has valuable pigment properties, these seem to be most fully developed when this pigment is used in junction with white lead, and then to the best advantage when the mix is used as a final coat over an elastic undercoating of white lead. So far other white base has been discovered which possesses at the same time other properties which render white lead valuable, namely, covering spreading capacity.

Of the inert pigments, lampblack is probably the most valuable. almost pure carbon, it is practically unchangeable except by fire. It has peculiar property of absorbing great quantities of linseed oil, and hence spreading over a large surface. French ochre, an earth pigment contai more or less of the hydrated oxide of iron, possesses the property of abso ing a large quantity of oil, and hence has considerable spreading capac and also holds very firmly to any wooden surface to which it may be app

The various mineral and metallic paints are almost all natural or artifi iron oxides. While these are cheap and useful for painting rough wo structures they are sometimes really quite dangerous for application to work, because, instead of preventing oxidation, they are apt to further

Coal tar is much used as a paint for the roughest class of work, both and iron; in the latter case especially for cast-iron pipes, smoke-stacks, work to be buried underground. It has the nature both of a resin and It has the disadvantage of becoming exceedingly brittle by the action of and softening at 115° F. Asphalt permits of somewhat wider range of tem ature, but otherwise exhibits the same peculiarities. These substances, they last, are probably the most valuable of paints, especially under but they are unfortunate in their tendency to flow or crawl on the sur to which they are applied, finally leaving the upper portions almost or bare. This is the case even under ground.

Red lead has long been regarded as the best possible preservative for dry iron. But in order to be most effective, the iron must be perfectly and free from any suspicions of rust, and absolutely dry. Red lead sh be perfectly pure and of the best and most careful preparation. That any well-known corroding house may be depended upon for purity, always for quality. It is simply a red oxide of lead. The best type is mineral, which is made by roasting white lead. On account of its this is not so frequently used as it would deserve. Red lead proper is directly from the metal, which is first oxidized to the yellow litharge then to the red oxide. This, however, does not give as good a paint as ...p, settlings, and tailings of the white lead works. ...uickly with linseed oil, it must be used within ...und, and, moreover, it is rather difficult to is great temptation to add some substance, such as whiting, to make it work freer, as well as to cost less money for material. inting iron work it is essential that the iron itself should be ab- from rust. Rust has the peculiar property of spreading and om a centre, if there be the slightest chance to do so. Hence, a nt of rust on the iron may grow under the surface of the paint, it be true, as Dr. Dudley asserts, that linseed oil is permeable noisture, and in time the paint will be flaked off by the rust un- ius gradually exposing the bare surface of the iron to the action ying agent, oxygen in the presence of water. It is necessary ll the scale possible from wrought iron by means of stiff wire d then to remove the rust by a pickle of very dilute acid, which ard be thoroughly washed off before the paint is applied. The e iron should be dry and at least moderately warmed before it The best method of painting a tin roof is to carefully remove oil or grease from the surface of the tin while it is yet bright ; then to apply a coat of red lead and linseed oil, or the best etallic paint, and to follow this with one or two coats of graphite graphite is almost unchangeable by atmospheric action, and is waterproof as well.

ad as a Preservative of Iron.—A. J. Whitney writes to *News*, August, 1891, that in 30 years' experience he has found e the best material for preserving iron under all circumstances.

ty of Paint Required for a Given Surface. (M. P. ft. of surface ÷ 200 = gallons of liquid paint for two coats; sq. e ÷ 18 = lbs. of pure white lead for three coats.

es of Paints.—*The Railroad and Engineering Journal*, vols. 90 and 1891, has a series of articles on paint as applied to *wooden* ts chemical nature, application, adulteration, etc., by Dr. C. B. mist, and F. N. Pease, assistant chemist, of the Penna. R. R. e results of a long series of experiments on paint as applied to poses.

e Paint. (M. P. Wood.)—Graphite, mixed with pure boiled which a small percentage of litharge, red lead, manganese, or lic salt has been added at the time of boiling to aid in the oxida- il, forms a most effective paint for metallic surfaces, as well as d fibrous substances. Wood surfaces protected by this paint, to the action of sea-water for a number of years, are found in te of preservation.

STEEL.

ON BETWEEN THE CHEMICAL COMPOSI- AND PHYSICAL CHARACTER OF STEEL.

ster (see Trans. A. I. M. E., vols. xxi and xxii, 1893–4) gives re- ral hundred analyses and tensile tests of basic Bessemer steel rom a study of them draws conclusions as to the relation of nposition to strength, the chief of which are condensed as

tions are that a pure iron, without carbon, phosphorus, man- on, or sulphur, if it could be obtained, would have a tensile 34,750 lbs. per square inch, if tested in a ⅜-inch plate. With se, a table is constructed by adding the following hardening own by increase of tensile strength, for the several elements

onstant effect of 800 lbs. for each 0.01%.
" " 500 " " 0.01%.

s, the effect is higher in high-carbon than in low-carbon steels.

n hundredths %... ...	9	10	11	12	13	14	15	16	17
P has an effect of lbs.	900	1000	1100	1200	1300	1400	1500	1500	1500

, the effect decreases as the per cent of manganese increases.

er cent......	.00 to .15	.15 to .20	.20 to .25	.25 to .30	.30 to .35	.35 to .40	.40 to .45	.45 to .50	.50 to .55	.55 to .65
reases for .01%	240	*240*	*220*	*200*	*180*	*160*	140	120	100	100 [illegible]
from *0 Mn...*	*3600*	*4800*	*5900*	*6900*	*7800*	*8600*	9300	9900	10,400	11,[illegible]

Silicon is so low in this steel that its hardening effect has no[t been con]sidered.

With the above additions for carbon and phosphorus the foll[owing table] has been constructed (abridged from the original by Mr. Webst[er). To the] figures given the additions for sulphur and manganese should b[e made as] above.

Estimated Ultimate Strengths of Basic Bessem[er] Plates.

For Carbon, .06 to .24; Phosphorus, .00 to .10; Manganese and Sul[phur ...] all cases.

Carbon.	.06	.08	.10	.12	.14	.16	.18	.20
Phos. .005	39,950	41,550	43,250	44,950	46,650	48,300	49,900	51,500
" .01	40,350	41,950	43,750	45,550	47,350	49,050	50,650	52,250
" .02	41,150	42,750	44,750	46,750	48,750	50,550	52,150	53,750
" .03	41,950	43,550	45,750	47,950	50,150	52,050	53,650	55,250
" .04	42,750	44,350	46,750	49,150	51,550	53,550	55,150	56,750
" .05	43,550	45,150	47,750	50,350	52,950	55,050	56,650	58,250
" .06	44,350	45,950	48,750	51,550	54,350	56,550	58,150	59,750
" .07	45,150	46,750	49,750	52,750	55,750	58,050	59,650	61,250
" .08	45,950	47,550	50,750	53,950	57,150	59,550	61,150	62,750
" .09	46,750	48,350	51,750	55,150	58,550	61,050	62,650	64,250
" .10	47,550	49,150	52,750	56,350	59,950	62,550	64,150	65,750
.001 Phos =	80 lbs.	80 lbs.	100 lb	120 lb	140 lb	150 lb	150 lb	150 lb

In all rolled steel the quality depends on the size of the bloo[m] from which it is rolled, the work put on it, and the temperature [at which it] is finished, as well as the chemical composition.

The above table is based on tests of plates ⅜ inch thick an[d ...] inches wide; for other plates Mr. Webster gives the following c[orrections] for thickness and width. They are made necessary only by th[e effect of] thickness and width on the finishing temperature in ordinary [practice.] Steel is frequently spoiled by being finished at too high a tempera[ture.]

Corrections for Size of Plates.

Plates. Inches thick.	Up to 70 ins. wide. Lbs.	Over 70 [...] L[bs.]
¾ and over	— 2000	
11/16 "	— 1750	
⅝ "	— 1500	
9/16 "	— 1250	
½ "	— 1000	
7/16 "	— 500	
⅜ "	0	
5/16 "	+ 3000	

Comparing the actual result of tests of 408 plates with the [calculated] results, Mr. Webster found the variation to range as in the table b[elow].

Summary of the Differences Between Calculate[d and] Actual Results in 408 Tests of Plate Steel.

In the first three columns the effects of sulphur were not consi[dered; in] the last three columns the effect of sulphur was estimated at [...] each .01% of S.

	Universal Mill.	Sheared.	Both Mills.	Universal Mill.	Sheared.
Per cent within 1000 lbs..	23.4	32.1	28.4	24.6	27.0
" " " 2000 " ..	40.9	48.9	45.6	48.5	54.9
" " " 3000 " ..	62.5	71.3	67.6	67.8	73.0
[illegible] " ..	75.5	81.0	78.7	82.5	85.2
[illegible] " ..	89.5	91.1	90.4	93.0	92.8

t figure in the table would indicate that if specifications were drawn r steel plates not to vary more than 5000 lbs. T. S. from a specified ual to a total range of 10,000 lbs.), there would be a probability of tion of 5% of the blooms rolled, even if the whole lot was made el of identical chemical analysis. In 1000 heats only 2% of the heats meet the requirements of the orders on which they were graded; of plates was much less than 1%, as one plate was rolled from each tested before rolling the remainder of the heat.

Hadfield (*Jour. Iron & Steel Inst., No.* 1, 1894) gives the strength of e Swedish iron, remelted and tested as cast, 20.1 tons (45,024 lbs.) n.; remelted and forged, 21 tons (47,040 lbs.). The analysis of the was: C, 0.08; Si, 0.04; S, 0.02; P, 0.02; Mn, 0.01; Fe, 99.82.

t of Oxygen upon Strength of Steel.—A. Lantz, of the orks, Germany, in a letter to Mr. Webster, says: "We have found he current year (1893) that oxygen plays an important rôle, till now served—such, indeed, that given a like content of carbon, phospho- manganese in the blows, a blow with greater oxygen content gives r hardness and less ductility than a blow with less oxygen content." thod used for determining oxygen is that of Prof. Ledebur, given in nd *Eisen*, May, 1892, p. 193. The variation in oxygen content may difference in strength of nearly one-half ton per square inch. *Iron & Steel Inst., No.* 1, 1894.)

GE OF VARIATION IN STRENGTH OF BESSEMER AND OPEN-HEARTH STEELS.

arnegie Steel Co. in 1888 published a list of 1057 tests of Bessemer en-hearth steel, from which the following figures are selected:

d of Steel.	No. of Tests.	Elastic Limit.		Ultimate Strength.		Elongation per cent in 8 inches.	
		High't.	Lowest	High't.	Lowest	High't.	Lowest
s. structural...	100	46,570	39,230	71,300	61,450	33.00	23.75
" ...	170	47,690	39,970	73,540	65,200	30.25	23.15
angles	72	41,890	32,630	63,450	56,130	34.30	26.25
I. fire-box.....	25	....	....	62,790	50,350	36.00	25.62
k.........	19	....	....	66,062	59,440	27.50	19.25
I. bridge	20	....	....	69,940	63,970	30.00	22.75

Requirements of Specifications.

tic limit, 35,000; tensile strength, 62,000 to 70,000; elong. 22% in 8 in.
tic limit, 40,000; tensile strength, 67,000 to 75,000.
tic limit, 30,000; tensile strength, 56,000 to 64,000; elong. 20% in 8 in.
sile strength, 50,000 to 62,000; elong. 26% in 4 in.
sile strength, 60,000 to 65,000; elong. 18% in 8 in.
sile strength, 64,000 to 70,000; elong. 20% in 8 in.

ngth of Open-hearth Structural Steel. (Pencoyd Iron —As a general rule, the percentage of carbon in steel determines its s and strength. The higher the carbon the harder the steel, the the tenacity, and the lower the ductility will be. The following list the average physical properties of good open-hearth steel:

tage bon.	Ultimate Tenacity, lbs. per sq. in.	Elastic Limit, lbs. per sq. in.	Stretch in 8 inches.	Reduction of Area, %.
	57,000	34,000	28 per cent.	55 per cent.
	62,000	37,000	26 "	50 "
	67,000	40,000	24 "	45 "
	72,000	43,000	22 "	40 "
	77,000	46,000	20 "	35 "
	82,000	49,000	18 "	30 "
	87,000	52,000	16 "	25 "

efficient of elasticity is *practically uniform* for all grades, a as for iron, *viz.*, *29,000,000 lbs.* These figures form the ave *rous series of tests* from rolled bars, and can only serve as proximation in single instances, when the variation from the be considerable. Steel below .10 carbon should be capable of without fracture, after being chilled from a red heat in cold of .15 carbon will occasionally submit to the same treatm usually bend around a curve whose radius is equal to the the specimen ; about 90% of specimens stand the latter bending fracture. As the steel becomes harder its ability to endure test becomes more exceptional, and when the carbon ratio little over 25% of specimens will stand the last-described bendin having about .40% carbon will usually harden sufficiently to and maintain an edge.

Mehrtens gives the following tables in *Stahl und Eisen* (*Iron* 1893) :

Basic Bessemer Steel.

680 Charges.

Elastic Limit, pounds per sq. in.	Charges within Range, per cent of total number.
35,500 to 38,400	15.0
38,400 to 39,800	31.6
39,800 to 41,200	27.5
41,200 to 42,700	16.0
42,700 to 46,400	9.9

Tensile Strength, pounds per sq. in.	Charges within Range, per cent of total number.
55,600 to 56,900	18.67
56,900 to 58,300	38.67
58,300 to 59,700	23.53
59,700 to 61,200	15.60
61,200 to 62,300	3.53

STRUCTURAL STEEL.

Elongation. per cent.	Charges within Range, per cent of total number.
21 to 25	2.65
25 to 26	8.53
26 to 27	17.35
27 to 28	26.76
28 to 29	23.68
29 to 30	14.41
30 to 32.5	6.62

RIVET STEEL.

25.2 to 26	20.0
26 to 27	15.0
27 to 28	25.0
28 to 29	25.0
29 to 29.8	15.0

Basic Open-hear tural Stee 489 Charg

Elastic Limit, pounds per sq. in.	Ch Ra of
34,400 to 37,000	
37,000 to 38,400	
38,400 to 39,800	
39,800 to 41,200	
41,200 to 42,700	
42,700 to 44,100	
44,100 to 48,400	
Tensile Strength.	
55,800 to 56,900	
56,900 to 58,300	
58,300 to 59,700	
59,700 to 61,200	
61,200 to 62,600	
62,600 to 65,100	
Elongation, per cent.	
20 to 25	
25 to 26	
26 to 27	
27 to 28	
28 to 29	
29 to 30	
30 to 37.1	

RIVET STEEL, 19

Tensile Strength.	
51,800	
51,900 to 53,300	
53,300 to 54,900	
54,900 to 56,300	
56,300 to 56,900	

Elongation all abov

In the basic Bessemer steel over 90% was below 0.08 phos were below 0.10; manganese was below 0.6 in over 90%, and be sulphur was below 0.05 in 84%, the maximum being 0.071; car 0.10, and silicon below 0.01 in all. In the basic open-hearth ste was below 0.06 in 96%, the maximum being 0.08; manganese be sulphur below 0.07 in 88%, the maximum being 0.12. The c from 0.09 to 0.14.

Low Tensile Strength of Very Pure Steel.—Sw open-hearth steel, tested by the author in 1881, showed a tens only 42,591 lbs. per sq. in. A piece of American nail-rod steel lbs. per sq. in. Both steels contained about .10 carbon and .01 and were very low in sulphur, manganese, and silicon. The were bars about 2 × 3/8 in. section.

Low Strength Due to Insufficient Work. Trans. A. I. M. E., 1886.)—Soft steel ingots, made in the ord boiler- nly from 10,000 to 20,000 lbs. tensile stren about 10% in 8 in., and a reduction of ar perly heated and rolled down from ive from 55,000 to 65,000 lbs. tensile strength, an elongation 23% to 33%, and a reduction of area of from 55% to 70%. Any hort of the above reduction in thickness ordinarily yields in- lts in its tensile tests.

of Soft Steel.—A. E. Hunt (Trans. A. I. M. E., 1883, vol. ft steel, no matter how low in carbon, will harden to a cer- being heated red-hot and plunged into water, and that it hen plunged into brine and less when quenched in oil.

was a heat of open-hearth steel of 0.15% carbon and 0.29% of ch gave the following results upon test-pieces from the same .

	Maximum Load. lbs. per sq. in.	Elongation in 8 in. Per cent.	Reduction of Area. Per cent.
lened....	55,000	27	62
ed in water.....	74,000	25	50
ed in brine......	84,000	22	43
ed in oil........	67,700	26	49

ility of such hardened steel does not decrease to the extent d tenacity would indicate, and is much superior to that of the high tenacity, still the greatly increased tenacity after tes that there must be a considerable molecular change in ardened, and that if such a hardening should be created plate, there must be very dangerous internal strains caused

ld Rolling.—Cold rolling of iron and steel increases the the ultimate strength, and decreases the ductility. Major ents on bars rolled and polished cold by Lauth's process ge increase of load required to give a slight permanent set insverse, 162%; torsion, 130%; compression, 161% on short ong, and 64% on columns 8 in. long; tension, 95%. The hard- ed by the weight required to produce equal indentations, %; and it was found that the hardness was as great in the s as elsewhere. Sir W. Fairbairn's experiments showed an ate tensile strength of 50%, and a reduction in the elongation 2 in. or 20%, to 0.79 in. or 7.9%.

of Tests of Full-size Eye-bars and Sample of Same Steel Used in the Memphis Bridge.
(Geo. S. Morison, Trans. A. S. C. E., 1893.)

Sized Eyebars, ide × 1 to 2 3/16" thick.			Sample Bars from Same Melts, about 1 in. area.			
gation.	Elastic Limit,	Max. Load,	Reduction,	Elongation,	Elastic Limit,	Max. Load,
p.c.	lbs. per	sq. in.	p. c.	p. c.	lbs. per	sq. in.
16.8	35,100	67,490	47.5	27.5	41,580	73,050
8.2	37,680	70,160	52.6	24.4	42,650	75,620
11.8	39,700	65,500	47.9	28.8	40,280	70,280
17.3	33,140	65,060	47.5	27.5	41,580	73,050
13.5	32,860	65,600	44.5	20.0	43,750	75,000
15.3	31,110	61,060	42.7	28.8	42,210	69,730
13.7	33,990	63,220	52.2	28.1	40,230	69,720
13.5	29,330	63,100	48.3	28.8	38,090	71,300
6.9	28,080	55,160	43.2	24.2	38,320	70,220
14.1	29,670	62,140	59.6	26.3	40,200	71,080
11.8	32,700	65,400	40.3	25.0	39,360	69,360
19.3	30,500	58,870	40.3	25.0	40,910	70,360
12.3	33,360	73,550	51.5	25.5	40,410	69,900
15.7	32,520	60,710	43.6	27.0	40,400	70,490
14.9	28,000	58,720	44.4	29.5	40,000	66,800
13.1	32,290	62,270	42.8	21.3	40,530	[illegible]
15.1	29,970	58,680	45.7	27.0	40,610	[illegible]

rength of the full-sized eye-bars was about 8000 lbs ess than that of the sample test-pieces.

TREATMENT OF STRUCTURAL STEEL.

(James Christie, Trans. A. S. C. E., 1893.)

Effect of Punching and Shearing.—There is no doubt that ste of higher tensile strength than is now accepted for structural purpo should not be punched or sheared, or that the softer material may conta elements prejudicial to its use however treated, but especially if punch But extensive evidence is on record indicating that steel of good quality, bars of moderate thickness and below or not much exceeding 80,000 lb tensile strength, is not any more, and frequently not as much, injured wrought iron by the process of punching or shearing.

The physical effects of punching and shearing as denoted by tensile te are for iron or steel:

Reduction of ductility; elevation of tensile strength at elastic limit; redu tion of ultimate tensile strength.

In very thin material the superficial disturbance described is less than thick; in fact, a degree of thinness is reached where this disturbance pra tically ceases. On the contrary, as thickness is increased the inju becomes more evident.

The effects described do not invariably ensue; for unknown reasons the are sometimes marked deviations from what seems to be a general result

By thoroughly annealing sheared or punched steels the ductility is to large extent restored and the exaggerated elastic limit reduced, the chan being modified by the temperature of reheating and the method of coolin

It is probable that the best results combined with least expenditure c be obtained by punching all holes where vital strains are not transferred the rivets; and by reaming for important joints where strains on rivet joints are vital, or wherever perforation may reduce sections to a minimu The reaming should be sufficient to thoroughly remove the material di turbed by punching; to accomplish this it is best to enlarge punched hol at least 1/8 in. diameter with the reamer.

Riveting.—It is the current practice to perforate holes 1/16 in. larg than the rivet diameter. For work to be reamed it is also a usual requi ment to punch the holes from 1/8 to 3/16 in. less than the finished diamet the holes being reamed to the proper size after the various parts a assembled.

It is also excellent practice to remove the sharp corner at both ends the reamed holes, so that a fillet will be formed at the junction of the bo and head of the finished rivets.

The rivets of either iron or mild steel should be heated to a bright red yellow heat and subjected to a pressure of not less than 50 tons per squ inch of sectional area.

For rivets of ordinary length this pressure has been found sufficient completely fill the hole. If, however, the holes and the rivets are exc tionally long, a greater pressure and a slower movement of the closing t than is used for shorter rivets has been found advantageous in compelli the more sluggish flow of the metal throughout the longer hole.

Welding.—No welding should be allowed on any steel that enters i structures.

Upsetting.—Enlarged ends on tension bars for screw-threads, eyeb etc., are formed by upsetting the material. With proper treatment an sufficient increment of enlarged sectional area over the body of the bar t result is entirely satisfactory. The upsetting process should be perform so that the properly heated metal is compelled to flow without folding lapping.

Annealing.—The object of annealing structural steel is for the purp of securing homogeneity of structure that is supposed to be impaired by equal heating, or by the manipulation necessarily attendant on certain p cesses. The objects to be annealed should be heated throughout t uniform temperature and uniformly cooled.

The physical effects of annealing, as indicated by tensile tests, depend the grade of steel, or the amount of hardening elements associated wit also on the temperature to which the steel is raised, and the method or of cooling the heated material.

The physical effects of annealing medium-grade steel, as indicated by t sile test, are reported very differently by different observers, some clai opposite results from others. It is evident, when all the attend 's are considered, that the obtained results must vary both in e.

ens, Trans. A. S. C. E. 1893, says: "A good mild steel can be
adily as wrought iron in the shop or the field, and even bear
eatment. It was, however, often thought necessary to require
annealing to remove the initial strains due to rolling. The an-
doubtedly of great advantage to all steel above 64,000 lbs.
square inch, but it is questionable whether it is necessary in
The distortions due to heating cause trouble in subsequent
, especially of thin plates. It cannot be denied, however, that
duces greater toughness.
al way all unannealed mild steel for a strength of 56,000 to
y be worked in the same way as wrought iron. Rough treat-
ing at a blue heat must, however, be prohibited. Such treat
be borne by wrought iron, although it does not suffer so much
Shearing is to be avoided, except to prepare rough plates,
afterwards be smoothed by machine tools or files before using.
lso to be avoided, because the edges of holes are thereby
nd the yield point. Reaming drilled holes is not necessary,
when sharp drills are used and neat work is done. A slight
g of the edges of drilled holes is all that is necessary. Work-
ial while heated should be avoided as far as possible, and the
ld bear this in mind when designing structures. Upsetting,
l bending ought to be avoided, but when necessary the material
ealed after completion.
ng of a mild-steel rivet should be finished as quickly as possible,
to the dangerous heat. For this reason machine work is the
s a special advantage in machine work from the fact that the
be retained upon the rivet until it has cooled sufficiently to
ation and the consequent loosening of the rivet."

g and Drilling of Steel Plates. (Proc. Inst. M. E.,
326.)—In Prof. Unwin's report the results of the greater num-
periments made on iron and steel plates lead to the general
at, while thin plates, even of steel, do not suffer very much
g, yet in those of ½ in. thickness and upwards the loss of te-
punching ranges from 10% to 23% in iron plates and from 11% to
of mild steel. Mr. Parker found the loss of tenacity in steel
s high as fully one third of the original strength of the plate.
es, on the contrary, there is no appreciable loss of strength.
ssible to remove the bad effects of punching by subsequent

[illegible]

Welding of Steel.—[illegible]

INFLUENCE OF ANNEALING UPON MAGNETIC CAPACITY.

[illegible]

1. [illegible]

2. [illegible]

[illegible]

Effect of Annealing upon the Magnetic Capacity of Different Wires; Tests by the Magnetic Balance.

Description.	Magnetic Capacity.	
	Bright as sent.	Annealed.
	deg. on scale.	deg. on scale.
Best Swedish charcoal iron, first variety	230	525
" " " " second "	256	370
" " " " third "	278	503
Swedish Siemens-Martin iron	165	430
Puddled iron, best best	212	340
Bessemer steel, soft	150	291
" " hard	115	172
Crucible fine cast steel	50	84

Crucible Fine Steel, Tempered.	Magnetic Capacity.
Bright-yellow heat, cooled completely in cold water	28
Yellow-red heat, cooled completely in cold water	32
[illegible] yellow, let down in cold water to straw color	35
" " " " " blue	43
" cooled completely in oil	51
" let down in water to white	58
[illegible], cooled completely in water	66
" " " oil	72
" " oil	84

SPECIFICATIONS FOR STEEL.

l Steel.—There has been a change during the ten years from the opinions of engineers, as to the requirements in specifica-tural steel, in the direction of a preference for metal of low h and great ductility. The following specifications of differ-given by A. E. Hunt and G. H. Clapp, Trans. A. I. M. E. 1890,

MBERS.	1879.	1881.	1882.	1885.	1887.	1888.
....	50,000	40@45,000	40,000	40,000	40,000	38,000
th.........	80,000	70@80,000	70,000	70,000	67@75,000	63@70,000
8 in........	12%	18%	18%	18%	20%	22%
a..........	20%	30%	45%	42%	42%	45%
....	O.H.	O.H. or B.	O.H.	Not spec.	O.H. or B.	O.H. or B.
N MEMBERS:						
....	Same	50@55,000	50,000	50,000	Same as tension	
th...... ...	as	80@90,000	80,000	80,000	members.	
8 in..	ten-	12%	15%	15%	"	
a	sion.	20%	35%	35%	"	

(*Iron Age*, Nov. 3, 1892) says: Regarding steel to be used under ditions as wrought iron, that is, to be punched without ream-ems to be a decided opinion (and a growing one) among engi-is not safe to use steel in this way, when the ultimate tensile bove 65,000 lbs. The reason for this is, not so much because marked change in the material of this grade, but because all lly Bessemer steel, has a tendency to segregations of carbon ns, producing places in the metal which are harder than they uld be. As long as the percentages of carbon and phosphorus the effect of these segregations is inconsiderable; but when ages are increased, the existence of these hard spots in the s more marked, and it is therefore less adapted to the treat-h wrought iron is subjected.

ide consensus of opinion that at an ultimate of 64,000 to 65,000 ntages of carbon and phosphorus (which are the two harden-reach a point where the steel has a tendency to become tender, when subjected to rough treatment.

steel, therefore, running in ultimate strength from 54,000 to in some cases to 64,000 lbs., is now generally considered a ial for this class of work.

sicker, engineer of tests of Carnegie, Phipps & Co., writes as ruing grades of structural steel (*Eng'g News*, June 2, 1892):

teel.—Steel shall be of three grades—soft, medium, high.

-Specimens from finished material for test, cut to size speci-all have an ultimate strength of from 54,000 to 62,000 lbs. per limit one half the ultimate strength; minimum elongation of ninimum reduction of area at fracture 50%. This grade of cold 180° flat on itself, without sign of fracture on the outside rtion.

eel.—Specimens from finished material for test, cut to size re, shall have an ultimate strength of 60,000 to 68,000 lbs. per limit one half the ultimate strength; minimum elongation 20% mum reduction of area at fracture, 40%. This grade of steel 80° to a diameter equal to the thickness of the piece tested, or flaw on the outside of the bent portion.

—Specimens from finished material for test, cut to size speci-all have an ultimate strength of 66 000 to 74,000 lbs. per sq. in ; ne half the ultimate strength; minimum elongation, 18% in 8 reduction of area at fracture, 35%. This grade of steel to bend diameter equal to three times the thickness of the test-piece, or flaw on the outside of the bent portion.

Engineers' Club of Phila., 1891, gives specifications for struc-follows: The phosphorus in acid open-hearth steel must be and in all Bessemer or basic steel must be less than 0.08%.

l will be tested in specimens of at least one half square inch om the *finished material*. Each melt of steel will be tested *on rolled, and also widely differing* gauges of the same

... of the fracture ... tensile tests ...

... square inches and in ...

... of both soft and medium ...

... found to be superior ...

... by means of a ...

... be punched ...

... diameter in the case of soft ...

... fracture. ...

Experience shows that the ...
absolute guarantee that the ...
... will be equally ...
and angles, has a ...
square inch, and a ...
... whereas ...
and in tensile strength ...
corresponding variation ...
... than wrought ...
... cold rolling, or ...
... that soft steel ...
a safe and satisf...
... the same ...
... generally that ...
dangerous substitute ...

... rigid requir...

ture as stated above for test-bars, and be capable of bending ithout sign of fracture on the convex surface of the bend.

hip, and Tank Plates. W. F. Mattes (*Iron Age*, July mends that the different qualities of steel plates be classified

	Tank.	Ship.	Shell.	Fire-box.
, longitudinal	Limit, 75,000	55,000 to 65,000	55,000 to 65,000	55,000 to 60,000
8-in. longitu- n, per cent....	....	20	22½	25
, longitudinal	....	Flat.	Flat.	Flat.
t, transverse	....	Over 1 in. diam.	Over ½ in. diam.	Flat.
mit..........	0.15	0.10	0.06	0.045
....	....	....	0.065	0.05
ction.........	Easy.	Careful.	Close.	Rigid.

ufacturing firm in Pittsburgh advertises six different grades lows:

x. Fire-box. Extra flange. Flange. Shell. Tank.

e average phosphorus content in these grades is, respectively: .03 .04 0.6 0.8 .10.

ecifications for steel plates are the following (1889):

tes Navy.—Shell: Tensile strength, 58,000 to 67,000 lbs. per sq. n, 22% in 8-in. transverse section, 25% in 8-in. longitudinal section. nsile strength, 50,000 to 58,000 lbs.; elongation. 26% in 8 inches. equirements: P. not over .035%; S. not over .040%.

g test: Specimen to stand being bent flat on itself.

test: Steel heated to cherry-red, plunged in water 82° F., and ound curve 1½ times thickness of the plate.

miralty.—Tensile strength, 58,240 to 67,200 lbs.; elongation in ne cold-bending and quenching tests as U. S. Navy.

Boiler-makers' Association.—Tensile strength, 55,000 to 65,000 on in 8 in., 20% for plates ⅜ in. thick and under; 22% for plates ; 25% for plates ¾ in. and over.

g test: For plates ½ in. thick and under, specimen must bend without fracture; for plates over ½ in. thick, specimen must nding 180° around a mandril, 1½ times the thickness of the

equirements: P. not over .040%; S. not over .030%.

Shipmasters' Association.—Tensile strength, 62,000 to 72,000 on, 16% on pieces 9 in. long.

rom plates, heated to a low red and cooled in water the tem- which is 82° F., to undergo without crack or fracture being a curve the diameter of which does not exceed three times of the piece tested.

hell-plates, Front Tube-plate, and Butt-strips. 1892.)—The metal desired is a homogeneous steel having a th of 60,000 lbs. per sq. in., and an elongation of 25% in a ally 8 in. long. These plates will not be accepted if the test-

strength of less than 55,000 lbs. per sq. in.; 2. An elongation iginally 8 in. long less than 20%; 3. A tensile strength over sq. in.: should, however, the elongation be 27% or over, plates jected for high strength.

ire-box Plates, including Back Tube-plate. 1892.)—The metal should show a tensile strength of 60,000 lbs. d an elongation of 28% in a test section originally 8 in. long.

al Composition.	Desired.	Will be Rejected.
.. ..	0.18 per cent.	over 0.25, below 0.15
us, not above.........	0.03 "	over 0.04
se, not above...... ..	0.40 "	over 0.55
ot above........... ..	0.02 "	over 0.04
not above......... ..	0.02 "	over 0.05
ot above............	0.03 "	over 0.05

These plates will not be accepted if the test-piece sho strength of less than 55,000 lbs. per sq. in.; 2 An elong originally 8 in. long, less than 22% (20% in plates ¼ inch thi strength over 65,000 lbs. per sq. in. (68,000 for plates ¼ in however, the elongation be 30% or over, plates will not be strength; 4. Any single seam or cavity more than ¼ in. lon three fractures obtained on test for homogeneity, as descri

Homogeneity test: A portion of the test-piece is nicked grooved on a machine, transversely about a sixteenth of three places about 1¼ in. apart. The first groove should side, 1¼ in. from the square end of the piece; the seco it on the opposite side; and the third, 1¼ in. from the opposite side from it. The test-piece is then put in a vis groove about ¼ in. above the jaw, care being taken to The projecting end of the test-piece is then broken off by mer, a number of light blows being used, and the bend from the groove. The piece is broken at the other two gro way. The object of this treatment is to open and render any seams due to failure to weld up, or to foreign interp cavities due to gas bubbles in the ingot. After rupture, fracture is examined, a pocket lens being used if necessary of the seams and cavities is determined. The length of the cavity determines the acceptance or rejection of the plate.

Dr. C. B. Dudley, chemist of the Penna. R. R. (Trans. A. xx. p. 709), gives as an example of the progressive improv cations the following: In the early days of steel boilers the force called for steel of not less than 50,000 lbs. tensile stren than 25% elongation. Some metal was received having 7 strength, and as the elongation was all right it was accepted plates were being flanged in the boiler-shop they crack pieces. As a result, an upper limit of 65,000 lbs. tensil established.

Am. Ry. Master Mechanics' Assn., 1894.—Same as Penna. tions of 1892, including homogeneity test.

Plate, Tank, and Sheet Steel. (Penna. R. R., 188 taken lengthwise of each plate, ⅛ in. thick and over, wit should have a tensile strength of 60,000 lbs. per sq. in., and 25% in a section originally 2 in. long.

Sheets will not be accepted if the tests show the tensile st 55,000 lbs. or greater than 70,000 lbs. per sq. in., nor if the below 20%.

Steel Billets for Main and Parallel Rods. (Pe —One billet from each lot of 25 billets or smaller shipment or parallel rods for locomotives will have a piece drawn fr hammer and a test-section will be turned down on this p diameter and 2 in. long. Such test-piece should show a te 85,000 lbs. and an elongation of 15%.

No lot will be acceptable if the test shows less than 8 strength or 12% elongation in 2 in.

Locomotive Spring Steel. (Penna. R. R., 1887.)—E more than 0.01 in. in thickness, or more than 0.02 in. in widt ordered, or which break where they are not nicked, or which nicked and held, fail to break square across where they ar returned. The metal desired has the following composition manganese, 0.25%; phosphorus, not over 0.03%; silicon, not phur, not over 0.03%; copper, not over 0.03%.

Shipments will not be accepted which show on analysis le over 1.10% of carbon, or over 0.50% of manganese, 0.05% of p of silicon, 0.05% of sulphur, and 0.05% of copper.

Steel for Locomotive Driving-axles. (Penn Steel for driving-axles should have a tensile strength of 85,0 and an elongation of 15% in section originally 2 in. long and taken midway between centre and circumference of the axl

Axles will not be accepted if tensile strength is less than elongation is below 12%.

Steel for Crank-pins. (Penna. R. R., 1886.)—Steel 1

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ecifications of the several dates given

uniformity between the two sides of the pin was so marked that it y determined not to put the lot of 50 pins in use. To guard against f this sort in future, the specifications are to be amended to require difference in ultimate strength of the two specimens shall not be 3000 lbs.

Car-axles. (Penna. R. R., 1891.)—For each 100 axles ordered 101 urnished, from which one will be taken at random, and subjected rescribed.

or passenger cars and passenger locomotive and tender trucks made of steel and be rough turned throughout. Two test-pieces from an axle, and the test sections of ⅝ in. diameter by 2 in. long t any part of the axle provided that the centre line of the test-l in. from the centre line of the axle. Such test-pieces should have trength of 80,000 lbs. per sq. in. and an elongation of 20%. Axles e accepted if the tensile strength is less than 75,000 lbs. or the below 15%, nor if the fractures are irregular.

r freight cars and freight-locomotive tender trucks must be made d will be subjected to the following test, which they must stand acture:

IN. DIAMETER AT CENTRE — Five blows at 20 ft. of a 1640-lb. weight, idway between supports 3 ft. apart; axle to be turned over after

IN. DIAMETER AT CENTRE—Five blows at 25 ft. of a 1640-lb. weight, dway between supports 3 ft. apart: axles to be turned over after

r Rails.—P. H. Dudley (Trans. A. S. C. E. 1893) recommends ng chemical composition for rails of the weights specified:

s per yard.....	60, 65, and 70 lbs.	75 and 80 lbs.	100 lbs.
....	.45 to .55%	.50 to .60%	.65 to .75%

weights: Manganese, .80% to 1.00%; silicon, .10% to .15%; phos-t over .06%; sulphur, not over .07%.

y itself up to or over 1% increases the hardness and tensile strength rapidly, and at the same time decreases the elongation. The carbon in the early rails ranged from 0.25 to 0.5 of 1% while in and very heavy sections it has been increased to 0.5, 0.6, and 0.75 h good irons and suitable sections it can run from 0.55 to 0.75 of ng to the section, and obtain fine-grain tough rails with low t.

se is a necessary ingredient in the first place to take up the oxide ned in the bath of molten metal during the blow. It also is of great

the diameter without showing cracks or flaws. The steel must
more than .035 of 1% of phosphorus, nor more than .04 of 1% of sul
A lot of 20 succesive tests of rivet steel of the low tensile stren
and 12 tests of the higher tensile strength gave the following res

	Low Steel.	Hig
Tensile strength, lbs. per sq. in...	51,230 to 54,100	59,100
Elastic limit, lbs. per sq. in.......	31,050 to 33,190	32,080
Elongation in 8 in., per cent......	30.5 to 35 25	28.5
Carbon, per cent	.11 to .14	.16
Phosphorus........................	.027 to .029	
Sulphur............................	.033 to .035	.033

The safest steel rivets are those of the lowest tensile strength,
are the least liable to become hardened and fracture by hamm
break from repeated concussive and vibratory strains to whi
subjected in practice. For calculations of the strength of rivete
tensile strength may be taken as the average of the figures abo
52,665 lbs., and the shearing strength at 45,000 lbs. per sq. in.

MISCELLANEOUS NOTES ON STEEL.

May Carbon be Burned Out of Steel?—Experime
the Laboratory of the Penna. Railroad Co. (Specifications for Sp
with the steel of spiral springs, show that the place from which
are taken for analysis has a very important influence on the am
bon found. If the sample is a piece of the round bar, and the
taken from the end of this piece, the carbon is always higher
borings are taken from the side of the piece. It is common to fi
ence of 0.10% between the centre and side of the bar, and in som
difference is as high as 0.23%. Furthermore, experiments made w
taken from the drawn out end of the bar show, usually, less c
samples taken from the round part of the bar, even though the b
be taken out of the side in both cases.

Apparently during the process of reducing the metal from the i
round bar, with successive heatings, the carbon in the outside o
burned out.

"Recalescence" of Steel.—If we heat a bar of copper
of constant strength, and note carefully the interval of time
passing from each degree to the next higher degree, we find th
tervals increase regularly, i.e., that the bar heats more and mor
its temperature approaches that of the flame. If we substitu
steel for one of copper, we find that these intervals increase regu
a certain point, when the rise of temperature is suddenly and in
greatly retarded or even completely arrested. After this the reg
temperature is resumed, though other like retardations may re
temperature rises farther. So if we cool a bar of steel slowly
temperature is greatly retarded when it reaches a certain point
ness. If the steel contains much carbon, and if certain favoring
be maintained, the temperature, after descending regularly, sud
spontaneously very abruptly, remains stationary a while, and
scends. This spontaneous reheating is known as "recalescence.

These retardations indicate that some change which absorbs
heat occurs within the metal. A retardation while the temperat
points to a change which absorbs heat; a retardation during coo
to some change which evolves heat. (Henry M. Howe, on "Heat
of Steel," Trans. A. I. M. E., vol. xxii.)

Effect of Nicking a Steel Bar.—The statement is somet
that, owing to the homogeneity of steel, a bar with a surface cra
in one of its edges is liable to fail by the gradual spreading of th
thus break under a very much smaller load than a sound bar.
is contended this does not occur, as this metal has a fibrous stru
Benjamin Baker has, however, shown that this theory, at least
statical stress is concerned, is opposed to the facts, as he purp
nicks in specimens of the mild steel used at the Forth Bridge
that the tensile strength of the whole was thus reduced by only
ton per square inch of section. In an experiment by the Union B
pany a full sized steel counter-bar, with a screw-turned buckle
was tested under a heavy statical stress, and at the same tim
[illegible] lbs. was allowed to drop on it from various height
[illegible] by ordinary statical strain, and showed a breaki

per square inch. The longer of the broken parts was then placed hine and put under the following loads, whilst a weight, as already , was dropped on it from various heights at a distance of five he sleeve-nut of the turn-buckle, as shown below:

ounds per sq. in.... .	50,000	55,000	60,000	63,000	65,000
	ft. in.	ft. in.	ft. in.	ft. in.	ft. in.
fall..................	2 1	2 6	3 0	4 0	5 0

ht was then shifted so as to fall dirctly on the sleeve-nut, and the ded as follows:

pecimen in lbs. per square inch	65,350	65,350	68,800
	ft.	ft.	ft.
fall..................................	3	6	6

seen that under this trial the bar carried more than when origi- d statically, showing that the nicking of the bar by screwing had iably weakened its power of resisting shocks.—*Eng'g News.*

ic Conductivity of Steel.—Louis Campredon reports in *Le l* the results of a series of experiments made to ascertain the rela- een electric resistance and chemical compositions of steel. The No. 17, 3 mm. diameter. The results are given in the table below:

Car-bon.	Silicon.	Sulphur.	Phos-phorus.	Manga-nese.	Total.	Electric Resist-ance, Ohms.
0.090	0.020	0.050	0.030	0.210	0.410	127.7
0.100	0.020	0.050	0.040	0.240	0.450	133.0
0.100	0.020	0.060	0.040	0.260	0.480	137.5
0.100	0.020	0.050	0.050	0.310	0.530	140.3
0.120	0.030	0.070	0.050	0.330	0.600	142.7
0.110	0.030	0.060	0.060	0.350	0.610	144.5
0.100	0.020	0.070	0.040	0.400	0.630	149.0
0.120	0.020	0.070	0.070	0.400	0.680	150.3
0.110	0.030	0.060	0.060	0.490	0.750	156.0
0.140	0.030	0.060	0.080	0.540	0.850	173.0

nination of these series of figures shows that the purer and softer etter is its electric conductivity, and, furthermore, that manga- element which most influences the conductivity.

ic Gravity of Soft Steel. (W. Kent, Trans. A. I. M. E., xiv. specimens of boiler-plate of C. 0.14, P. 0.03 gave an average sp. 2, maximum variation 0.008. The pieces were first planed to re- possible scale indentations, then filed smooth, then cleaned in huric acid, and then boiled in distilled water, to remove all traces n the surface.

ures of specific gravity thus obtained by careful experiment on ooth pieces of steel are, however, too high for use in determining ts of rolled plates for commercial purposes. The actual average of these plates is always a little less than is shown by the calipers, t of the oxide of iron on the surface, and because the surface is tly smooth and regular. A number of experiments on commercial d comparison of other authorities, led to the figure 7.854 as the pecific gravity of open-hearth boiler-plate steel. This figure is embered as being the same figure with change of position of the oint (.7854) which expresses the relation of the area of a circle to circumscribed square. Taking the weight of a cubic foot of water s 62.36 lbs. (average of several authorities), this figure gives 489.775 weight of a cubic foot of steel, or the even figure, 490 lbs., may be convenient figure, and accurate within the limits of the error of on.

ion method of approximating the weight of iron plates is to con- m to weigh 40 lbs. per square foot one inch thick. Taking this d adding 2% gives almost exactly the weight of steel boiler-plate ve ($40 \times 12 \times 1.02 = 489.6$ lbs. *per* cubic foot).

onal *Failures of* Bessemer Steel.—G. H. Cl... A. *n their paper on* "The Inspection of Materials of Con...

the United States " (Trans. A. I. M. E., vol. xix), say: Numerous could be cited to show the unreliability of Bessemer steel for struct poses. One of the most marked, however, was the following: A 12-i weighing 30 lbs. to the foot, 20 feet long, on being unloaded fr broke in two about 6 feet from one end.

The analyses and tensile tests made do not show any cause for th The cold and quench bending tests of both the original ¾-in. ro pieces, and of pieces cut from the finished material, gave satisfa sults; the cold-bending tests closing down on themselves withou fracture.

Numerous other cases of angles and plates that were so hard in to break off short in punching, or, what was worse, to break the have come under our observation, and although makers of Besse claim that this is just as likely to occur in open-hearth as in Besse we have as yet never seen an instance of failure of this kind in op steel having a composition such as C 0.25%, Mn 0.70%, P 0.80%.

J. W. Wailes, in a paper read before the Chemical Section of t Association for the Advancement of Science, in speaking of m failures of steel, states that investigation shows that " these failu in steel of one class, viz., soft steel made by the Bessemer process

Segregation in Steel Ingots. (A. Pourcel, Trans. A. I. M —H. M. Howe, in his " Metallurgy of Steel," gives a *résumé* of obs with the results of numerous analyses, bearing upon the phenome regation.

In 1881 Mr. Stubbs, of Manchester, showed the heterogeneous analyses made upon different parts of an ingot of large section.

A test-piece taken 24 inches from the head of the ingot 7.5 feet gave by analysis very different results from those of a test-piece inches from the bottom.

	C.	Mn.	Si.	S.
Top	0.92	0.535	0.043	0.161
Bottom	0.37	0.498	0.006	0.025

Windsor Richards says he had often observed in test-pieces ta different points of one plate variations of 0.05% of carbon. Segr specially pronounced in an ingot in its central portion, and at space of the piping.

It is most observable in large ingots, but in blocks of smaller w limited dimensions, subjected to the influence of solidification a casting within thick walls will permit, it may still be observed An ingot of Martin steel, weighing about 1000 lbs., and having a 1.10 feet and a section of 10.24 inches square, gave the following:

	C.	S.	P.
1. Upper section:	C.	S.	P.
Border	0.330	0.040	0.033
Centre	0.530	0.077	0.057
2. Lower section:	C.	S.	P.
Border	0.280	0.029	0.016
Centre	0.290	0.030	0.038
3. Middle section:	C.	S.	P.
Border	0.320	0.025	0.025
Centre	0.320	0.048	0.048

Segregation is less marked in ingots of extra-soft metal cast in moulds of considerable thickness. It is, however, still importan plains the difference often shown by the results of tests on pie from different portions of a plate. Two samples, taken from the s of a flat ingot, one on the outside and the other in the centre, 7.9 in the upper edge, gave:

	C.	S.	P.
Centre........................	0.14	0.053	0.072
Exterior......................	0.11	0.036	0.027

Manganese is the element most uniformly disseminated in ha steel.

For cannon of large calibre, if we reject, in addition to the pa sand and called the *masselotte* (sinking-head), one third of the u of the ingot, we can obtain a tube practically homogeneous in co because the central part is naturally removed by the boring of *With extra-soft* steels, destined for ship- or boiler-plates, the so *practically perfect* homogeneity lies in the obtaining of a metal ma serving its name of extra-soft metal.

ious consequences of segregation must be suppressed by reduc-
as possible, the elements subject to liquation.

Uses of Steel for Structural Purposes. (G. G. Trans. A. S. C. E. 1893).—The Pennsylvania Railroad Company iced Bessemer steel in America in locomotive boilers in the year steel was too hard and brittle for such use. The first plates el boilers had a tenacity of 85,000 to 92,000 lbs. and an elongation 10%. The results were not favorable, and the steel works were to offer a material of less tenacity and more ductility. The re- were therefore reduced to a tenacity of 78,000 lbs. or less, and on was increased to 15% or more. Even with this, between the d 1880, many explosions occurred and many careful examina- ade to determine their cause. It was found on examining the hat there were incipient changes in the metal, many cracks , and points near them were corroded with rust, all caused by tools in manufacturing. It was evident that the material le, and that the treatment must be changed. In the beginning arker, chief engineer of the Lloyds, stated that there was then ish steamer in possession of a steel boiler; a year later there 1878 there were but five large English steamers built of steel, there were 116 building. The use of Bessemer steel in bridge- tried first on the Dutch State railways in 1863-64, then in Eng- tria. In 1874 a bridge was built of Bessemer steel in Austria. of cast steel for bridges was in America, for the St. Louis Arch or the wire of the East River Bridge. These gave an impetus ingot metal, and before 1880 the Glasgow and Plattsmouth the Missouri River were also built of ingot metal. Steel eye- plied for the first time in the Glasgow Bridge. Since 1880 the of mild steel in all kinds of engineering structures has steadily

STEEL CASTINGS.

Engineering Congress, Dept. of Marine Eng'g, Chicago, 1893.)

rican steel-founders had successfully produced a considerable avy and difficult castings, of which the following are the most pecimens:

up to 24,000 lbs.; stern-posts up to 54,000 lbs.; stems up to draulic cylinders up to 11,000 lbs.; shaft-struts up to 32,000 lbs.; p to 7500 lbs.; stern-pipes up to 8000 lbs.

tage of success in these classes of castings since 1890 has ranged he more difficult forms to 90% in the simpler ones; the tensile been from 62,000 to 78,000 lbs., elongation from 15% to 25%. The nce recorded is that of a guide, cast in January, 1893, which 000 lbs. tensile strength and 15.6% elongation.

teel castings of which anything is generally known were s made for the Philadelphia & Reading R. R. in July, 1867, by Butcher Steel Works, now the Midvale Steel Co. The moulds f a mixture of ground fire-brick, black-lead crucible-pots and fire-clay, and washed with a black-lead wash. The steel n crucibles, and was about as hard as tool steel. The surface ings was very smooth, but the interior was very much honey- s was before the days when the use of silicon was known for eel. The sponginess, which was almost universal, was a great eir general adoption.

ep was to leave the ground pots out of the moulding mixture the mould with finely ground fire-brick. This was a great im- specially in very heavy castings; but this mixture still clung so e casting that only comparatively simple shapes could be made y. A mould made of such a mixture became almost as hard as d was such an obstacle to the proper shrinkage of castings, t all complicated in shape, they had so great a tendency to ake their successful manufacture almost impossible. By this of silicon had been discovered, and the only obstacle in the way od castings was a suitable moulding mixture. This was ulti- in mixtures having the various kinds of silica sand as the tituent.

most *fertile sources of defects* in castings is a bad *shapes can be cast successfully if* they are so desig cool uniformly. Mr. Cramp says while he is not yet prepared to sta anything that can be cast successfully in iron can be cast in steel, tions seem to point that way in all cases where it is possible to put able sinking-heads for feeding the casting.

H. L. Gantt (Trans. A. S. M. E., xii. 710) says: Steel castings n shrink much more than iron ones, but with less regularity. The am shrinkage varies with the composition and the heat of the metal; the the metal the greater the shrinkage; and, as we get smoother castin hot metal, it is better to make allowance for large shrinkage and p metal as hot as possible. Allow 3/16 or ¼ in. per ft. in for shrinkage, and ¼ in. for finish on machined surfaces, except suc cast "up." Cope surfaces which are to be machined should, in hard castings, have an allowance of from ⅜ to ½ in. for finish, as mass of metal slowly rising in a mould is apt to become crusty on t face, and such a crust is sure to be full of imperfections. On sm castings ⅛ in. on drag side and ¼ in. on cope side will be sufficient. should have less than ¼ in. finish on a side and very large ones shou as much as ½ in. on a side. Blow-holes can be entirely prevented ings by the addition of manganese and silicon in sufficient quantit both of these cause brittleness, and it is the object of the conscientio maker to put no more manganese and silicon in his steel than is ju cient to make it solid. The best results are arrived at when all port the castings are of a uniform thickness, or very nearly so.

The following table will illustrate the effect of annealing on strength and elongation of steel castings:

Carbon.	Unannealed.		Annealed.	
	Tensile Strength.	Elongation.	Tensile Strength.	Elong
.23%	68,738	22.40%	67,210	31.
.37	85,540	8.20	82,228	21.
.53	90,121	2.35	106,415	9.

The proper annealing of large castings takes nearly a week.

The proper steel for roll pinions, hammer dies, etc., seems to be th taining about .60% of carbon. Such castings, properly annealed, hav well and seldom broken. Miscellaneous gearing should contain carb to 60%, gears larger in diameter being softest. General machinery c should, as a rule, contain less than .40% of carbon, those exposed to shocks containing as low at .20% of carbon. Such castings will give a strength of from 60,000 to 80,000 lbs. per sq. in. and at least 15% exten a 2 in. long specimen. Machinery and hull castings for war-vessels United States Navy, as well as carriages for naval guns, contain from .30% of carbon.

The following is a partial list of castings in which steel seems rapidly taking the place of iron: Hydraulic cylinders, crossheads and for large engines, roughing rolls, rolling-mill spindles, coupling-box pinions, gearing, hammer-heads and dies, riveter stakes, castings for car-couplers, etc.

For description of methods of manufacture of steel castings by the mer, open-hearth, and crucible processes, see paper by P. G. Salom, A. I. M. E. xiv, 118.

Specifications for steel castings issued by the U. S. Navy Departme (abridged): Steel for castings must be made by either the open-hea the crucible process, and must not show more than .06% of phosphoru castings must be annealed, unless otherwise directed. The tensile st of steel castings shall be at least 60,000 lbs., with an elongation of a 15% in 8 in. for all castings for moving parts of the machinery, and a 10% in 8 in. for other castings. Bars 1 in. sq. shall be capable of be cold, without fracture, through an angle of 90°, over a radius not g than 1½ in. All castings must be sound, free from injurious roug ugginess, pitting, shrinkage, or other cracks, cavities, etc.

nnsylvania [illegible] pecifications, 1888: Steel castings should h *stren* [illegible] per sq. in. and an elongation of 15% in *lly* [illegible] astings will not be accepted if tensile

60,000 lbs., nor if the elongation is less than 12%, nor if cast- blow-holes and shrinkage cracks. Castings weighing 80 lbs. or have cast with them a strip to be used as a test-piece. The di- of this strip must be ¾ in. sq. by 12 in. long.

GANESE, NICKEL, AND OTHER "ALLOY" STEELS.

nese Steel. (H. M. Howe, Trans. A. S. M. E., vol. xii.)—Man- el is an alloy of iron and manganese, incidentally, and probably ly, containing a considerable proportion of carbon.

:t of small proportions of manganese on the hardness, strength, ity of iron is probably slight. The point at which manganese have a predominant effect is not known: it may be somewhere As the proportion of manganese rises above 2.5% the strength ty diminish, while the hardness increases. This effect reaches a with somewhere about 6% of manganese. When the proportion ment rises beyond 6% the strength and ductility both increase, ardness diminishes slightly, the maximum of both strength and eing reached with about 14% of manganese. With this proportion s still so hard that it is very difficult to cut it with steel tools. As tion of manganese rises above 15% the ductility falls off abruptly, h remaining nearly constant till the manganese passes 18%, when iminishes suddenly.

taining from 4% to 6.5% of manganese, even if it have but 0.37% of reported to be so extremely brittle that it can be powdered under nmer when cold; yet it is ductile when hot.

se steel is very free from blow-holes; it welds with great diffi- oughness is increased by quenching from a yellow heat; its elec- nce is enormous, and very constant with changing temperature; thermal conductivity. Its remarkable combination of great hard- a cannot be materially lessened by annealing, and great tensile with astonishing toughness and ductility, at once creates and usefulness. The fact that manganese steel cannot be softened, r remains so hard that it can be machined only with great diffi- up a barrier to its usefulness.

owing comparative results of abrasion tests of manganese and were reported by T. T. Morrell:

{ BY PRESSURE AGAINST A REVOLVING HARDENED-STEEL SHAFT.

ss of weight of manganese steel	1.0
" blue-tempered hard tool steel	0.4
" annealed hard tool steel	7.5
" hardened Otis boiler-plate steel	7.0
" annealed " " "	14.0

ABRASION BY AN EMERY-WHEEL.

ss of weight of hard manganese-steel wheels	1.00
" softer " "	1.19
" hardest carbon-steel wheels	1.23
" soft " "	2.85

lness of manganese steel seems to be of an anomalous kind. The rd, but under some conditions not rigid. It is very hard in its to abrasion; it is not always hard in its resistance to impact.

ese steel forges readily at a yellow heat, though at a bright white umbles under the hammer. But it offers greater resistance to on, i.e., it is harder when hot, than carbon steel.

t important single use for manganese-steel is for the pins which buckets of elevated dredgers. Here abrasion chiefly is to be

important use is for the links of common chain-elevators.

aterial for stamp-shoes, for horse-shoes, for the knuckles of an car-coupler, manganese steel has not met expectations.

ese steel has been regularly adopted for the blades of the Cyclone . Some manganese-steel wheels are reported to have run over les each without turning, on a New England railroad.

l Steel.—The remarkable tensile strength and ductility of nickel hown by the *test-bars and the behavior of nickel-st*[illegible] *er shot tests, are witness of the valuable qualities con*[illegible] *e addition of a few per cent of nickel.*

The following tests were made on nickel steels by Mr. Maunsel Whit
the Bethlehem Iron Company (*Eng. & M. Jour.*, Sept. 16, 1893.):

	Specimen from—	Diam., in.	Length, in.	Tensile Str'gth, lbs. per sq. in.	Elastic Limit, lbs. per sq. in.	p. c. ex.	p. c. cont.	
3¼% nickel steel.	Forged bars.*	.625	4	276,800	......	2.75	....	Spec
		"	"	246,595	....	4.25	6.0	treatm
		"	"	105,300	....	19.25	55.0	Annea
	1¼-in. round rolled bar.†	.564	4	142,800	74,000	13.0	28.2	
		"	"	143,200	74,000	12.32	27.6	
		"	"	117,600	64,000	17.0	46.0	
		"	"	119,200	65,000	16.66	42.1	
		"	"	91,600	51,000	22.25	53.2	
		"	"	91,200	51,000	21.62	53.4	
		"	"	85,200	53,000	21.82	49.5	
		"	"	86,000	48,000	21.25	47.4	
27% nickel steel	1¼ in. sq. bar, rolled.‡	.798	8	115,464	51,820	36.25	66.23	
		"	"	112,600	60,000	37.87	62.82	
		"	"	102,010	39,180	41.37	69.59	Annea
		"	"	102,510	40,200	44.00	68.34	"
	1-in. round bar, rolled.§	.500	2	114,590	56,020	47.25	68.4	
		"	"	115,610	59,080	45.25	62.3	
		"	"	105,240	45,170	49.65	72.8	Annea
		"	"	106,780	45,170	55.50	63.6	"

* Forged from 6-in. ingot to ⅝ in. diam., with conical heads for holdi
† Showing the effect of varying carbon.
‡ Rolled down from 14-in. ingot to 1¼-in. square billet, and turned to
§ Rolled down from 14-in. ingot to 1-in. round, and turned to size.

Nickel steel has shown itself to be possessed of some exceedingly val properties; these are, resistance to cracking, high elastic limit, and h geneity. Resistance to cracking, a property to which the name of non bility has been given, is shown more remarkably as the percentage of n increases. Bars of 27% nickel illustrate this property. A 1¼-in. squar was nicked ¼ in. deep and bent double on itself without further fra than the splintering off, as it were, of the nicked portion. Sudden failu rupture of this steel would be impossible; it seems to possess the tough of rawhide with the strength of steel. With this percentage of nicke steel is practically non corrodible and non-magnetic. The resistanc cracking shown by the lower percentages of nickel is best illustrated in many trials of nickel-steel armor.

The elastic limit rises in a very marked degree with the addition of a 3% of nickel, the other physical properties of the steel remaining uncha or perhaps slightly increased.

In such places (shafts, axles, etc.) where failure is the result of the fa of the metal this higher elastic limit of nickel steel will tend to prolon definitely the life of the piece, and at the same time, through its sup toughness, offer greater resistance to the sudden strains of shock.

Howe states that the hardness of nickel steel depends on the propo of nickel and carbon jointly, nickel up to a certain percentage incre the hardness, beyond this lessening it. Thus while steel with 2% of n and 0.90% of carbon cannot be machined, with less than 5% nickel it ca worked cold readily, provided the proportion of carbon be low. As proportion of nickel rises higher, cold-working becomes less easy. It fo easily whether it contain much or little nickel.

The presence of manganese in nickel steel is most important, as it app that without the aid of manganese in proper proportions, the conditio treatment would not be successful.

Tests of Nickel Steel.—Two heats of open-hearth steel were made the Cleveland Rolling Mill Co., one ordinary steel made with 9000 lbs. scrap and pig, and 165 lbs. ferro-manganese, the other the same wit addition of 3%, or 540 lbs. of nickel. Tests of six plates rolled from heat, 0.24 to 0.3 in. thick, gave results as follows:

Ordinary steel 52,500 to 56,500; E. L. 32,800 to 37,900; elong. 22
ckel steel ..370 to 67,100; " 47,100 to 48,800; " 25¼

n—in fact, higher than silicon. According to Berthonet, the co-
f aluminum to Al_2O_3 equals 7900 cal.; silicon to SiO_2 is stated as 7800.
tion of aluminum may be classed along with that of silicon, sulphur,
rus, arsenic, and copper, as giving no increase of hardness to iron,
distinction to carbon, manganese, chromium, tungsten, and nickel.
e, whilst for some special purposes aluminum may be employed in
ufacture of iron, at any rate with our present knowledge of its
es, this use cannot be large, especially when taking into considera-
fact of its comparatively high price. Its special advantage seems to
t combines in itself the advantages of both silicon and manganese;
ng as alloys containing these metals are so cheap and aluminum
extensive use seems hardly probable.

tead, in discussion of Mr. Hadfield's paper, said: Every one of our
s indicated that aluminum can kill the most fiery steel, providing,
, that it is added in sufficient quantity to combine with all the oxy-
h the steel contains. The metal will then be absolutely dead, and
r like dead-melted silicon steel. If the aluminum is added as metal-
num, and not as a compound, and if the addition is made just be-
steel is cast, 1/10% is ample to obtain perfect solidity in the steel.

me Steel. (F. L. Garrison, *Jour. F. I.*, Sept. 1891.)—Chromium
s the hardness of iron, perhaps also the tensile strength and elastic
t it lessens its weldibility.

chrome, according to Berthier, is made by strongly heating the
ides of iron and chromium in brasqued crucibles, adding powdered
if the oxide of chromium is in excess, and fluxes to scorify the
atter and prevent oxidation. Chromium does not appear to give
power of becoming harder when quenched or chilled. Howe states
ome steels forge more readily than tungsten steels, and when not
ng over 0.5 of chromium nearly as well as ordinary carbon steels of
entage of carbon. On the whole the status of chrome steel is not
ory. There are other steel alloys coming into use, which are so
tter, that it would seem to be only a question of time when it will
irely out of the race. Howe states that many experienced chemists
nd no chromium, or but the merest traces, in chrome steel sold in
kets.

Langley (Trans. A. S. C. E. 1892) says: Chromium, like manganese,
hardener of iron even in the absence of carbon. The addition of 1%
chromium to a carbon steel will make a metal which gets exces-
ard. Hitherto its principal employment has been in the production
l shot and shell. Powerful molecular stresses result during cooling,
shells frequently break spontaneously months after they are made.

then, when the percentage of tungsten is high, it has to be treated very carefully; and in order to avoid breaking it, not only is it necessary to reheat it several times while it is being hammered, but when the tool has acquired the desired shape hammering must still be continued gently and with numerous blows until it becomes nearly cold. Then only can it be cooled entirely.

Tungsten is not only employed to produce steel of an extraordinary hardness, but more especially to obtain a steel which, with a moderate hardness allies great toughness, resistance, and ductility. Steel from Assailly, used for this purpose, contained carbon, 0.52%; silicon, 0.04%; tungsten, 0.3[illegible] phosphorus, 0.04%; sulphur, 0.005%.

Mechanical tests made by Styffe gave the following results:

Breaking load per square inch of original area, pounds..	172,424
Reduction of area, per cent	0.54
Average elongation after fracture, per cent	13

According to analyses made by the Duc de Luynes of ten specimens of the celebrated Oriental damasked steel, eight contained tungsten, two of them in notable quantities (0.518% to 1%), while in all of the samples analyzed nickel was discovered ranging from traces to nearly 4%.

Stein & Schwartz of Philadelphia, in a circular say: It is stated that tungsten steel is suitable for the manufacture of steel magnets, since it retains its magnetism longer than ordinary steel. Mr. Kniesche has made tungsten up to 98% fine a specialty. Dr. Heppe, of Leipsig, has written a number of articles in German publications on the subject. The following instructions are given concerning the use of tungsten: In order to produce cast iron possessing great hardness an addition of one half to one and one half of tungsten is all that is needed. For bar iron it must be carried up to 1% to 2%, but should not exceed 2½%. For puddled steel the range is larger but an addition beyond 3½% only increases the hardness, so that it is brought up to 1½% only for special tools, coinage dies, drills, etc. For tires 2½% to 3 have proved best, and for axles ½ to 1½%. Cast steel to which tungsten has been added needs a higher temperature for tempering than ordinary steel and should be hardened only between yellow, red, and white. Chisels made of tungsten steel should be drawn between cherry-red and blue, and stand well on iron and steel. Tempering is best done in a mixture of 5 parts of yellow rosin, 3 parts of tar, and 2 parts of tallow, and then the article is once more heated and then tempered as usual in water of about 15° C.

Whitworth Compressed Steel. (Proc. Inst. M. E., May, 1887, p. 167.)—In this system a gradually increasing pressure up to 6 or 8 tons per square inch is applied to the fluid ingot, and within half an hour or less after the application of the pressure the column of fluid steel is shortened 1½ inch per foot or one eighth of its length; the pressure is then kept on for several hours, the result being that the metal is compressed into a perfectly solid and homogeneous material, free from blow-holes.

In large gun-ring ingots during cooling the carbon is driven to the centre, the centre containing 0.8 carbon and the outer ring 0.3. The centre is bored out until a test shows that the inside of the ring contains the same percentage of carbon as the outside.

Compressed steel is made by the Bethlehem Iron Co. and the Carnegie Steel Co. for armor-plate and for gun and other heavy forgings.

CRUCIBLE STEEL.

Selection of Grades by the Eye, and Effect of Heat Treatment. (J. W. Langley, *Amer. Chemist*, November, 1876.)—In 1874, Miller, Metcalf & Parkin, of Pittsburgh, selected eight samples of steel which were believed to form a set of graded specimens, the order being based on the quantity of carbon which they were supposed to contain. They were numbered from one to eight. On analysis, the quantity of carbon was found to follow the order of the numbers, while the other elements present—silicon, phosphorus, and sulphur—did not do so. The method of selection is described as follows:

The steel is melted in black-lead crucibles capable of holding about eighty pounds; when thoroughly fluid it is poured into cast-iron moulds, and when cold the top of the ingot is broken off, exposing a freshly-fractured surface. The appearance presented is that of confused groups of crystals, all appearing to have started from the outside and to have met in the centre; the general form is common to all ingots of whatever composition, but to the trained eye, and [illegible] one long and critically exercised, a minute but la able difference is perceived between varying samples of steel, and ference is now known to be owing almost wholly to variations in the t of combined carbon, as the following table will show. Twelve samlected by the eye alone, and analyses of drillings taken direct from ot before it had been heated or hammered, gave results as below:

Iron by Diff.	Carbon.	Diff. of Carbon.	Silicon.	Phos.	Sulph.
99.614	.302	....	.019	.047	.018
99.455	.490	.188	.034	.005	.016
99.363	.529	.039	.043	.047	.018
99.270	.649	.120	.039	.030	.012
99.119	.801	.152	.029	.035	.016
99.086	.841	.040	.039	.024	.010
99.044	.867	.026	.057	.014	.018
99.040	.871	.004	.053	.024	.012
98.900	.955	.084	.059	.070	.016
98 861	1.005	.050	.088	.034	.012
98.752	1.058	.053	.120	.064	.006
98.834	1.079	.021	.039	.044	.004

the carbon is seen to increase in quantity in the order of the numhile the other elements, with the exception of total iron, bear no relathe numbers on the samples. The mean difference of carbon is .071. ld steels the discrimination is less perfect.

appearance of the fracture by which the above twelve selections ade can only be seen in the cold ingot before any operation, except ginal one of casting, has been performed upon it. As soon as it is red, the structure changes in a remarkable manner, so that all trace primitive condition appears to be lost.

her method of rendering visible to the eye the molecular and chemiinges which go on in steel is by the process of hardening or temper-When the metal is heated and plunged into water it acquires an e of hardness, but a loss of ductility. If the heat to which the steel en raised just before plunging is too high, the metal acquires intense ss, but it is so brittle as to be worthless; the fracture is of a bright, ar, or sandy character. In this state it is said to be burned, and it again be restored to its former strength and ductility by annealing; ined for all practical purposes, but in just this state it again shows nces of structure corresponding with its content in carbon. The of these changes can be illustrated by plunging a bar highly heated end and cold at the other into water, and then breaking it off in of equal length, when the fractures will be found to show appearcharacteristic of the temperature to which the sample was raised.

specific gravity of steel is influenced not only by its chemical analyt by the heat to which it is subjected, as is shown by the following densities referred to 60° F.):

c gravities of twelve samples of steel from the ingot; also of six mmered bars, each bar being overheated at one end and cold at the her, in this state plunged into water, and then broken into pieces of ual length.

	1	2	3	4	5	6	7	8	9	10	11	12
....	7.855	7.836	7.841	7.829	7.838	7.824	7.819	7.818	7.813	7.807	7.803	7.805
ned 1...	.....	.....	7.818	7.791	.....	7.789	.....	7.752	.....	7.744	.....	7.690
2...	.....	.....	7.814	7.811	.....	7.784	.....	7.755	.....	7.749	.....	7.741
3...	.....	.....	7.823	7.830	.....	7.780	.....	7.758	.....	7.755	.....	7.769
4...	.....	.....	7.826	7.849	.....	7.808	.....	7.773	.....	7.789	.....	7.798
5...	.....	.....	7.831	7.806	.....	7.812	.....	7.790	.....	7.812	.....	7.811
old 6...	.....	.....	7.844	7.824	.....	7.829	.....	7.825	.....	[illegible]	.....	7.825

* Order of samples from bar.

Effect of Heat on the Grain of Steel. (W. Metcalf.—Jea Steel, p. 642.)—A simple experiment will show the alteration produ high-carbon steel by different methods of hardening. If a bar of st be nicked at about 9 or 10 places, and about half an inch apart, a specimen is obtained for the experiment. Place one end of the ba good fire, so that the first nicked piece is heated to whiteness, while of the bar, being out of the fire, is heated up less and less as we ap the other end. As soon as the first piece is at a good white heat, w course burns a high carbon steel, and the temperature of the rest of gradually passes down to a very dull red, the metal should be take the fire and suddenly plunged in cold water, in which it should be quite cold. It should then be taken out and carefully dried. An ex tion with a file will show that the first piece has the greatest ha while the last piece is the softest, the intermediate pieces gradually from one condition to the other. On now breaking off the pieces nick it will be seen that very considerable and characteristic chang been produced in the appearance of the metal. The first burnt piec open or crystalline in fracture; the succeeding pieces become clo closer in the grain until one piece is found to possess that p even grain and velvet-like appearance which is so much prized by enced steel users. The first pieces also, which have been too muc ened, will probably be cracked; those at the other end will not be ha through. Hence if it be desired to make the steel hard and str temperature used must be high enough to harden the metal throu not sufficient to open the grain.

Changes in Ultimate Strength and Elasticity d Hammering, Annealing, and Tempering. (J. W. L Trans. A. S. C. E. 1892.)—The following table gives the result of tes on some round steel bars, all from the same ingot, which were te tensile stresses, and also by bending till fracture took place:

Number.	Treatment.	Angle of cold bend, degrees.	Carbon.		Diameter, in.	Elastic limit, pounds per square inch.	Tensile, pounds per square inch.	Elongation, per cent.
			Total.	Semi-graphite.				
1	Cold-hammered bar	153	1.25	.47	.575	92,420	141,500	2.00
2	Bar drawn black....	75	1.25	.47	.577	114,700	138,400	6.00
3	Bar annealed	175	1.31	.70	.580	68,110	98,410	10.00
4	Bar hardened and drawn black	30	1.09	.36	.578	152,800	248,700	8.33

The total carbon given in the table was found by the color test, w affected, not only by the total carbon, but by the condition of the c
The analysis of the steel was:

Silicon	.242	Manganese....................	
Phosphorus....	.02	Carbon (true total carbon, combustion)...	
Sulphur	.009		

Heating Tool Steel. (Miller, Metcalf & Parkin, 1877.)—Th three distinct stages or times of heating: First, for forging; seco hardening; third, for tempering.

The first requisite for a good heat for forging is a clean fire and p fuel, so that jets of hot air will not strike the corners of the piece; th fire should be regular, and give a good uniform heat to the whole pa forged. It should be keen enough to heat the piece as rapidly as and allow it to be thoroughly heated through, without being so fier overheat the corners.

Steel should not be left in the fire any longer than is necessary to clear through, as "soaking" in fire is very injurious; and, on the othe it is necessary that it should be hot through, to prevent surface cra

By observing these precautions a piece of steel may always be ... ng to even a h... heat, when there is much forgin

nd most economical of welding fluxes is clean, crude borax, be first thoroughly melted and then ground to fine powder.
teel is properly heated, it should be forged to shape as quickly nd just as the red heat is leaving the parts intended for cutting parts should be refined by rapid, light blows, continued until pears.
ond stage of heating, for hardening, great care should be used: ect the cutting edges and working parts from heating more the body of the piece; next, that the whole part to be hardened niformly through, without any part becoming visibly hotter r. A uniform heat, as low as will give the required hardness, r hardening.
variation of heat, which is great enough to be seen, there will tion in grain, which may be seen by breaking the piece; and h variation in temperature, there is a very good chance for a een. Many a costly tool is ruined by inattention to this point. of too high heat is to open the grain; to make the steel coarse. an irregular heat is to cause irregular grain, irregular strains,

s the piece is properly heated for hardening, it should be l thoroughly quenched in plenty of the cooling medium, water, as the case may be.
nce of the cooling bath, to do the work quickly and uniformly ery necessary to good and safe work.
a large piece safely a running stream should be used.
en hardening is caused by the use of too small baths.
rd stage of heating, to temper, the first important requisite is mity. The next is time; the more slowly a piece is brought emper, the better and safer is the operation.
nsive tools are to be made it is a wise precaution to try small steel at different temperatures, so as to find out how low a heat necessary hardness. The lowest heat is the best for any steel.
to Forge.—The trouble in the forge fire is usually uneven t too high heat. Suppose the piece to be forged has been put ot fire, and forced as quickly as possible to a high yellow heat, lmost up to the scintillating point. If this be done, in a few outside will be quite soft and in a nice condition for forging, ddle parts will not be more than red-hot. Now let the piece be r the hammer and forged, and the soft outside will yield so readily than the hard inside, that the outer particles will be torn ile the inside will remain sound.
e case to be reversed and the inside to be much hotter than the t is, that the inside shall be in a state of semi-fusion, while the rd and firm. Now let the piece be forged, and the outside will d and the whole piece will appear perfectly good until it is l then it is found to be hollow inside.
ase, if the piece had been heated soft all through, or if it had been all through, it would have forged perfectly sound.
ses a high heat is more desirable to save heavy labor but in here a fine steel is to be used for cutting purposes it must be d that very heavy forging refines the bars as they slowly cool. mith heats such refined bars until they are soft, he raises the s them coarse, and he cannot get them fine again unless he has y steam-hammer at command and knows how to use it well.
ing. (Miller, Metcalf & Parkin.)—Annealing or softening is d by heating steel to a red heat and then cooling it very slowly, from getting hard again.
r the degree of heat, the more will steel be softened, until the ness is reached, when the steel is melted.
t follow that the higher a piece of steel is heated the softer it n cooled, no matter how slowly it may be cooled; this is proved hat an ingot is always harder than a rolled or hammered bar t.
there is nothing gained by heating a piece of steel hotter than ht, cherry-red; on the contrary, a higher heat has several dis- First. If carried too far, it may leave the steel actually harder l red heat would leave it. Second. If a scale is raised on the ale will be harsh, *granular oxide of iron*, and will spoil the tools t it. *Third.* A high scaling heat continued for a long time

If any number of forces be applied at a point, some in one direction and others in a contrary direction, their resultant is equal to the sum of those that act in one direction, diminished by the sum of those that act in the opposite direction; or, the resultant is equal to the algebraic sum of the components.

Parallelogram of Forces.—If two forces acting on a point be represented in direction and intensity by adjacent sides of a parallelogram their resultant will be represented by that diagonal of the parallelogram which passes through the point. Thus *OR*, Fig. 88, is the resultant of *OQ* and *OP*.

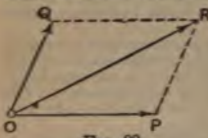

FIG. 88.

Polygon of Forces.—If several forces are applied at a point and act in a single plane, the resultant is found as follows:

Through the point draw a line representing the first force; through the extremity of this draw a line representing the second force; and so on throughout the system; finally, draw a line from the starting-point to the extremity of the last line drawn, and this will be the resultant required.

Suppose the body *A*, Fig. 89, to be urged in the directions *A*1, *A*2, *A*3, *A*4 and *A*5 by forces which are to each other as the lengths of those lines. Suppose these forces to act successively and the body to first move from A to 1: the second force *A*2 then acts and finding the body at 1 would take it to 2′; the third force would then carry it to 3′, the fourth to 4′, and the fifth to 5′. The line *A*5′ represents in magnitude and direction the resultant of all the forces considered. If there had been an additional force, *Ax*, in the group, the body would be returned by that force to its original position, supposing the forces to act successively, but if they had acted simultaneously the body would never have moved at all; the tendencies to motion balancing each other.

FIG. 89.

It follows, therefore, that if the several forces which tend to move a body can be represented in magnitude and direction by the sides of a closed polygon taken in order, the body will remain at rest; but if the forces are represented by the sides of an open polygon, the body will move and the direction will be represented by the straight line which closes the polygon.

Twisted Polygon.—The rule of the polygon of forces holds true when the forces are not in one plane. In this case the lines *A*1, 1-2′, 2′-3′ etc., form a twisted polygon, that is, one whose sides are not in one plane.

Parallelopipedon of Forces.—If three forces acting on a point be represented by three edges of a parallelopipedon which meet in a common point, their resultant will be represented by the diagonal of the parallelopipedon that passes through their common point.

Thus *OR*, Fig. 90, is the resultant of *OQ*, *OS*, and *OP*. *OM* is the resultant of *OP* and *OQ*, and *OR* is the resultant of *OM* and *OS*.

Moment of a Force.—The moment of a force (sometimes called statical moment), with respect to a point, is the product of the force by the perpendicular distance from the point to the direction of the force. The fixed point is called the centre of mo-

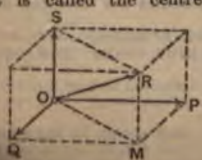

FIG. 90.

FIG. 91.

pendicular distance is the lever-arm of the force; and the
neasures the tendency of the force to produce rotation about
oments.
s expressed in pounds and the distance in feet, the moment
foot-pounds. It is necessary to observe the distinction be-
nds of statical moment and foot-pounds of work or energy.

ver, Fig. 91 (from Trautwine), if the weights n and m repre-
ir moments about the point f are respectively $n \times af$ and
stead of the weight m a pulling force to balance the weight
the direction bs, or by or bd, s, y, and d being the amounts of
eir respective moments are $s \times ft$, $y \times fb$, $d \times fh$.
acting on the lever are in equilibrium it remains at rest, and
each side of f are equal, that is, $n \times af = m \times fc$, or $s \times ft$,
$\times hf$.
of the resultant of any number of forces acting together in
is equal to the algebraic sum of the moments of the forces
y.

oment. Stability.—The statical moment of a body is
its weight by the distance of its line of gravity from some
rotation. The line of gravity is a vertical line drawn from
avity through the body. The stability of a body is that re-
its weight alone enables it to oppose against forces tending
r to slide it along its foundation.
inst turning on an edge the moment of the forces tending to
en with reference to that edge, must be less than the stati-
When a body rests on an inclined plane, the line of gravity
alls toward the lower edge of the body, and the condition of
erturned by its own weight is that the line of gravity must
edge. In the case of an inclined tower resting on a plane
ion holds—the line of gravity must fall within the base. The
bility against sliding along a horizontal plane is that the hor-
ent of the force exerted tending to cause it to slide shall be
oduct of the weight of the body into the coefficient of fric-
e base of the body and its supporting plane. This coefficient
e tangent of the angle of repose, or the maximum angle at
rting plane might be raised from the horizontal before the
in to slide. (See Friction.)

ity of a Dam against overturning about its lower edge
comparing its statical moment referred to that edge with
essure of the water against its upper side. The horizontal
uare foot at the bottom of the dam is equal to the weight of
er of one square foot in section, and of a height equal to the
bottom below water-level; or, if H is the height, the pressure
er square foot $= 62.4 \times H$ lbs. At the water-level the pres-
it increases uniformly to the bottom, so that the sum of the
ertical strip one foot in breadth may be represented by the
e whose base is $62.4 \times H$ and whose altitude is H, or $62.4H^2 \div 2$.
avity of a triangle being $\frac{1}{3}$ of its altitude, the resultant of
al pressures may be taken as equivalent to the sum of the
at $\frac{1}{3}H$, and the moment of the sum of the pressures is
$H^3 \div 6$.

orces.—If two forces are parallel and act in the same direc-
ant is parallel to both, and lies between them, and the inten-
tant is equal to the sum of the intensities of the two forces.
the resultant of the forces n and m acts vertically down-
equal to $n + m$.
forces act at the extremities of a straight line and in the
he resultant divides the line joining the points of application
nts, inversely as the components. Thus in Fig. 91, $m : n$ ::
g. 92, $P : Q :: SN : SM$.
of two parallel forces
e directions is parallel
out both, on the side
ction of the greater,
is equal to the differ-
tensities of the two

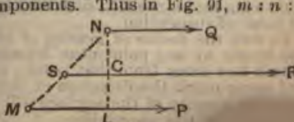

FIG. 92.

Thus the resultant of the two forces Q and P, Fig. 93, is equal
R. Of any two parallel force
resultant each is proportional
tance between the other two;
Figs. 92 and 93, $P : Q : R :: SN$

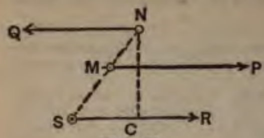

FIG. 93.

Couples.—If P and Q be e
in opposite directions, $R = 0$;
have no resultant. Two such
stitute what is called a couple

The tendency of a couple is
rotation; the measure of thi
called *the moment of the co*
product of one of the forces by the distance between the two.

Since a couple has no single resultant, no single force can
couple. To prevent the rotation of a body acted on by a couple
tion of two other forces is required, forming a second couple.
94, P and Q forming a couple, may be balanced by a second couple formed by R and S. The point of application of either R or S may be a fixed pivot or axis.

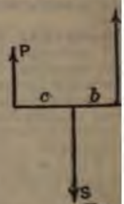

FIG.

Moment of the couple $PQ = P(c + b + a) =$ moment of $RS = Rb$. Also, $P + R = Q + S$.

The forces R and S need not be parallel to P and Q, but if not, then their components parallel to PQ are to be taken instead of the forces themselves.

Equilibrium of Forces.—A system of forces applied at points of a solid body will be in equilibrium when they have no tendency to produce motion, either of translation or of rotation.

The conditions of equilibrium are: 1. The algebraic sum of
nents of the forces in the direction of any three rectangular ax
separately equal to 0.

2. The algebraic sum of the moments of the forces, with res
three rectangular axes, must be separately equal to 0.

If the forces lie in a plane: 1. The algebraic sum of the compo
forces, in the direction of any two rectangular axes, must be
equal to 0.

2. The algebraic sum of the moments of the forces, with re
point in the plane, must be equal to 0.

If a body is restrained by a fixed axis, as in case of a pulley, o
axle, the forces will be in a equilibrium when the algebraic sum
ments of the forces with respect to the axis is equal to 0.

CENTRE OF GRAVITY.

The centre of gravity of a body, or of a system of bodies rigidl
together, is that point about which, if suspended, all the part
equilibrium, that is, there will be no tendency to rotation. It
through which passes the resultant of the efforts of gravitation
the elementary particles of a body. In bodies of equal heavin
out, the centre of gravity is the centre of magnitude.

(The centre of magnitude of a figure is a point such that if t
divided into equal parts the distance of the centre of magni
whole figure from any given plane is the mean of the distances of
of magnitude of the several equal parts from that plane.)

If a body be suspended at its centre of gravity, it will be in eq
all positions. If it be suspended at a point out of its centre of
will swing into a position such that its centre of gravity is vertic
its point of suspension.

To find the centre of gravity of any plane figure mechanical
the figure by any point near its edge, and mark on it the di
plumb-line hung from that point; then suspend it from some
and again m[illegible]rection of the plumb-line in like manner
[illegible] urface will be at the point of intersec
[illegible] ne.

[illegible] **ity of Regular Figures,** wheth
[illegible] eometrical centre; for instance, a

am, regular polygon, circle, circular ring, prism, cylinder, heroid, middle frustums of spheroid, etc.

ngle: On a line drawn from any angle to the middle of the op- at a distance of one third of the line from the side; or at the n of such lines drawn from any two angles.

pezium or trapezoid: Draw a diagonal, dividing it into two tri- raw a line joining their centres of gravity. Draw the other naking two other triangles, and a line joining their centres. The n of the two lines is the centre of gravity required.

or of a circle: On the radius which bisects the arc, $2cr \div 3l$ from c being the chord, r the radius, and l the arc.

icircle: On the middle radius, $.4244r$ from the centre.

drant: On the middle radius, $.6002r$ from the centre.

nent of a circle: $c^3 \div 12a$ from the centre. c = chord, a = area.

bolic surface: In the axis, 3/5 of its length from the vertex.

-parabola (surface): 3/5 length of the axis from the vertex, and mi-base from the axis.

or pyramid: In the axis, ¼ of its length from the base.

boloid: In the axis, ⅔ of its length from the vertex.

nder, or regular prism: In the middle point of the axis.

tum of a cone or pyramid: Let a = length of a line drawn from of the cone when complete to the centre of gravity of the base, and tion of it between the vertex and the top of the frustum; then centre of gravity of the frustum from centre of gravity of its

$$\frac{3a'^3}{4(a^2 + aa' + a'^2)}.$$

bodies, fixed one at each end of a straight bar, the common ravity is in the bar, at that point which divides the distance eir respective centres of gravity in the inverse ratio of the n this solution the weight of the bar is neglected. But it may a third body, and allowed for as in the following directions:

than two bodies connected in one system: Find the common avity of two of them; and find the common centre of these two a third body, and so on to the last body of the group.

method, by the principle of moments: To find the centre of system of bodies, or a body consisting of several parts, whose tres are known. If the bodies are in a plane, refer their several two rectangular co-ordinate axes. Multiply each weight by its m one of the axes, add the products, and divide the sum by the weights: the result is the distance of the centre of gravity from Do the same with regard to the other axis. If the bodies are ne, refer them to three planes at right angles to each other, and the mean distance of the sum of the weights from each of the s.

MOMENT OF INERTIA.

ent of inertia of the weight of a body with respect to an axis is ic sum of the products obtained by multiplying the weight of tary particle by the square of its distance from the axis. If the inertia with respect to any axis = I, the weight of any element = w, and its distance from the axis = r, we have $I = \Sigma(wr^2)$.

ent of inertia varies, in the same body, according to the position It is the least possible when the axis passes through the centre To find the moment of inertia of a body, referred to a given the body into small parts of regular figure. Multiply the weight by the square of the distance of its centre of gravity from the um of the products is the moment of inertia. The value of the inertia thus obtained will be more nearly exact, the smaller and rous the parts into which the body is divided.

OF INERTIA OF REGULAR SOLIDS.—Rod, or bar, of uniform thick- espect to an axis perpendicular to the length of the rod,

$$I = W\left(\frac{l^2}{3} + d^2\right), \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

of rod, $2l$ = length, d = distance of centre of gravity from axis.

ar plate, axis in its $\Big\}$ $I = W\left(\frac{r^2}{4} + d^2\right);$ . . . .

f plate.

Circular plate, axis perpendicular to the plate, $I = W\left(\frac{r^2}{2} + d^2\right)$. . . . . .

Circular ring, axis perpendicular to its own plane, $I = W\left(\frac{r^2 + r'^2}{2} + d^2\right)$. . . .

r and r' are the exterior and interior radii of the ring.

Cylinder, axis perpendicular to the axis of the cylinder, $I = W\left(\frac{r^2}{4} + \frac{l^2}{3} + d^2\right)$. . . .

r = radius of base, $2l$ = length of the cylinder.

By making $d = 0$ in any of the above formulæ we find the mo inertia for a parallel axis through the centre of gravity.

The moment of inertia, Σwr^2, numerically equals the weight of which, if concentrated at the distance unity from the axis of rotation require the same work to produce a given increase of angular velocity actual body requires. It bears the same relation to angular acce which weight does to linear acceleration (Rankine). The term mo inertia is also used in regard to areas, as the cross-sections of beams strain. In this case $I = \Sigma ar^2$, in which a is any elementary area, distance from the centre. (See Moment of Inertia, under Strength terials, p. 247.)

CENTRE AND RADIUS OF GYRATION.

The centre of gyration, with reference to an axis, is a point at w the entire weight of a body be concentrated, its moment of inertia main unchanged; or, in a revolving body, the point in which the weight of the body may be conceived to be concentrated, as if a p platinum were substituted for a pound of revolving feathers, the velocity and the accumulated work remaining the same. The dist this point from the axis is the *radius of gyration*. If W = the weig body, $I = \Sigma wr^2$ = its moment of inertia, and k = its radius of gyrat

$$I = Wk^2 = \Sigma wr^2; \quad k = \sqrt{\frac{\Sigma wr^2}{W}}.$$

The moment of inertia = the weight × the square of the radius of gy

To find the radius of gyration divide the body into a considerable of equal small parts—the more numerous the more nearly exact is sult,—then take the mean of all the squares of the distances of th from the axis of revolution, and find the square root of the mean Or, if the moment of inertia is known, divide it by the weight and the square root. For radius of gyration of an area, as a cross-sect beam, divide the moment of inertia of the area by the area and ext square root.

The radius of gyration is the least possible when the axis passes the centre of gravity. This minimum radius is called the principa of gyration. If we denote it by k and any other radius of gyratio we have for the five cases given under the head of moment of inerti the following values:

(1) Rod, axis perpen. to length, $k = l\sqrt{\frac{1}{3}}$; $k' = \sqrt{\frac{l^2}{3} + d^2}$.

(2) Circular plate, axis in its plane, $k = \frac{r}{2}$; $k' = \sqrt{\frac{r^2}{4} + d^2}$.

(3) Circular plate, axis perpen. to plane, $k = r\sqrt{\frac{1}{2}}$; $k' = \sqrt{\frac{r^2}{2} + d^2}$.

(4) Circular ring, axis perpen. to plane, $k = \sqrt{\frac{r^2 + r'^2}{2}}$; $k' = \sqrt{\frac{r^2 + r'^2}{2} + \ldots}$

(5) Cylinder, axis per- pen. to [illegible] $k = \sqrt{\frac{r^2}{4} + \frac{l^2}{3}}$; $k' = \sqrt{\frac{r^2}{4} + \frac{l^2}{3} + \ldots}$

adii of Gyration and Squares of Radii of Gyration.

of gyration of sections of columns, see page 249.)

or Solid.	Rad. of Gyration.	Square of R. of Gyration.
xis at its base.......	$.5773h$	$\frac{1}{3}h^2$
" mid-height.......	$.2886h$	$1/12h^2$
axis at end.......	$.5773l$	$\frac{1}{3}l^2$
" mid-length..	$.2886l$	$1/12l^2$
n:		
eferred to axis $2a$...	$.577\sqrt{b^2+c^2}$	$(b^2+c^2) \div 3$
ngth l, base b, axis id-breadth.........	$.289\sqrt{4l^2+b^2}$	$\frac{4l^2+b^2}{12}$
be:		
h', axis mid-length..	$.289\sqrt{h^2+h'^2}$	$(h^2+h'^2) \div 12$
$= h$, " " ..	$.408h$	$h^2 \div 6$
tube: sides b, h, ...	$.289h\sqrt{\frac{h+3b}{h+b}}$	$\frac{h^2}{12} \cdot \frac{h+3b}{h+b}$
d. r, diam. h, ax. diam.	$\frac{1}{2}r$	$\frac{1}{4}r^2 = h^2 \div 16$
ms. h, h', axis diam.	$\frac{1}{4}\sqrt{h^2+h'^2}$	$(h^2+h'^2) \div 16$
linder: length l, mid-length.......	$.289\sqrt{l^2+3r^2}$	$\frac{l^2}{12}+\frac{r^2}{4}$
olid wheel of uni- or cylinder of any to axis of cyl.....	$.7071r$	$\frac{1}{2}r^2$
nder, or flat ring: , outer and inner	$.7071\sqrt{R^2+r^2}$	$(R^2+r^2) \div 2$
longitudinal axis; length........	$.289\sqrt{l^2+3(R^2+r^2)}$	$\frac{l^2}{12}+\frac{R^2+r^2}{4}$
axis its diameter. ...	$.289\sqrt{l^2+6R^2}$	$\frac{l^2}{12}+\frac{R^2}{2}$
xis, longitud'l axis..	r	r^2
, axis its centre.....	r	r^2
" " diam.......	$.7071r$	$\frac{1}{2}r^2$
axis its diam........	$.6325r$	$2/5r^2$
orial radius r, re- tis a............. ...	$.6325r$	$2/5r^2$
rad. of base, rev.	$.5773r$	$\frac{1}{3}r^2$
xes a, b, c; revolv-	$.4472\sqrt{b^2+c^2}$	$\frac{b^2+c^2}{5}$
adii R, r, revolving	$.6325\sqrt{\frac{R^5-r^5}{R^3-r^3}}$	$\frac{2}{5}\frac{R^5-r^5}{R^3-r^3}$
radius r	$.8165r$	$\frac{2}{3}r^2$
d. of base, rev. on	$.5477r$	$0.3r^2$

F OSCILLATION AND OF PERCUSSION.

scillation.—If a body oscillate about a fixed horizontal through its centre of gravity, there is a point in the line entre of gravity perpendicular to the axis whose motion would be if the whole mass were collected at that point ibrate as a pendulum about the fixed axis. This point is of oscillation.

of Oscillation, or distance of the centre of oscillation f suspension = the square of the radius of gyration ÷ dis- re of gravity from the point of suspension or axis. The ion and suspension are convertible.

e, or *uniform thin bar or cylinder*, be susp *t as an axis, the centre of oscillation is* a the rod from the axis. If the point of suspension is at ⅙ the lengt the end, the centre of oscillation is also at ⅚ the length from the ax is, it is at the other end. In both cases the oscillation will be perfo the same time. If the point of suspension is at the centre of grav length of the equivalent simple pendulum is infinite, and therefore t of vibration is infinite.

For a sphere suspended by a cord, $r =$ radius, $h =$ distance of motion from the centre of the sphere, $h' =$ distance of centre of os from centre of the sphere, $l =$ radius of oscillation $= h + h' = h +$

If the sphere vibrate about an axis tangent to its surface, $h = r$, a $+ 2/5r$. If $h = 10r$, $l = 10r + \frac{r}{25}$.

Lengths of the radius of oscillation of a few regular plane figures plates, suspended by the vertex or uppermost point.

1st. When the vibrations are flatwise, or perpendicular to the plan figure:

In an isosceles triangle the radius of oscillation is equal to ¾ of th of the triangle.

In a circle, ⅝ of the diameter.

In a parabola, 5/7 of the height.

2d. When the vibrations are edgewise, or in the plane of the figur

In a circle the radius of oscillation is ¾ of the diameter.

In a rectangle suspended by one angle, ⅔ of the diagonal.

In a parabola, suspended by the vertex, 5/7 of the height, plus ⅓ parameter.

In a parabola, suspended by the middle of the base, 4/7 of the hei ½ the parameter.

Centre of Percussion.—The centre of percussion of a body ing about a fixed axis is the point at which, if a blow is struck by t the percussive action is the same as if the whole mass of the body w centrated at the point. This point is identical with the centre of os

THE PENDULUM.

A body of any form suspended from a fixed axis about which it by the force of gravity is called a *compound pendulum*. The ide concentrated at the centre of oscillation suspended from the centr pension by a string without weight, is called a *simple pendulum*. T valent simple pendulum has the same weight as the given body, the same moment of inertia, referred to an axis passing through of suspension, and it oscillates in the same time.

The ordinary pendulum of a given length vibrates in equal times angle of the vibrations does not exceed 4 or 5 degrees, that is, 2° or side of the vertical. This property of a pendulum is called its isoc

The time of vibration of a pendulum varies directly as the squar the length, and inversely as the square root of the acceleration du ity at the given latitude and elevation above the earth's surface.

If $T =$ the time of vibration, $l =$ length of the simple pendulum, eration $= 32.16$, $T = \pi \sqrt{\frac{l}{g}}$; since π is constant, $T \propto \frac{\sqrt{l}}{\sqrt{g}}$. At a gi tion g is constant and $T \propto \sqrt{l}$. If l be constant, then for any $T \propto \frac{1}{\sqrt{g}}$. If T be constant, $gT^2 = \pi^2 l$; $l \propto g$; $g = \frac{\pi^2 l}{T^2}$. From this the force of gravity at any place may be determined if the leng simple pendulum, vibrating seconds, at that place is known. At N this length is 39.1017 inches = 3.2585 ft., whence $g = 32.16$ ft. At L length is 39.1393 inches. At the equator 39.0152 or 39.0168 inches, to different authorities.

Time of vibration of a pendulum of a given length at New York

$$= t = \sqrt{\frac{l}{39.1017}} = \frac{\sqrt{l}}{6.253},$$

... in seconds and l in inches. Length of a pendulum havin ... vibration, $l = t^2 \times 39.1017$ inches.

f vibration of a pendulum may be varied by the addition of a point above the centre of suspension, which counteracts the , and lengthens the period of vibration. By varying the height weight the time is varied.
weight of the upper bob of a compound pendulum, vibrating n the weight of the lower bob, and the distances of the weights it of suspension are given:

$$w = W \frac{(39.1 + D) - D^2}{(39.1 + d) + d^2}.$$

eight of the lower bob, $w =$ the weight of the upper bob; $D =$ of the lower bob and $d =$ the distance of the upper bob from uspension, in inches.
eans of a second bob, short pendulums may be constructed to wly as longer pendulums.
ng w or d until the lower weight is entirely counterbalanced, ibration may be made infinite.
Pendulum.—A weight suspended by a cord and revolving speed in the circumference of a circular horizontal plane is r, the distance of the plane below the point of suspension be- in equilibrium by three forces—the tension in the cord, the cen- , which tends to increase the radius r, and the force of gravity ward. If $v =$ the velocity in feet per second, the centre of e weight, as it describes the circumference, $g = 32.16$, and r en in feet, the time in seconds of performing one revolution is

$$t = \frac{2\pi r}{v} = 2\pi \sqrt{\frac{h}{g}}; \quad h = \frac{gt^2}{4\pi^2} = .8146t^2.$$

ond, $h = .8146$ foot $= 9.775$ inches.
ple of the conical pendulum is used in the ordinary fly-ball steam-engines. (See Governors.)

CENTRIFUGAL FORCE.

olving in a curved path of radius $= R$ in feet exerts a force, fugal force, F, upon the arm or cord which restrains it from straight line, or "flying off at a tangent." If $W =$ weight of pounds, $N =$ number of revolutions per minute, $v =$ linear he centre of gravity of the body, in feet per second, $g = 32.16$,

$$F = \frac{Wv^2}{gR} = \frac{Wv^2}{32.16R} = \frac{W4\pi^2RN^2}{3600g} = \frac{WRN^2}{2933} = .0003410WRN^2 \text{ lbs.}$$

ber of revolutions per second, $F = 1.2276\,WRn^2$.
fugal force in fly-wheels, see Fly-wheels.)

ITY, ACCELERATION, FALLING BODIES.

is the rate of motion, or the distance passed over by a body in

e in feet passed over in t seconds, and $v =$ velocity in feet per velocity is uniform,

$$v = \frac{s}{t}; \quad s = vt; \quad t = \frac{s}{v}.$$

ity varies uniformly, the mean velocity $v_0 = \frac{v_1 + v_2}{2}$, in which ity at the beginning and v_2 the velocity at the end of the time t.

$$s = \frac{v_1 + v_2}{2}t. \quad \ldots \ldots \ldots \ldots (1)$$

tion is the change in velocity which takes place in a unit of of acceleration $= a = 1$ foot per second in one ... For ying velocity, the acceleration is a constant q...

$$\frac{v_2 - v_1}{t}; \quad v_2 = v_1 + at; \quad v_1 = v_2 - at; \quad t = \frac{v_2}{}$$

If the body start from rest, $v_1 = 0$; then

$$v_0 = \frac{v^2}{2};\quad v_2 = 2v_0;\quad a = \frac{v_2}{t};\quad v_2 = at;\quad v_2 - at = 0;\quad t = \frac{v_2}{a}.$$

Combining (1) and (2), we have

$$s = \frac{v_2^2 - v_1^2}{2a};\quad s = v_1 t + \frac{at^2}{2};\quad s = v_2 t - \frac{at^2}{2}.$$

If $v_1 = 0$, $s = \frac{v_2}{2}t$.

Retarded Motion.—If the body start with a velocity v_1 and come [illegible] rest, $v_2 = 0$; then $s = \frac{v_1}{2}t$.

In any case, if the change in velocity is v,

$$s = \frac{v}{2}t;\quad s = \frac{v^2}{2a};\quad s = \frac{a}{2}t^2.$$

For a body starting from or ending at rest, we have the equations

$$v = at;\quad s = \frac{v}{2}t;\quad s = \frac{at^2}{2};\quad v^2 = 2as.$$

Falling Bodies.—In the case of falling bodies the acceleration [illegible] to gravity is 32.16 feet per second in one second, $= g$. Then if v = velocity acquired at the end of t seconds, or final velocity, and h = height or space in feet passed over in the same time,

$$v = gt = 32.16t = \sqrt{2gh} = 8.02\sqrt{h} = \frac{2h}{t};$$

$$h = \frac{gt^2}{2} = 16.08t^2 = \frac{v^2}{2g} = \frac{v^2}{64.32} = \frac{vt}{2};$$

$$t = \frac{v}{g} = \frac{v}{32.16} = \sqrt{\frac{2h}{g}} = \frac{\sqrt{h}}{4.01} = \frac{2h}{v};$$

u = space fallen through in the Tth second $= g(T - \tfrac{1}{2})$.

Value of g.—The value of g increases with the latitude, and decreases with the elevation. At the latitude of Philadelphia, 40°, its value is 32.16 at the sea-level. Everett gives $g = 32.173 - .082 \cos 2 \text{ lat.} - .000003$ height in feet. At Paris, lat. 48° 50′ N., g = 980.87 cm. = 32.181 ft.

Values of $\sqrt{2g}$, calculated by an equation given by C. S. Pierce, are given in a table in Smith's Hydraulics, from which we take the following:

Latitude	0°	10°	20°	30°	40°	50°	[illegible]
Value of $\sqrt{2g}$..	8.0112	8.0118	8.0137	8.0165	8.0199	8.0235	[illegible]

The value of $\sqrt{2g}$ decreases about .0004 for every 1000 feet increase in elevation above the sea-level.

For all ordinary calculations for the United States, g is generally taken at 32.16, and $\sqrt{2g}$ at 8.02. In England $g = 32.2$, $\sqrt{2g} = 8.025$. Practical limiting values of g for the United States, according to Pierce, are:

Latitude 49° at sea-level .... $g = 32.186$
" 25° 10,000 feet above the sea .... $g = 32.089$

From the above formula for falling bodies we obtain the following:

During the first second the body starting from a state of rest (resistance of the air neglected) falls $g \div 2 = 16.08$ feet; the acquired velocity is 32.16 ft. per sec.; the distance fallen in two seconds is $h = \frac{gt^2}{2} = 16.08 \times 4 =$ 64.32 ft.; and the acquired velocity is $v = gt = 64.32$ ft. The acceleration, or increase of velocity in each second, is constant, and is 32.16 ft. per sec. Solving the equations for different times, we find for

Seconds, t	..		1	2	3	4	5
Acceleration, g	32.16	×	1	1	1	1	1
Velocity acquired at end of time, v....	32.16	×	1	2	3	4	5
Height of fall in each second, u......	$\frac{32.16}{2}$	×	1	3	5	7	9
[illegible] height of fall, h..................	$\frac{32.16}{2}$	×	1	4	9	16	25

nts graphically the velocity, space, etc., of a body falling for e vertical line at the left is onds, the horizontal lines lf the acquired velocities ich second. The area of gle at the top represents en throug.. in the first 16.08 feet, and each of the is an equal space. The gles between each pair of represents the height of cond, and the number of n any horizontal line and otal height fallen during figures under h, u, and v it are to be multiplied by the actual velocities and given times.

h	u	v	t
1	1	2	1″
4	3	4	2″
9	5	6	3″
16	7	8	4″
25	9	10	5″
36	11	12	6″

FIG. 95.

nd Linear Velocity

ody.—Let r = radius of a feet, n = number of revo- ute, v = linear velocity of circumference in feet per second, and $60v$ = velocity in feet

$$v = \frac{2\pi rn}{60}, \quad 60v = 2\pi rn.$$

city is a term used to denote the angle through which any ly turns in a second, or the rate at which any point in it equal to unity is moving, expressed in feet per second. The velocity is the angle which at a distance = radius from the nded by an arc equal to the radius. This unit angle = $\frac{180}{\pi}$. $2\pi \times 57.3° = 360°$, or the circumference. If A = angular r, $A = \frac{v}{r} = \frac{2\pi n}{60}$.

orresponding to a Given Acquired Velocity.

Velocity.	Height.	Velocity.	Height.	Velocity.	Height.	Velocity.	Height.	Velocity.	Height.
eet sec.	feet.	feet p.sec.	feet.	feet p.sec.	feet.	feet p.sec.	feet.	feet p.sec.	feet.
13	2.62	34	17.9	55	47.0	76	89.8	97	146
14	3.04	35	19.0	56	48.8	77	92.2	98	149
15	3.49	36	20.1	57	50.5	78	94.6	99	152
16	3.98	37	21.3	58	52.3	79	97.0	100	155
17	4.49	38	22.4	59	54.1	80	99.5	105	171
18	5.03	39	23.6	60	56.0	81	102.0	110	188
19	5.61	40	24.9	61	57.9	82	104.5	115	205
20	6.22	41	26.1	62	59.8	83	107.1	120	224
21	6.85	42	27.4	63	61.7	84	109.7	130	263
22	7.52	43	28.7	64	63.7	85	112.3	140	304
23	8.21	44	30.1	65	65.7	86	115.0	150	350
24	8.94	45	31.4	66	67.7	87	117.7	175	476
25	9.71	46	32.9	67	69.8	88	120.4	200	622
26	10.5	47	34.3	68	71.9	89	123.2	300	1399
27	11.3	48	35.8	69	74.0	90	125.9	400	2488
28	12.2	49	37.3	70	76.2	91	128.7	500	3887
29	13.1	50	38.9	71	78.4	92	131.6	600	5597
30	14.0	51	40.4	72	80.6	93	134.5	700	7618
31	14.9	52	42.0	73	82.9	94	137.4	800	9952
32	15.9	53	43.7	74	85.1	95	140.3	[illegible]	[illegible]
33	16.9	54	45.3	75	87.5	96	143.[illegible]	[illegible]	[illegible]

Falling Bodies: Velocity Acquired by a Body Fall[ing a] Given Height.

Height.	Velocity.	Height.	Velocity.	Height.	Velocity.	Height.	Velocity.	Height.	Velocity.	Height.
feet.	feet p.sec.	feet.	feet p.sec.	feet.	feet p.sec.	feet.	feet p.sec.	feet.	feet p.sec.	feet.
.005	.57	.39	5.01	1.20	8.79	5.	17.9	23.	38.5	72
.010	.80	.40	5.07	1.22	8.87	.2	18.3	.5	38.9	73
.015	.98	.41	5.14	1.24	8.94	.4	18.7	24.	39.3	74
.020	1.13	.42	5.20	1.26	9.01	.6	19.0	.5	39.7	75
.025	1.27	.43	5.26	1.28	9.08	.8	19.3	25	40.1	76
.030	1.39	.44	5.32	1.30	9.15	6.	19.7	26	40.9	77
.035	1.50	.45	5.38	1.32	9.21	.2	20.0	27	41.7	78
.040	1.60	.46	5.44	1.34	9.29	.4	20.3	28	42.5	79
.045	1.70	.47	5.50	1.36	9.36	.6	20.6	29	43.2	80
.050	1.79	.48	5.56	1.38	9.43	.8	20.9	30	43.9	81
.055	1.88	.49	5.61	1.40	9.49	7.	21.2	31	44.7	82
.060	1.97	.50	5.67	1.42	9.57	.2	21.5	32	45.4	83
.065	2.04	.51	5.73	1.44	9.62	.4	21.8	33	46.1	84
.070	2.12	.52	5.78	1.46	9.70	.6	22.1	34	46.8	85
.075	2.20	.53	5.84	1.48	9.77	.8	22.4	35	47.4	86
.080	2.27	.54	5.90	1.50	9.82	8.	22.7	36	48.1	87
.085	2.34	.55	5.95	1.52	9.90	.2	23.0	37	48.8	88
.090	2.41	.56	6.00	1.54	9.96	.4	23.3	38	49.4	89
.095	2.47	.57	6.06	1.56	10.0	.6	23.5	39	50.1	90
.100	2.54	.58	6.11	1.58	10.1	.8	23.8	40	50.7	91
.105	2.60	.59	6.16	1.60	10.2	9.	24.1	41	51.4	92
.110	2.66	.60	6.21	1.65	10.3	.2	24.3	42	52.0	93
.115	2.72	.62	6.32	1.70	10.5	.4	24.6	43	52.6	94
.120	2.78	.64	6.42	1.75	10.6	.6	24.8	44	53.2	95
.125	2.84	.66	6.52	1.80	10.8	.8	25.1	45	53.8	96
.130	2.89	.68	6.61	1.90	11.1	10.	25.4	46	54.4	97
.14	3.00	.70	6.71	2.	11.4	.5	26.0	47	55.0	98
.15	3.11	.72	6.81	2.1	11.7	11.	26.6	48	55.6	99
.16	3.21	.74	6.90	2.2	11.9	.5	27.2	49	56.1	100
.17	3.31	.76	6.99	2.3	12.2	12.	27.8	50	56.7	125
.18	3.40	.78	7.09	2.4	12.4	.5	28.4	51	57.3	150
.19	3.50	.80	7.18	2.5	12.6	13.	28.9	52	57.8	175
.20	3.59	.82	7.26	2.6	12.9	.5	29.5	53	58.4	200
.21	3.68	.84	7.35	2.7	13.2	14.	30.0	54	59.0	225
.22	3.76	.86	7.44	2.8	13.4	.5	30.5	55	59.5	250
.23	3.85	.88	7.53	2.9	13.7	15.	31.1	56	60.0	275
.24	3.93	.90	7.61	3.	13.9	.5	31.6	57	60.6	300
.25	4.01	.92	7.69	3.1	14.1	16.	32.1	58	61.1	350
.26	4.09	.94	7.78	3.2	14.3	.5	32.6	59	61.6	400
.27	4.17	.96	7.86	3.3	14.5	17.	33.1	60	62.1	450
.28	4.25	.98	7.94	3.4	14.8	.5	33.6	61	62.7	500
.29	4.32	1.00	8.02	3.5	15.0	18.	34.0	62	63.2	550
.30	4.39	1.02	8.10	3.6	15.2	.5	34.5	63	63.7	600
.31	4.47	1.04	8.18	3.7	15.4	19.	35.0	64	64.2	700
.32	4.54	1.06	8.26	3.8	15.6	.5	35.4	65	64.7	800
.33	4.61	1.08	8.34	3.9	15.8	20.	35.9	66	65.2	900
.34	4.68	1.10	8.41	4.	16.0	.5	36.3	67	65.7	1000
.35	4.74	1.12	8.49	.2	16.4	21.	36.8	68	66.1	2000
.36	4.81	1.14	8.57	.4	16.8	.5	37.2	69	66.6	3000
.37	4.88	1.16	8.64	.6	17.2	22.	37.6	70	67.1	4000
.38	4.94	1.18	8.72	.8	17.6	.5	38.1	71	67.6	5000

Parallelogram of Velocities.—The principle of the com[position] and resolution of forces may also be applied to velocities or to [spaces] moved in given intervals of time. Referring to Fig. 88, page 41[.], [a body] at *O* has a force applied to it which acting alone would give [it a velocity] represented by *OQ* per second, and at the same time it is

Another force which acting alone would give it a velocity OP per second, the result of the two forces acting together for one second will carry it to R, OR being the diagonal of the parallelogram of OQ and OP, and the resultant velocity. If the two component velocities are uniform, the resultant will be uniform and the line OR will be a straight line; but if either velocity is a varying one, the line will be a curve. Fig. 96 shows the resultant velocities, also the path traversed by a body acted on by two forces, one of which would carry it at a uniform velocity over the intervals 1, 2, 3, B, and the other of which would carry it by an accelerated motion over the intervals a, b, c, D in the same times. At the end of the respective intervals the body will be found at C_1, C_2, C_3, C, and the mean velocity during each interval is represented by the distances between these points. Such a curved path is traversed by a shot, the impelling force from the gun giving it a uniform velocity in the direction the gun is aimed, and gravity giving it an accelerated velocity downward. *The path of a projectile is a parabola.* The distance it will travel is greatest when its initial direction is at an angle 45° above the horizontal.

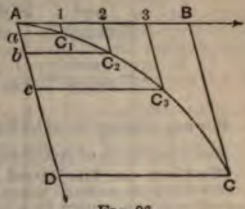

FIG. 96.

Mass—Force of Acceleration.—*The mass* of a body, or the quantity of matter it contains, is a constant quantity, while the weight varies according to the variation in the force of gravity at different places. If g = the acceleration due to gravity, and w = weight, then the mass $m = \frac{w}{g}$, $w = mg$. Weight here means the resultant of the force of gravity on the particles of a body, such as may be measured by a spring-balance, or by the extension or deflection of a rod of metal loaded with the given weight.

Force has been defined as that which causes, or tends to cause, or to destroy, motion. It may also be defined (Kennedy's Mechanics of Machinery) as the cause of acceleration; and the unit of force as the force required to produce unit acceleration in a unit of free mass.

Force equals the product of the mass by the acceleration, or $f = ma$.

Also, if v = the velocity acquired in the time t, $ft = mv$; $f = mv \div t$; the acceleration being uniform.

The force required to produce an acceleration of g (that is, 32.16 ft. per sec.) in one second is $f = mg = \frac{w}{g}g = w$, or the weight of the body. Also, $f = ma = m\frac{v_2 - v_1}{t}$, in which v_2 is the velocity at the end, and v_1 the velocity at the beginning of the time t, and $f = mg = \frac{w}{g}\frac{(v_2 - v_1)}{t} = \frac{w}{g}a$; $\frac{f}{w} = \frac{a}{g}$; or, the force required to give any acceleration to a body is to the weight of the body as that acceleration is to the acceleration produced by gravity. (The weight w is the weight where g is measured.)

EXAMPLE.—Tension in a cord lifting a weight. A weight of 100 lbs. is lifted vertically by a cord a distance of 80 feet in 4 seconds, the velocity uniformly increasing from 0 to the end of the time. What tension must be maintained in the cord? Mean velocity $= v_0 = 20$ ft. per sec.; final velocity $= v_1 = 2v_0 = 40$; acceleration $a = \frac{v_2}{t} = \frac{40}{4} = 10$. Force $f = ma = \frac{wa}{g} = \frac{100}{32.16} \times 10 = 31.1$ lbs. This is the force required to produce the acceleration only; to it must be added the force required to lift the weight without acceleration, or 100 lbs., making a total of 131.1 lbs.

The Resistance to Acceleration is the same as the force required to produce the acceleration $= \frac{w}{g}\frac{(v_2 - v_1)}{t}$.

Formulæ for Accelerated Motion.—For cases of uniformly accelerated motion other than those of falling bodies, we have the formulæ ... given, $f = \frac{w}{g}a$, $= \frac{w}{g}\frac{v_2 - v_1}{t}$. If the body starts from rest, $v_1 = 0$, v_2

$= v$, and $f = \frac{w}{g}\frac{v}{t}$, $fgt = wv$. We also have $s = \frac{vt}{2}$. Transforming and substituting for g its value 32.16, we obtain

$$f = \frac{wv^2}{64.32s} = \frac{wv}{32.16t} = \frac{ws}{16.08t^2}; \quad w = \frac{32.16ft}{v} = \frac{64.32fs}{v^2};$$

$$s = \frac{wv^2}{64.32f} = \frac{16.08ft^2}{w} = \frac{vt}{2}; \quad v = 8.02\sqrt{\frac{fs}{w}} = \frac{32.16ft}{w};$$

$$t = \frac{wv}{32.16f} = \frac{1}{4.01}\sqrt{\frac{ws}{f}}$$

For any change in velocity $f = w\left(\frac{v_2^2 - v_1^2}{64.32s}\right)$.

(See also Work of Acceleration, under Work.)

Motion on Inclined Planes.—The velocity acquired by a body descending an inclined plane by the force of gravity (friction neglected) is equal to that acquired by a body falling freely from the height of the plane.

The times of descent down different inclined planes of the same height vary as the length of the planes.

The rules for uniformly accelerated motion apply to inclined planes. If a is the angle of the plane with the horizontal, $\sin a$ = the ratio of the height to the length $= \frac{h}{l}$, and the constant accelerating force is $g \sin a$. The velocity at the end of t seconds is $v = gt \sin a$. The distance passed over in t seconds is $l = \frac{1}{2} gt^2 \sin a$. The time of descent is

$$t = \sqrt{\frac{2l}{g \sin a}} = \frac{l}{4.01\sqrt{h}}.$$

MOMENTUM, VIS-VIVA.

Momentum, or quantity of motion in a body, is the product of the mass by the velocity at any instant $= mv = \frac{w}{g}v$.

Since the moving force = product of mass by acceleration, $f = ma$; and if the velocity acquired in t seconds $= v$, or $a = \frac{v}{t}$, $f = \frac{mv}{t}$; $ft = mv$; that is, the product of a constant force into the time in which it acts equals numerically the momentum.

Since $ft = mv$, if $t = 1$ second $mv = f$, whence momentum might be defined as numerically equivalent to the number of pounds of force that will stop a moving body in 1 second, or the number of pounds of force which acting during 1 second will give it the given velocity.

Vis-viva, or living force, is a term used by early writers on Mechanics to denote the energy stored in a moving body. Some defined it as the product of the mass into the square of the velocity, mv^2, $= \frac{w}{g}v^2$ others as one half of this quantity or $\frac{1}{2}mv^2$, or the same as what is now known as energy. The term is now practically obsolete, its place being taken by the word energy.

WORK, ENERGY, POWER.

Work is the overcoming of resistance through a certain distance. It is measured by the product of the resistance into the space through which it is overcome. It is also measured by the product of the moving force into the distance through which the force acts in overcoming the resistance. Thus in lifting a body from the earth against the attraction of gravity, the resistance is the weight of the body, and the product of this weight into the height the body is lifted is the work done.

Unit of Work.—In British measures, is the *foot-pound*, or the *unit of work* ... overcoming a pressure or weight equal to ... *through* ...

work performed by a piston in driving a fluid before it, or by a fluid ing a piston before it, may be expressed in either of the following

Resistance × distance traversed
= intensity of pressure × area × distance traversed ;
= intensity of pressure × volume traversed.

work performed in lifting a body is the product of the weight of the nto the height through which its centre of gravity is lifted.
nachine lifts the centres of gravity of several bodies at once to heights the same or different, the whole quantity of work performed in so s the sum of the several products of the weights and heights; but uantity can also be computed by multiplying the sum of all the s into the height through which their common centre of gravity is (Rankine.)

rer is the rate at which work is done, and is expressed by the quo- the work divided by the time in which it is done, or by units of work ond, per minute, etc., as foot-pounds per second. The most common power is the *horse-power*, established by James Watt as the power of London draught-horse to do work during a short interval, and used to measure the power of his steam-engines. This unit is 33,000 foot- per minute = 550 foot-pounds per second = 1,980,000 foot-pounds per

Expressions for Force, Work, Power, etc.

undamental conceptions in Dynamics are:

ce, Time, Space, represented by the letters F, T, S.

ocity = space divided by time, $V = \frac{S}{T}$, if V be uniform.

k = product of force into space $= FS = W = FVT$. (V uniform.)

rer = rate of work = work divided by time $= \frac{FS}{T} = P$ = product of nto velocity $= FV$.

r exerted for a certain time produces work; $PT = FS = FVT = W$.

rt is a name applied to a force which acts on a body in the direction otion.

istance is that which is opposed to a moving force. It is equal and e force.

se-power Hours, an expression for work measured as the t of a power into the time during which it acts $= PT$. Sometimes it ummation of a variable power for a given time, or the average power lied by the time.

rgy, or stored work, is the capacity for performing work. It is ed by the same unit as work, that is, in foot-pounds. It may be *potential*, as in the case of a body of water stored in a reservoir, e of doing work by means of a water-wheel, or *actual*, sometimes *kinetic*, which is the energy of a moving body. Potential energy is ed by the product of the weight of the stored body into the distance h which it is capable of acting, or by the product of the pressure it into the distance through which that pressure is capable of acting. ial energy may also exist as stored heat, or as stored chemical energy, uel, gunpowder, etc., or as electrical energy, the measure of these s being the amount of work that they are capable of performing. energy of a moving body is the work which it is capable of performing a retarding resistance before being brought to rest, and is equal to rk which must be done upon it to bring it from a state of rest to its velocity.

measure of actual energy is the product of the weight of the body e height from which it must fall to acquire its actual velocity. If v = ocity in feet per second, according to the principle of falling bodies, eight due to the velocity $= \frac{v^2}{2g}$, and if w = the weight, the energy = wh. As the quantity $\frac{w}{g}$ is called the mass $= m$, energy is equal to half ss into the square of the velocity $= \frac{1}{2}mv^2$. Since energy is the capacity forming work, the units of work and energy are equivalent, or $FS = \frac{wv^2}{2g} = wh$. Energy exerted = work done.

The actual energy of a rotating body whose angular velocity is moment of inertia $\Sigma wr^2 = I$ is $\frac{A^2I}{2g}$, that is, the product of the mo inertia into the height due to the velocity, A, of a point whose dista the axis of rotation is unity; or it is equal to $\frac{wv^2}{2g}$, in which w is the the body and v is the velocity of the centre of gyration.

Work of Acceleration.—The work done in giving accelera body is equal to the product of the force producing the accelerati the resistance to acceleration, into the distance moved in a given tim force, as already stated equals the product of the mass into the acc or $f = ma = \frac{w}{g}\frac{v_2 - v_1}{t}$. If the distance traversed in the time t work $= fs = \frac{w}{g}\frac{v_2 - v_1}{t}s$.

EXAMPLE.—What work is required to move a body weighing 100 zontally a distance of 80 ft. in 4 seconds, the velocity uniformly in friction neglected?

Mean velocity $v_0 = 20$ ft. per second; final velocity $= v_2 = 2v_0 = 4$ velocity $v_1 = 0$; acceleration, $a = \frac{v_2 - v_1}{t} = \frac{40}{4} = 10$; force $= \frac{w}{g}a =$ $10 = 31.1$ lbs.; distance 80 ft.; work $= fs = 31.1 \times 80 = 2488$ foot-pou

The energy stored in the body moving at the final velocity of second is

$$\tfrac{1}{2}mv^2 = \frac{1}{2}\frac{w}{g}v^2 = \frac{100 \times 40^2}{2 \times 32.16} = 2488 \text{ foot-pounds},$$

which equals the work of acceleration,

$$fs = \frac{w}{g}\frac{v_2}{t}s = \frac{w}{g}\frac{v_2}{t}\frac{v_2}{2}t = \frac{1}{2}\frac{w}{g}v_2^2.$$

If a body of the weight W falls from a height H, the work of acc is simply WH, or the same as the work required to raise the bo same height.

Work of Accelerated Rotation.—Let A = angular vel solid body rotating about an axis, that is, the velocity of a parti radius is unity. Then the velocity of a particle whose radius is r i If the angular velocity is accelerated from A_1 to A_2, the increa velocity of the particle is $v_2 - v_1 = r(A_1 - A_2)$, and the work of acc it is

$$\frac{w}{g} \times \frac{v_2^2 - v_1^2}{2} = \frac{wr^2}{g}\frac{A_2^2 - A_1^2}{2},$$

in which w is the weight of the particle.

The work of acceleration of the whole body is

$$\Sigma\left\{\frac{w}{g} \times \frac{v_2^2 - v_1^2}{2}\right\} = \frac{A_2^2 - A_1^2}{2g} \times \Sigma wr^2.$$

The term Σwr^2 is the moment of inertia of the body.

"Force of the Blow" of a Steam Hammer or Oth ing Weight.—The question is often asked: "With what for falling hammer strike?" The question cannot be answered dir it is based upon a misconception or ignorance of fundamental m laws. The energy, or capacity of doing work, of a body raised t height and let fall cannot be expressed in pounds, simply, but on pounds, which is the product of the weight into the height throu it falls, or the product of its weight ÷ 64.32 into the square of the in feet per second, which it acquires after falling through the giv If F = weight of the body, M its mass, g the acceleration due t S the height of fall, and v the velocity at the end of the fall, the the body just before striking, is $FS = \frac{1}{2}Mv^2 = Wv^2 \div 2g = W$ which is the general equation of energy of a moving body. Ju energy of the body [illegible] uct of a force into a distance, so th does when it stri[illegible] manifestation of a force, which c *pressed simply* [illegible] is the overcoming of a resistance *a certain distan* [illegible] essed as the product of the avera the distance through which it is exerted. If a hammer weighing ills 10 ft., its energy is 1000 foot-pounds. Before being brought to ust do 1000 foot-pounds of work against one or more resistances. of various kinds, such as that due to motion imparted to the body enetration against friction, or against resistance to shearing or ormation, and crushing and heating of both the falling body and the ck. The distance through which these resisting forces act is gen- leterminate, and therefore the average of the resisting forces, emselves generally vary with the distance, is also indeterminate.

t of Bodies.—If two inelastic bodies collide, they will move on s one mass, with a common velocity. The momentum of the com- s is equal to the sum of the momenta of the two bodies before im- m_1 and m_2 are the masses of the two bodies and v_1 and v_2 their re- elocities before impact, and v their common velocity after impact, $v = m_1v_1 \times m_2v_2$,

$$v = \frac{m_1v_1 + m_2v_2}{m_1 + m_2}.*$$

dies move in opposite directions $v = \frac{m_1v_1 - m_2v_2}{m_1 + m_2}$, or, the velocity lastic bodies after impact is equal to the algebraic sum of their before impact, divided by the sum of their masses.

nelastic bodies of equal momenta impinge directly upon one an- opposite directions they will be brought to rest.

t of Inelastic Bodies Causes a Loss of Energy, and equal to the sum of the energies due to the velocities lost and the bodies, respectively.

$$+ \tfrac{1}{2}m_2v_2^2 - \tfrac{1}{2}(m_1 + m_2)v^2 = \tfrac{1}{2}m_1(v_1 - v)^2 + \tfrac{1}{2}m_2(v_2 - v)^2.$$

$_1 - v$ is the velocity lost by m_1 and $v - v_2$ the velocity gained by m_2.

—Let $m_1 = 10$, $m_2 = 8$, $v_1 = 12$, $v_2 = 15$.

dies collide they will come to rest, for $v = \frac{10 \times 12 - 8 \times 15}{10 + 8} = 0$.

gy loss is

$$+ \tfrac{1}{2}8 \times 225 - \tfrac{1}{2}18 \times 0 = \tfrac{1}{2}10(12 - 0)^2 + \tfrac{1}{2}8(15 - 0)^2 = 1620 \text{ ft. lbs.}$$

comes of the energy lost? Ans. It is used doing internal work ies themselves, changing their shape and heating them.

erfectly elastic bodies, let e = the elasticity, that is, the ratio force of restitution, or the internal force tending to restore the body after it has been compressed, bears to the force of compres- et m_1 and m_2 be the masses, v_1 and v_2 their velocities before im- $_1'v_2'$ their velocities after impact: then

$$v_1' = \frac{m_1v_1 + m_2v_2}{m_1 + m_2} - \frac{m_2e(v_1 - v_2)}{m_1 + m_2};$$

$$v_2' = \frac{m_1v_1 + m_2v_2}{m_1 + m_2} + \frac{m_1e(v_1 - v_2)}{m_1 + m_2}.$$

dies are perfectly elastic, their relative velocities before and after the same. That is: $v_1' - v_2' = v_2 - v_1$.

npact of bodies, the sum of their momenta after impact is the e sum of their momenta before impact.

$$m_1v_1' + m_2v_2' = m_1v_1 + m_2v_2.$$

onstration of these and other laws of impact, see Smith's Me- so, Weisbach's Mechanics.

of Recoil of Guns.—(*Eng'g*, Jan. 25, 1884, p. 72.)

the weight of the gun and carriage;
the maximum velocity of recoil;
the weight of the projectile;
the muzzle velocity of the projectile.

ce the momentum of the gun and carriage is equ mer projectile, we have $WV = wv$, or $V = wv \div W$

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tement by Prof. W. D. Marks, in Nystrom's M that this *formula is in error is itself* erroneou

Taking the case of a 10-inch gun firing a 400-lb. projectile with a muzzl velocity of 1400 feet per second, the weight of the gun and carriage being 2 tons = 49,280 lbs., we find the velocity of recoil =

$$V = \frac{1400 \times 400}{49,280} = 11 \text{ feet per second.}$$

Now the energy of a body in motion is $WV^2 \div 2g$.

Therefore the energy of recoil $= \dfrac{49,280 \times 11^2}{2 \times 32.2} = 92,593$ foot-pounds.

The energy of the projectile is $\dfrac{400 \times 1400^2}{2 \times 32.2} = 12,173,913$ foot-pounds.

Conservation of Energy.—No form of energy can ever be pr duced except by the expenditure of some other form, nor annihilated ex cept by being reproduced in another form. Consequently the sum total of energy in the universe, like the sum total of matter, must always remai the same. (S. Newcomb.) Energy can never be destroyed or lost; it ca be transformed, can be transferred from one body to another, but n matter what transformations are undergone, when the total effects of th exertion of a given amount of energy are summed up the result will b exactly equal to the amount originally expended from the source. This la is called the Conservation of Energy. (Cotterill and Slade.)

A heavy body sustained at an elevated position has potential energ When it falls, just before it reaches the earth's surface it has actual kinetic energy, due to its velocity. When it strikes it may penetrate th earth a certain distance or may be crushed. In either case friction resul by which the energy is converted into heat, which is gradually radiate into the earth or into the atmosphere, or both. Mechanical energy and he are mutually convertible. Electric energy is also convertible into heat mechanical energy, and either kind of energy may be converted into th other.

Sources of Energy.—The principal sources of energy on the earth surface are the muscular energy of men and animals, the energy of th wind, of flowing water, and of fuel. These sources derive their energ from the rays of the sun. Under the influence of the sun's rays vegetatio grows and wood is formed. The wood may be used as fuel under a stea boiler, its carbon being burned to carbonic acid. Three tenths of its he energy escapes in the chimney and by radiation, and seven tenths appea as potential energy in the steam. In the steam-engine, of this seven tent six parts are dissipated in heating the condensing water and are waste the remaining one tenth of the original heat energy of the wood is co verted into mechanical work in the steam-engine, which may be used drive machinery. This work is finally, by friction of various kinds, or p sibly after transformation into electric currents, transformed into he which is radiated into the atmosphere, increasing its temperature. Th all the potential heat energy of the wood is, after various transformatio converted into heat, which, mingling with the store of heat in the atmo phere, apparently is lost. But the carbonic acid generated by the combu tion of the wood is, again, under the influence of the sun's rays, absorb by vegetation, and more wood may thus be formed having potential ener equal to the original.

Perpetual Motion.—The law of the conservation of energy, th which no law of mechanics is more firmly established, is an absolute barr to all schemes for obtaining by mechanical means what is called "perpet motion," or a machine which will do an amount of work greater than t equivalent of the energy, whether of heat, of chemical combination, of el tricity, or mechanical energy, that is put into it. Such a result would the creation of an additional store of energy in the universe, which is i possible by any human agency.

The Efficiency of a Machine is a fraction expressing the ratio the useful work to the whole work performed, which is equal to the ener expended. The limit to the efficiency of a machine is unity, denoting t efficiency of a perfect machine in which no work is lost. The differe between the energy expended and the useful work done, or the loss usually expended either in overcoming friction or in doing work on bod surrounding the machine from which no useful work is received. Thus ... propel... vessel part of the energy exerted in the cylin ıl work of giving motion to the vessel, and the remainder is oming the friction of the machinery and in making currents the surrounding water.

ANIMAL POWER.

a Man against Known Resistances. (Rankine.)

f Exertion.	R, lbs.	V, ft. per sec.	$\frac{T''}{3600}$ (hours per day).	RV, ft.-lbs. per sec.	RVT, ft.-lbs. per day.
s own weight up adder	143	0.5	8	72.5	2,088,000
weights with rope. ering the rope un-	40	0.75	6	30	648,000
ghts by hand......	44	0.55	6	24.2	522,720
weights up-stairs rning unloaded....	143	0.13	6	18.5	399,600
up earth to a 5 ft. 3 in	6	1.3	10	7.8	280,800
arth in barrow up 1 in 12, ½ horiz. ft. per sec. and re- nloaded	132	0.075	10	9.9	356,400
pulling horizon- stan or oar).......	26.5	2.0	8	53	1,526,400
rank or winch . .	12.5	5.0	?	62.5	....
	18.0	2.5	8	45	1,296,000
	20.0	14.4	2 min.	288	....
ımp..	13.2	2.5	10	33	1,188,000
g.................	15	?	8?	?	480,000

s.—R, resistance; V, effective velocity = distance through ercome ÷ total time occupied, including the time of moving ay; T'', time of working, in seconds per day; $T'' \div 3600$, same s per day; RV, effective power, in foot-pounds per second; ork.

rmance of a Man in Transporting Loads Horizontally. (Rankine.)

f Exertion.	L, lbs.	V, ft.-sec.	$\frac{T}{3600}$ (hours per day).	LV, lbs. con- veyed 1 foot.	LVT, lbs. con- veyed 1 foot.
nloaded, transport- wn weight.........	140	5	10	700	25,200,000
load L in 2-whld. return unloaded..	224	1⅔	10	373	13,428,000
wh. barrow, ditto..	132	1⅔	10	220	7,920,000
with burden.....	90	2½	7	225	5,670,000
burden, returning l	140	1⅔	6	233	5,032,800
urden, for 30 sec- y	252	0	....	0	....
	126	11.7	....	1474.2	....
	0	23.1	....	[illegible]	....

s.—L, load; V, *effective velocity, com* *g, in seconds per day; $T'' \div 3600$, same t* *per second, in lbs. conveyed one foot; L*

In the first line only of each of the two tables above is the w
man taken into account in computing the work done.

Clark says that the average net daily work of an ordinary
pump, a winch, or a c
taken at 3300 foot-pound
or one-tenth of a horse
hours a day; but for sh
from four to five times
be exerted.

Mr. Glynn says that
exert a force of 25 lbs.
of a crane for short per
for continuous work a f
is all that should be assu
through 220 feet per mi

Man-wheel.—Fig.
of a very efficient man
ing-machine which the
Berne, Switzerland, in 1
of the wheel was wide
three men to walk abr
nine men could work in

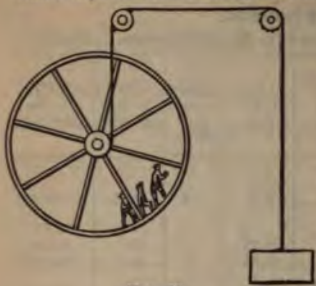


FIG. 97.

Work of a Horse against a Known Resistance

Kind of Exertion.	R.	V.	$\frac{T.}{3600}$	RV
1. Cantering and trotting, drawing a light railway carriage (thoroughbred)......... ..	min. 22½ mean 30½ max. 50	14⅗	4	447
2. Horse drawing cart or boat, walking (draught-horse)....	120	3.6	8	432
3. Horse drawing a gin or mill, walking	100	3.0	8	300
4. Ditto, trotting	66	6.5	4½	429

EXPLANATION.—R, resistance, in lbs.; V, velocity, in feet pe
÷ 3600, hours work per day; RV, work per second; RVT, work

The average power of a draught-horse, as given in line 2 of th
being 432 foot-pounds per second, is $432/550 = 0.785$ of the conve
assigned by Watt to the ordinary unit of the rate of work of p
It is the mean of several results of experiments, and may be c
average of ordinary performance under favorable circumstan

Performance of a Horse in Transporting Horizontally. (Rankine.)

Kind of Exertion.	L.	V.	T.	LV
5. Walking with cart, always loaded....	1500	3.6	10	5400
6. Trotting, ditto....	750	7.2	4½	5400
7. Walking with cart, going loaded, returning empty; V, mean velocity..............	1500	2.0	10	3000
8. Carrying burden, walking...	270	3.6	10	972
9. Ditto, trotting	180	7.2	7	1296

EXPLANATION.—L, load in lbs.; V, velocity in feet per seco
working hours per day; LV, transport per second; LVT, trans

This table has reference to conveyance on common roads on
[illegible]tly in bad order as respects the resistance to traction up

[illegible]e **Gin.**—In this machine a horse works less ad
[illegible]rawing [illegible] ng a straight track. In order

e results may be realized with a horse-gin, the diameter of the cir-rack in which the horse walks should not be less than about forty

n, Mules, Asses.—Authorities differ considerably as to the power e animals. The following may be taken as an approximative com-between them and draught-horses (Rankine):

Load, the same as that of average draught-horse; best velocity and wo thirds of horse.

—Load, one half of that of average draught-horse; best velocity, e with horse; work one half.

Load, one quarter that of average draught-horse; best velocity the work one quarter.

uction of Draught of Horses by Increase of Grade ads. (*Engineering Record*, Prize Essays on Roads, 1892.)—Experi-n English roads by Gayffier & Parnell:

g load that can be drawn on a level 100:

e of.	1 in 100.	1 in 50.	1 in 40.	1 in 30.	1 in 26.	1 in 20.	1 in 10.
can draw only	90.	81.	72.	64.	54.	40.	25.

Resistance of Carriages on Roads is (according to Gen. iven approximately by the following empirical formula:

$$R = \frac{W}{r}[a + b(u - 3.28)].$$

formula R = total resistance; r = radius of wheel in inches; W = d; u = velocity in feet per second; while a and b are constants. lues are: For good broken-stone road, a = .4 to .55, b = .024 to .026; l roads, a = .27, b = .0684.

e states that on gravel the resistance is about double, and on times, the resistance on good broken-stone roads.

ELEMENTS OF MACHINES.

ect of a machine is usually to transform the work or mechanical erted at the point where the machine receives its motion into the point where the final resistance me. The specific end may be to he character or direction of mo-rom circular to rectilinear, or *vice* change the velocity, or to overcome esistance by the application of a force. In all cases the total energy uals the total work done, the latter the overcoming of all the frictional s of the machine as well as the use-performed. No increase of power ained from any machine, since this ble according to the law of conser-nergy. In a frictionless machine the f the force exerted at the driving-the velocity of the driving-point, tance it moves in a given interval quals the product of the resistance listance through which the resist-ercome in the same time.

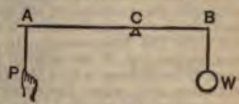


FIG. 98.

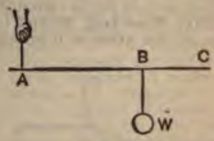


FIG. 99.

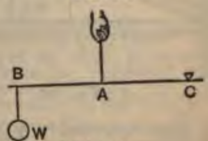


FIG. 100.

st simple machines, or elementary are reducible to three classes, viz., , the Cord, and the Inclined Plane.

class includes every machine con-a solid body capable of revolving , as the Wheel and Axle.

nd class includes every machine in e is transmitted by means of flexi-s, ropes, etc., as the Pulley.

d class includes every machine in ard surface inclined to the direc-tion is introduced, as the Wedge and the Screw.

er is an inflexible rod capable of motion about a fixed point, lcrum. The rod *may be straight or* bent at any angle, o erally *regarded, at first, as without* weight, but its weig considered as another force applied in a vertical direction at i gravity.

The arms of a lever are the portions of it intercepted betwee P, and fulcrum, C, and between the weight, W, and fulcrum.

Levers are divided into three kinds or orders, according to positions of the applied force, weight, and fulcrum.

In a lever of the first order, the fulcrum lies between the poin the force and weight act. (Fig. 98.)

In a lever of the second order, the weight acts at a point b fulcrum and the point of action of the force. (Fig. 99.)

In a lever of the third order, the point of action of the force that of the weight and the fulcrum. (Fig. 100.)

In all cases of levers the relation between the force exerted P, and the weight lifted, or resistance overcome, W, is expre equation $P \times AC = W \times BC$, in which AC is the lever-arm of is the lever-arm of W, or moment of the force = the moment o ance. (See Moment.)

In cases in which the direction of the force (or of the resistanc right angles to the arm of the lever on which it acts, the "lever- length of a perpendicular from the fulcrum to the line of dire force (or of the resistance). $W : P :: AC : BC$, or, the ratio of the r the applied force is the inverse ratio of their lever-arms. Also, i velocity of W, and V_p is the velocity of P, $W : P :: V_p : V_w$, $= W \times V_w$.

If S_p is the distance through which the applied force acts, an distance the weight is lifted or through which the resistance i $W : P :: S_p : S_w$; $W \times S_w = P \times S_p$, or the weight into the distanc equals the force into the distance through which it is exerted.

These equations are general for all classes of machines as levers, it being understood that friction, which in actual machin the resistance, is not at present considered.

The Bent Lever.—In the bent lever (see Fig. 91, page 416 arm of the weight m is cf instead of bf. The lever is in equilib $n \times af = m \times cf$, but it is to be observed that the action of a bent be very different from that of a straight lever. In the latter, so force and the resistance act in lines parallel to each other, the r lever-arms remains constant, although the lever itself changes tion with the horizontal. In the bent lever, however, this rat thus, in the cut, if the arm bf is depressed to a horizontal directi tance cf lengthens while the horizontal projection of af shortens becoming zero when the direction of af becomes vertical. As t approaches the vertical, the weight m which may be lifted w force s is very great, but the distance through which it may very small. In all cases the ratio of the weight m to the weight verse ratio of the horizontal projection of their respective lever-a

The Moving Strut (Fig. 101) is similar to the bent lever, one of the arms is missing, and that the force and the resista overcome act at the same single arm. The resista case shown in the cut weight W, but its res being moved, R, which n ply that due to its frict horizontal plane, or some posing force. When the tween the strut and the plane changes, the rat resistance to the app changes. When the angl very small, a moderate overcome a very great which tends to become

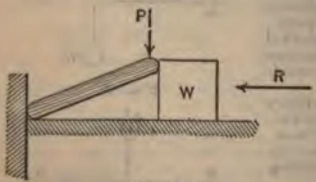


FIG. 101.

the angle approaches zero. If a = the angle, $P \times \cos a = R$ a = 5 degrees, $\cos a$ = .99619, $\sin a$ = .08716, R = 11.44 P.

The stone-crusher (Fig. 102) shows a practical example of the moving struts.

The Toggle-joint is an elbow or knee-joint consisting of connected tha[illegible] be brought into a straight line and m duce great [illegible] sure when a force is applied to bring the

1. It is a case of two moving struts placed end to end, the moving eing applied at their point of junction, in a direction at right angles lirection of the resistance, the other end of one of the struts resting a fixed abutment, and that of the other against the body to be If α = the angle each strut makes with the straight line joining the about which their outer ends rotate, the ratio of the resistance applied force is $R : P :: \cos \alpha : 2 \sin \alpha$; $2R \sin \alpha = P \cos \alpha$. The

FIG. 102.

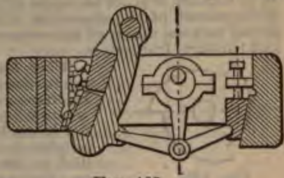


FIG. 103.

aries when the angle varies, becoming infinite when the angle s zero.

toggle-joint is used where great resistances are to be overcome 1 very small distances, as in stone-crushers (Fig. 103).

Inclined Plane, as a mechanical element, is supposed perfectly id smooth, unless friction be considered. It assists in sustaining a body by its reaction. This reaction, however, being normal to the cannot entirely counteract the weight of the body, which acts vertiownward. Some other force must therefore le to act upon the body, in order that it may ained.

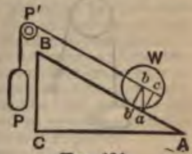


FIG. 104.

e sustaining force act parallel to the plane 4), the force is to the weight as the height of ne is to its length, measured on the incline. force act parallel to the base of the plane, ver is to the weight as the height is to the

e force act at any other angle, let i = the f the plane with the horizon, and e = the f the direction of the applied force with the f the plane. $P : W :: \sin i : \cos e$; $P \times \cos e = W \sin i$.

lems of the inclined plane may be solved by the parallelogram of thus:

he weight W be kept at rest on the incline by the force P, acting in e bP', parallel to the plane. Draw the vertical line ba to represent ight: also bb' perpendicular to the plane, and complete the parallelo-$'c$. Then the vertical weight ba is the resultant of bb', the measure of t given by the plane to the weight, and bc, the force of gravity tenddraw the weight down the plane. The force required to maintain ight in equilibrium is represented by this force bc. Thus the force e weight are in the ratio of bc to ba. Since the triangle of forces abc lar to the triangle of the incline ABC, the latter may be substituted former in determining the relative magnitude of the forces, and

$$P : W :: bc : ab :: BC : AB.$$

Wedge is a pair of inclined planes united by their bases. In the ation of pressure to the head or butt end of the wedge, to cause it to ate a resisting body, the applied force is to the resistance as the ess of the wedge is to its length. Let t be the thickness, l the length, resistance, and P the applied force or pressure on the head of the . Then, friction neglected, $P : W :: t : l$; $P = \frac{Wt}{l}$; $W = \frac{Pl}{t}$.

Screw is an inclined plane wrapped around a cylinder in such a at the height of the plane is parallel to the axis of the cylinder. If ew is formed upon the internal surface of a hollow cylinder, it is called a nut. When *force is applied* to raise a weight [illegible] ance by *means of a screw and nut*, either the screw [illegible] may be fixed, the other being movable. The force is generally applied at the of a wrench or lever-arm, or at the circumference of a wheel. If r = ra of the wheel or lever-arm, and p = pitch of the screw, or distance betw threads, that is, the height of the inclined plane for one revolution of the screw, P = the applied force, and W = the resistance overcome, then, neglecting resistance due to friction, $2\pi r \times P = Wp$; $W = 6.283Pr \div p$. The ratio of P to W is thus independent of the diameter of the screw. In actual screws, much of the power transmitted is lost through friction.

Fig. 106.

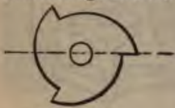


Fig. 105.

The Cam is a revolving inclined plane. It may be either an inclined plane wrapped around a cylinder in such a way that the height of the plane is radial to the cylinder, such as the ordinary lifting-cam, used in stamp-mills (Fig. 105), or it may be an inclined plane curved edgewise, and rotating plane parallel to its base (Fig. 106). The relation of the weight to the ap force is calculated in the same manner as in the case of the screw.

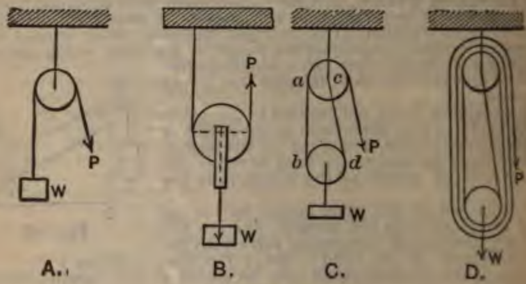


Fig. 107.

Pulleys or Blocks.—P = force applied, or pull; W = weight or resistance. In the simple pulley A (Fig. 107) the point P on the rope descends the same amount that the weight is lifted, therefore P In B and C the point P moves twice as far as the weight is lifted, t fore $W = 2P$. In B and C there is one movable block, and two plies rope engage with it. In D there are three sheaves in the movable each with two plies engaged, or six in all. Six plies of the rope are fore shortened by the same amount that the weight is lifted, and the P moves six times as far as the weight, consequently $W = 6P$. In ge the ratio of W to P is equal to the number of plies of the rope th shortened, and also is equal to the number of plies that engage the block. If the lower block has 2 sheaves and the upper 3, the end of th is fastened to a hook in the top of the lower block, and then there plies shortened instead of 6, and $W = 5P$. If V = velocity of W, an velocity of P, then in all cases $VW = vP$, whatever the number of sh or their arrangement. If the hauling rope, at the pulling end, passe around a sheave in the upper or stationary block, it makes no differe what direction the rope is led from this block to the point at which th on the rope is applied; but if it first passes around the movable block necessary that the pull be exerted in a direction parallel to the line of the resistance, or a line joining the centres of the two blocks, in o tain the maximum effect. If the rope pulls on the lower block gle, the block will be pulled out of the line drawn between the f the upper block, and the effective pull will be less than the actu in the ratio of the cosine of the angle the pulling rope makes rtical, or line of action of the resistance, to unity.

ntial Pulley. (Fig. 108.)—Two pulleys, B and C, of different as one piece about a fixed axis, A. An end- *BDECLKH*, passes over both pulleys. The pulleys are shaped so as to hold the chain and from slipping. One of the bights or loops in hain hangs, DE, passes under and supports the ock F. The other loop or bight, HKL, hangs is called the hauling part. It is evident that of the hauling part is equal to that of the of the pulley B.

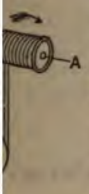
hat the velocity-ratio may be exactly uniform, of the sheave F should be an exact mean be- adii of B and C.

that the point B of the cord BD moves through se length $= AB$, during the same time the the cord CE will move downward a distance $=$ length of the bight or loop $BDEC$ will be y $AB - AC$, which will cause the pulley F to lf of this amount. If $P =$ the pulling force on K, and W the weight lifted at F, then $P \times$ $\frac{1}{2}(AB - AC)$.

tethe length of chain required for a differential the following sum: Half the circumference of e circumference of $B +$ half the circumference ce the greatest distance of F from $A +$ the of loop HKL. The last quantity is fixed convenience.

FIG. 108.

fferential Windlass (Fig. 109) is identical in principle with the differential pulley, the difference in construction being that in the differential windlass the running block hangs in the bight of a rope whose two parts are wound round, and have their ends respectively made fast to two barrels of different radii, which rotate as one piece about the axis A. The differential windlass is little used in practice, because of the great length of rope which it requires.



09.

The Differential Screw (Fig. 110) is a compound screw of different pitches, in which the threads wind the same way. N_1 and N_2 are the two nuts; S_1S_1, the longer-pitched thread; S_2S_2, the shorter-pitched thread: in the figure both these threads are left-handed. At each turn of the screw the nut N_2 advances relatively to N^2 through a distance equal to the difference of the pitch. The use of the differential screw is to combine the slowness lue to a fine pitch with the strength of thread which can be means of a coarse pitch only.

and Axle, or Windlass, resembles two pulleys on one axis, rent diameters. If a weight be lifted by means of a rope wound e, the force being applied at the heel, the action is like that of a ch the shorter arm is equal to the axle plus half the thick- rope, and the longer arm is radius of the wheel. A wheel therefore sometimes classed al lever. If $P =$ the applied force, $D =$ diameter of the wheel, ight lifted, and d the diameter of the axle $+$ the diameter of $= Wd$.

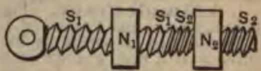


FIG. 110.

-wheel Gearing is a combination of two or more wheels g 111). If a series of wheels and pinions gear into each other, ut, friction neglected, the weight lifted, or resistance over- the force applied inversely as the distances through which given time. If R, R_1, R_2 be the radii of the successive wheels the pitch-line of the teeth, and r, r_1, r_2 the radii of the c pinions, P the applied force, and W the weight lifted, I

$R \times R_1 \times R_2 = W \times r \times r_1 \times r_2$, or the applied force is to the w as the product of the radii of the pinions is to the product of the ra the wheels; or, as the product of the numbers expressing the te each pinion is to the product of the numbers expressing the teeth in wheel.

Endless Screw, or Worm-gear. (Fig. 112.)—This gear is monly used to convert motion at high speed into motion at very

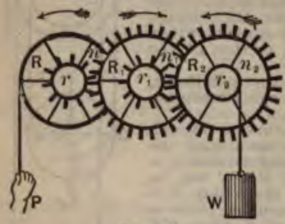


FIG. 111.

FIG. 112.

speed. When the handle P describes a complete circumference, the line of the cog-wheel moves through a distance equal to the pitch screw, and the weight W is lifted a distance equal to the pitch of the multiplied by the ratio of the diameter of the axle to the diameter pitch-circle of the wheel. The ratio of the applied force to the w lifted is inversely as their velocities, friction not being considered; b friction in the worm-gear is usually very great, amounting sometin three or four times the useful work done.

If $v =$ the distance through which the force P acts in a given time, second, and $V =$ distance the weight W is lifted in the same tim radius of the crank or wheel through which P acts, $t =$ pitch of the and also of the teeth on the cog-wheel, $d =$ diameter of the and $D =$ diameter of the pitch-line of the cog-wheel, $v = \frac{6.28}{}$ $\times V$; $V = v \times td \div 6.283rd$. $Pv = WV +$ friction.

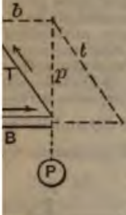
STRESSES IN FRAMED STRUCTURES.

Framed structures in general consist of one or more triangles, fo reason that the triangle is the one polygonal form whose shape can changed without distorting one of its sides. Problems in stresses of s framed structures may generally be solved either by the application triangle, paralellogram, or polygon of forces, by the principle of the or by the method of moments. We shall give a few examples, referri student to the works of Burr, Dubois, Johnson, and others for more rate treatment of the subject.

1. **A Simple Crane.** (Figs. 113 and 114.)—A is a fixed mast, B a br boom, T a tie, and P the load. Required the strains in B and T. The w P, considered as acting at the end of the boom, is held in equilibriu three forces: first, gravity acting downwards; second, the tension in T third, the thrust of B. Let the length of the line p represent the mag of the downward force exerted by the load, and draw a parallelogram sides bt parallel, respectively, to B and T, such that p is the diagonal parallelogram. Then b and t are the components drawn to the same as p, p being the resultant. Then if the length p represents the loa the tension in the tie, and b is the compression in the brace.

Or, more simply, T, B, and that portion of the mast included between or A' may represent a triangle of forces, and the forces are proportio *the length of* the sides of the triangle; that is, if the height of the trian = *the load*, then $B =$ the compression in the brace, and $T =$ the tension tie; or if $P =$ the load in pounds, the tension in $T = P \times \frac{T}{A'}$, and th

$B = P \times \frac{B}{A'}$. Also, if a = the angle the inclined member makes
ast, the other member being horizontal, and the triangle being
d, then the length of the inclined member = height of the tri-
cant a, and the strain in the inclined member = P secant a. Also,
n the horizontal member = P tan a.
tion by the triangle or parallelogram of forces, and the equations
$T = P \times T/A'$, and Compression in $B = P \times B/A'$, hold true even
gle is not right-angled, as in Fig. 115; but the trigonometrical rela-



. 113.

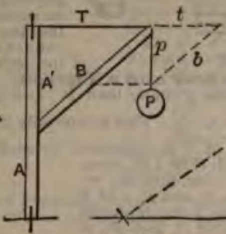


FIG. 114.

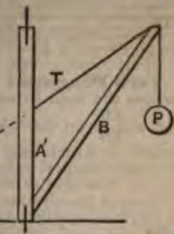


FIG. 115. See pag

given do not hold, except in the case of a right-angled triangle.
t that as A' decreases, the strain in both T and B increases, tend-
me infinite as A' approaches zero. If the tie T is not attached to
ut is extended to the ground, as shown in the dotted line, the
t remains the same.

yed Crane or Derrick. (Fig. 116.)—The strain in B is, as
B/A', A' being that portion of the vertical included between B and
T may be attached to A. If, however, the tie T is attached to B
extremity, there may be in addition a bending strain in B due to
to turn about the point of attachment of T as a fulcrum.
n in T may be calculated by the principle of moments. The mo-
is Pc, that is, its weight $\times$ its perpendicular distance from the
ation of B on the mast. The moment of the strain on T is the
the strain into the perpendicular distance from the line of its

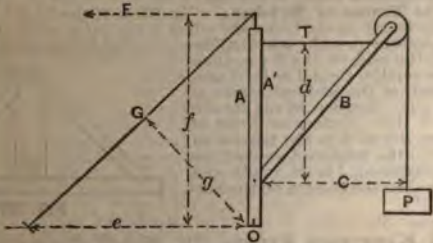


FIG. 116.

the same point of rotation of B, or Td. The strain in T there-
d. As d decreases the strain on T increases, tending to infin-
roaches zero.
on the *guy-rope is also calculated* by the method of moments.
t of the load about the bottom of the mast O is, as before, Pc.
horizontal the strain in it is F and its moment is Ff, and F =
is inclined, the moment is the strain $G \times$ the perpendicular
e line of its direction from O, or Gg, and $G = Pc \div g$.
having the least strain is the horizontal one F, and the strain

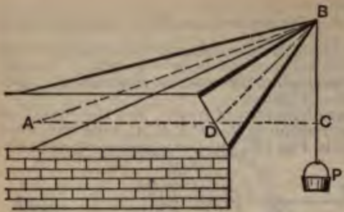


FIG. 117.

in G = the strain in $F \times$ cant of the angle betwee G. As G is made more vertical g decreases, a strain increases, becomi nite when $g = 0$.

3. **Shear-poles** **Guys.** (Fig. 117.)—Resu strain in both masts = $\div BC$. Resultant strain guys $= P \times AB \div BC$. Th on each mast (or guy) wil the above, multiplied by cant of half the angle th (or guys) make with eac

Two Diagonal Braces and a Tie-rod. (Fig. 118.)—Supp braces are used to sustain a single load P. Compressive stress o $\frac{1}{2}P \times \frac{AD}{AB}$; on $CA = \frac{1}{2}P \times \frac{CA}{AB}$. This is true only if CB and BD are length, in which case $\frac{1}{2}$ of P is supported by each abutment C an they are unequal in length (Fig. 119), then, by the principle of the lever, find the reactions of the abutments R_1 and R_2. If P is the load applied at the point B on the lever CD, the fulcrum being D, then $R_1 \times CD = P \times BD$ and $R_2 \times CD = P \times BC$; $R_1 = P \times BD \div CD$; $R_2 = P \times BC \div CD$.

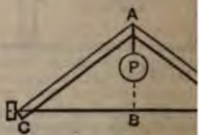


FIG. 118.

The strain on $AC = R_1 \times AC \div AB$, and on $AD = R_2 \times AD \div AB$.

The strain on the tie $= R_1 \times CB \div AB = R_2 \times BD \div AB$.

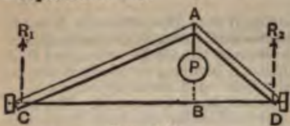


FIG. 119.

When $CB = BD$, $R_1 = R_2$. T on CB and BD is the same, the braces are of equal le not, and is equal to $\frac{1}{2}P \times \frac{1}{2}C$

If the braces support a unifo as a pair of rafters, the strain by such a load are equivalent caused by one half of the load at the centre. The horizonta of the braces against each oth apex equals the tensile strain in the tie.

King-post Truss or Bridge. (Fig. 120.)—If the load is dis over the whole length of the truss, the effect is the same as if half were placed at the centre, the other half being carried by the abutme P = one half the load on the truss, then tension in the vertical tie $AB = P$. Compression in each of the inclined braces $= \frac{1}{2}P \times AD \div AB$. Tension in the tie $CD = \frac{1}{2}P \times BD \div AB$. Horizontal thrust of inclined brace AD at D = the tension in the tie. If W = the total load on one truss uniformly distributed, l = its length and d = its depth, then the tension on the horizontal tie $= \frac{Wl}{8d}$.

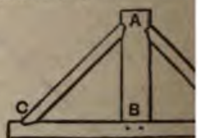


FIG. 120.

Inverted King-post Truss. (Fig. 121.)—If P = a load a B, or one half of a uniformly distributed load, then compression on (the floor-beam CD not being co to have any resistance to a slight b Tension on AC or $AD = \frac{1}{2}P \times A$ Compression on $CD = \frac{1}{2}P \times BD$

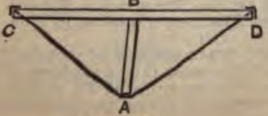


FIG. 121.

Queen-post Truss. (Fig. uniformly loaded, and the queen vide the length into three equal load may be considered to be three equal parts, two parts and P_2, are concentrated at th mainder is equally divided between the abutments and supported irectly. The two parts P_1 and P_2 only are considered to affect

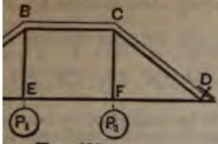


FIG. 122.

the members of the truss. Strain in the vertical ties BE and CF each equals P_1 or P_2. Strain on AB and CD each $= P_1 \times CD \div CF$. Strain on the tie AE or EF or $ED = P_1 \times FD \div CF$. Thrust on BC = tension on EF.

For stability to resist heavy unequal loads the queen-post truss should have diagonal braces from B to F and from C to E.

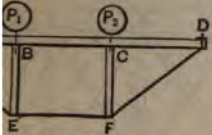


FIG. 123.

Inverted Queen-post Truss. (Fig. 123.)—Compression on EB and FC each $= P_1$ or P_2. Compression on AB or BC or $CD = P_1 \times AB \div FB$. Tension on AE or $FD = P_1 \times AE \div EB$. Tension on EF = compression on BC. For stability to resist unequal loads, ties should be run from C to E and from B to F.

'russ of Five Panels. (Fig. 124.)—Four fifths of the load may s concentrated at the points E, K, L and F, the other fifth being

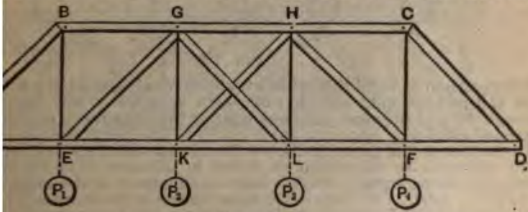


FIG. 124.

directly by the two abutments. For the strains in BA and CD nay be considered as a queen-post truss, with the loads P_1, P_2 ed at E and the loads P_3, P_4 concentrated at F. Then, compreson $AB = (P_1 + P_2) \times AB \div BE$. The strain on CD is the same if nd panel lengths are equal. The tensile strain on BE or CF = 'hat portion of the truss between E and F may be considered as ueen-post truss, supporting the loads P_2, P_3 at K and L. The G or $HF = P_2 \times EG \div GK$. The diagonals GL and KH receive no ss the truss is unequally loaded. The verticals GK and HL each nsile strain equal to P_2 or P_3.

train in the horizontal members: BG and CH receive a thrust e horizontal component of the thrust in AB or CD, $= (P_1 + P_2)$ e ABE, or $(P_1 + P_2) \times AE \div BE$. GH receives this thrust and ition, a thrust equal to the horizontal component of the thrust in or, in all, $(P_1 + P_2 + P_3) \times AE \div BE$.

on in AE or FD equals the thrust in BG or HC, and the tension and LF equals the thrust in GH.

r Whipple Truss. (Fig. 125.)—In this truss the diagonals are e verticals are struts or columns.

on by the method of distribution of strains: Consider first the 'he truss having six bays or panels, 5/6 of the load is transmitted ment H, and 1/6 to the abutment O, on the principle of the lever. sixths must *be transmitted through JA* and AH, write on these *be figure 5. The one sixth is transmitted* successively through *DL, etc., passing alternately* through a tie and a strut. Write *nbers, up to the strut GO* inclusive, the figure 1. Then consider *of which 4/6 goes to AH* and 2/6 to GO. Write on KB, BJ, JA, *igure 4, and on* KD, DL, LE, etc., the figure 2. The load P_2

FIG. 117.

Two Diagonal Braces and
braces are used to sustain a
$\frac{1}{2}P \times \frac{AD}{AB}$; on $CA = \frac{1}{2}P \times \frac{CA}{AB}$.
length, in which case $\frac{1}{2}$ of P is
they are unequal in length (Fig. 11
by the principle of the lever,
actions of the abutments R_1 and
is the load applied at the point
lever CD, the fulcrum being D,
$CD = P \times BD$ and $R_2 \times CD =$
$R_1 = P \times BD \div CD$; $R_2 = P \times$

The strain on $AC = R_1 \times AC$
on $AD = R_2 \times AD \div AB$.

The strain on the tie $= R_1$
$= R_2 \times BD \div AB$.

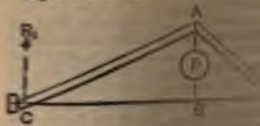


FIG. 119.

apex equals the tensile

King-post Truss
over the whole length
were placed at the
P ... If the

...chords—**Method of Moments.**—Let the truss be ... load acting on it = W. Weight supported at ... of the abutment = $W/2$. Length of the truss = L. ... length = W/L. Horizontal distance from the nearest ... say M in Fig. 125) in the chord where the strain is to ... Horizontal strain at that point (tension on the lower ... the upper) = H. Depth of the truss = D. By the ... we take the difference of the moments, about the point ... of the abutment and of the load between and the abutment ... that difference with the moment of the resistance, or of ... horizontal chord, considered with reference to a point in ... about which the truss would turn if the first chord were ...

... of the reaction of the abutment is $Wx/2$. The moment of ... the abutment to M is $W/Lx \times$ the distance of its centre of ... which is $x/2$, or moment $= Wx^2 \div 2L$. Moment of the stress ... $= \frac{Wx}{2} - \frac{Wx^2}{2L}$, whence $H = \frac{W}{2D}\left(x - \frac{x^2}{L}\right)$. If $x = 0$ or L. ... $H = \frac{WL}{8D}$, which is the horizontal strain at the middle ... before given.

... **Truss.** (Fig. 126.)—In the Howe truss the diagonals are ... verticals are ties. The calculation of strains may be made

FIG. 126.

... method as described above for the Pratt truss.

...rren Girder. (Fig. 127.)—In the Warren girder, or triangular ... are no vertical struts, and the diagonals may transmit either

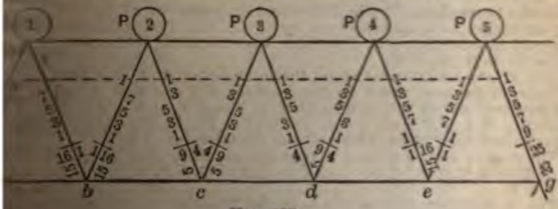


FIG. 127.

... compression. The strains in the diagonals may be calculated by ... of distribution of strains as in the case of the rectangular truss. ... principle of the lever, the load P_1 being 1/10 of the length of the ... the line of the nearest support a, transmits 9/10 of its weight to a ... to g. Write 9 on the right hand of the strut $1a$, to represent the ... and 1 on the right hand of $1b$, $2c$, $3d$, etc., to represent com... on the left hand of $b2$, $c3$, etc., to represent tension. The load P_2 ... 2/10 of its weight to a and 8/10 to g. Write 7 on each member from ... on each member from 2 to g, placing the figures representing ... on the right hand of the member, and those representing ... Proceed in the same manner with all the loads, then transmit 3/6 in each direction; write 3 on each of the members which this stress passes, and so on for all the loads, when the figure several members will appear as on the cut. Adding them up, we following totals:

Tension on diagonals

AJ	BH	BK	CJ	CL	DK	DM	EL	EN	FM
15	0	10	1	6	3	3	6	1	10

Compression on verticals

AH	BJ	CK	DL	EM	FN	G
15	10	7	6	7	10	1

Each of the figures in the first line is to be multiplied by $1/6P \times$ se angle HAJ, or $1/6P \times AJ \div AH$, to obtain the tension, and each figu lower line is to be multiplied by $1/6P$ to obtain the compression. T onals HB and FO receive no strain.

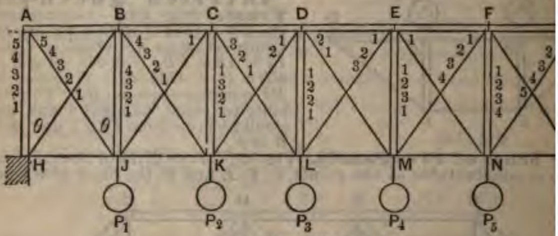


FIG. 125.

It is common to build this truss with a diagonal strut at HB instea post HA and the diagonal AJ; in which case 5/6 of the load P is through JB and the strut BH, which latter then receives a strain = secant of HBJ.

The strains in the upper and lower horizontal members or chords from the ends to the centre, as shown in the case of the Burr tru receives a thrust equal to the horizontal component of the tension i $15/6P \times \tan AJB$. BC receives the same thrust + the horizontal co of the tension in BK, and so on. The tension in the lower chord of ea is the same as the thrust in the upper chord of the same panel. (F lation of the chord strains by the method of moments, see below.)

The maximum thrust or tension is at the centre of the chords and to $\frac{WL}{8D}$, in which W is the total load supported by the truss, L is th and D the depth. This is the formula for maximum stress in th of a truss of any form whatever.

The above calculation is based on the assumption that all the load etc., are equal. If they are unequal the value of each has to be ta account in distributing the strains. Thus the tension in AJ, with loads, instead of being $15 \times 1/6\ P$ secant θ would be sec $\theta \times (5/6 P_1 +$ $3/6\ P_3 + 2/6\ P_4 + 1/6\ P_5$.) Each panel load, P_1 etc., includes its fra the weight of the truss.

General Formula for Strains in Diagonals and Ver — Let n = total number of panels, x = number of any vertical co from the nearest end, counting the end as 1, r = rolling load for eac P = total load for each panel,

$$\text{Strain on verticals} = \frac{[(n-x)+(n-x)^2-(x-1)+(x-1)^2]P}{2n} + \frac{r(x-1)+}{2n}$$

For a uniformly distributed load, leave out the last term,

$$[r(x-1)+(x-1)^2] \div 2n.$$

Strain on principal diagonals = strain on verticals $\times$ secant θ, *secant of the angle the diagonal makes with the vertical.*

Strain on the counterbraces: The strain on the counterbrace in panel is 0, if the load is uniform. On the 2d, 3d, 4th, etc., it is P $\times \frac{1}{n}$, $\frac{1+2}{n}$, $\frac{1+2+3}{n}$, etc., P being the total load in one panel.

train in the Chords—Method of Moments.—Let the truss be formly loaded, the total load acting on it $= W$. Weight supported at h end, or reaction of the abutment $= W/2$. Length of the truss $= L$. ight on a unit of length $= W/L$. Horizontal distance from the nearest tment to the point (say M in Fig. 125) in the chord where the strain is to determined $= x$. Horizontal strain at that point (tension on the lower rd, compression in the upper) $= H$. Depth of the truss $= D$. By the thod of moments we take the difference of the moments, about the point of the reaction of the abutment and of the load between and the abut- nts, and equate that difference with the moment of the resistance, or of strain in the horizontal chord, considered with reference to a point in opposite chord, about which the truss would turn if the first chord were ered at M.

he moment of the reaction of the abutment is $Wx/2$. The moment of load from the abutment to M is $W/Lx \times$ the distance of its centre of vity from M, which is $x/2$, or moment $= Wx^2 \div 2L$. Moment of the stress the chord $= HD = \frac{Wx}{2} - \frac{Wx^2}{2L}$, whence $H = \frac{W}{2D}\left(x - \frac{x^2}{L}\right)$. If $x = 0$ or L, $= 0$. If $x = L/2$, $H = \frac{WL}{8D}$, which is the horizontal strain at the middle the chords, as before given.

he Howe Truss. (Fig. 126.)—In the Howe truss the diagonals are ts, and the verticals are ties. The calculation of strains may be made

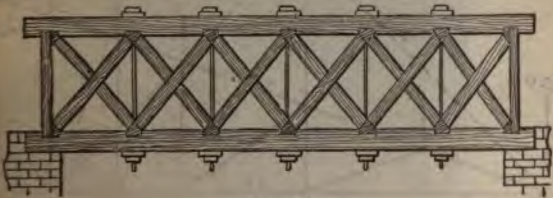


Fig. 126.

he same method as described above for the Pratt truss.

he Warren Girder. (Fig. 127.)—In the Warren girder, or triangular s, there are no vertical struts, and the diagonals may transmit either

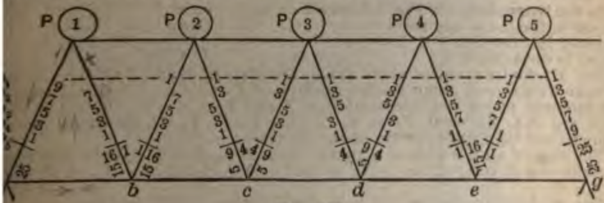


Fig. 127.

sion or compression. The strains in the diagonals may be calculated by method of distribution of strains as in the case of the rectangular truss. n the principle of the lever, the load P_1 being 1/10 of the length of the n from the line of the nearest support a, transmits 9/10 of its weight to a 1/10 to g. Write 9 on the right hand of the strut $1a$, to represent the pression, and 1 on the *right hand of* $1b$, $2c$, $3d$, etc., to represent com- sion, *and on the left hand of* $b2$, $c3$, *etc.*, to represent tension. The load P_2 *smits 7/10 of its weight to* a *and 3/10 to* g. Write 7 on each member from *a and 3 on each member from* 2 to g, placing the figures representing *ression on the right hand of* the member, and those representing *n on the left. Proceed* in the same manner with all the loads, then sum up the figures on each side of each diagonal, and write the differ… of each sum beneath, and on the side of the greater sum, to show wheth… the difference represents tension or compression. The results are as follo… Compression, 1*a*, 25; 2*b*, 15; 3*c*, 5; 3*d*, 5; 4*e*, 15; 5*g*, 25. Tension, 1*b*, 15; … 5; 4*d*, 5; 5*e*, 15. Each of these figures is to be multiplied by 1/10 of on… the loads as P_1, and by the secant of the angle the diagonals make wit… vertical line.

The strains in the horizontal chords may be determined by the method … moments as in the case of rectangular trusses.

Roof-truss.—*Solution by Method of Moments.*—The calculatio… strains in structures by the method of statical moments consists in takin… cross-section of the structure at a point where there are not more th… three members (struts, braces, or chords).

To find the strain in either one of these members take the moment … the intersection of the other two as an axis of rotation. The sum of … moments of these members must be 0 if the structure is in equilibri… But the moments of the two members that pass through the point of re… ence or axis are both 0, hence one equation containing one unknown qu… tity can be found for each cross-section.

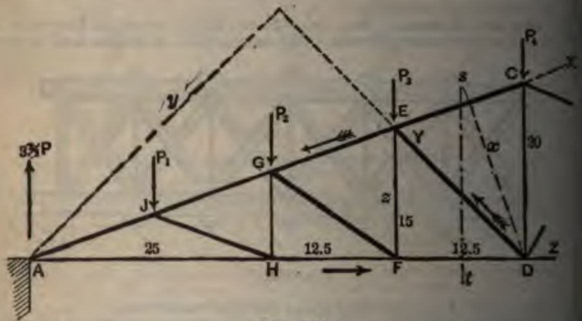


FIG. 128.

In the truss shown in Fig. 128 take a cross-section at *ts*, and determin… strain in the three members cut by it, viz., *CE*, *ED*, and *DF*. Let $X =$ fo… exerted in direction *CE*, $Y =$ force exerted in direction *DE*, $Z =$ force … erted in direction *FD*.

For X take its moment about the intersection of Y and Z at $D = Xx$. F… Y take its moment about the intersection of X and Z at $A = Yy$. For Z … its moment about the intersection of X and Y at $E = Zz$. Let $z =$ 15, … 18.6, $y = 38.4$, $AD = 50$, $CD = 20$ ft. Let P_1, P_2, P_3, P_4 be equal loads … shown, and $3\frac{1}{2}\,P$ the reaction of the abutment A.

The sum of all the moments taken about D or A or E will be 0 whe… structure is at rest. Then $-Xx + 3.5P \times 50 - P_3 \times 12.5 - P_2 \times 25 -$ … $37.5 = 0$.

The $+$ signs are for moments in the direction of the hands of a wat… "clockwise" and $-$ signs for the reverse direction or anti-clockwise. …

$P = P_1 = P_2 - P_3$, $-18.6X + 175P - 75P = 0$; $-18.6X = -100P$; … $100P \div 18.6 = 5.376P$.

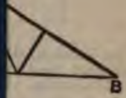
$-Yy + P_3 \times 37.5 + P_2 \times 25 + P_1 \times 12.5 = 0$; $38.4Y = 75P$; $Y = 75P \div$ … $= 1\,953P$.

$-Zz + 3.5P \times 37.5 - P_1 \times 25 - P_2 \times 12.5 - P_3 \times 0 = 0$; $15Z =$ … $6.25P$.

In the same manner the forces exerted in the other member… found as follows: $EG = 6.73P$; $GJ = 8.07P$; $JA = 9.42P$; $JH =$ … $1.59P$; $AH = 8.75P$; $HF = 7.50P$.

The Fink Roof-truss. (Fig. 129.)—An analysis by Prof. … brick (*Van N. Mag.*, Aug. 1880) gives the following results:

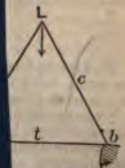
STANDARD
RIVET HOLE
DISTANCES
U.S.A.
Jos. L. White



PL/D − 3 PD/L;
S ÷ L;
PS ÷ L;
PS ÷ D;
S ÷ D;
2 PS ÷ D.

16 ft., with four
tons each at the
ansmit no strain
= 32 ft., D = 16
472, S ÷ D = 2,
are as follows:

= 25.94 tons.
= 3.58 "
= 7.16 "
= 4 "
= 8 "
= 12 "



IG. 129a.

T (soft steel),

sum up the figures on each side of each diagonal, and of each sum beneath, and on the side of the greater su the difference represents tension or compression. T Compression, 1a, 25; 2b, 15; 3c, 5; 3d, 5; 4e, 15; 5g, 5; 4d, 5; 5e, 15. Each of these figures is to be mul the loads as P_1, and by the secant of the angle the vertical line.

The strains in the horizontal chords may be deter moments as in the case of rectangular trusses.

Roof-truss.—*Solution by Method of Mome* strains in structures by the method of statical mom cross-section of the structure at a point where t three members (struts, braces, or chords).

To find the strain in either one of these members the intersection of the other two as an axis of ro moments of these members must be 0 if the stru But the moments of the two members that pass th ence or axis are both 0, hence one equation contai tity can be found for each cross-section.

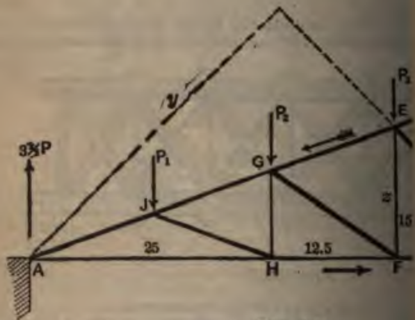


FIG. 128.

In the truss shown in Fig. 128 take a cross-section strain in the three members cut by it, viz., *CE, ED,* exerted in direction *CE*, Y = force exerted in dire erted in direction *FD*.

For X take its moment about the intersection of Y take its moment about the intersection of X and its moment about the intersection of X and Y at 18.6, $y = 38.4$. $AD = 50$, $CD = 20$ ft. Let P_1, P_2, shown, and $3\frac{1}{2}$ P the reaction of the abutment A.

The sum of all the moments taken about D or A structure is at rest. Then $- Xx + 3.5P \times 50 - P_1$ $37.5 = 0$.

The +. signs are for moments in the direction of "clockwise" and − signs for the reverse direction $P = P_1 = P_2 - P_3$. $-18.6X + 175P - 75P = 0$; − $100P + 18.6 = 5.376P$.

$- Yy + P_3 \times 37.5 + P_2 \times 25 + P_1 \times 12.5 = 0$; 38.4 $= 1.953P$.

$- Zz + 3.5P \times 37.5 - P_1 \times 25 - P_2 \times 12.5 - P_3 \times 0$ $6.25P$.

In the same manner the forces exerted in the oth found as follows: $EG = 6.73P$; $GJ = 8.07P$; $JA = 9.4$ $1.59P$; $AH = 8.75P$; $HF = 7.50P$.

The Fink Roof-truss. (Fig. 129.)—An analy brick (*Van N. Mag.*, Aug. 1880) gives the following

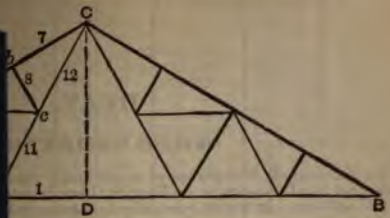


FIG. 129.

oad on roof;
panels on both rafters;
oad at each joint b, d, f, etc.;
on at $A = \frac{1}{2}W = \frac{1}{2}NP = 4P$;
$C = L$; $CD = D$;
n on De, eg, gA, respectively;
ession on Cb, bd, df, and fA.

7, or $bC = c_1 = 7/2\ PL/D - 3\ PD/L$;
8, " bc or $fg = PS \div L$;
9, " $de = 2PS \div L$;
10, " cd or $dg = \frac{1}{2}PS \div D$;
PD/L; 11, " $ec = PS \div D$;
PD/L; 12, " $cC = 3/2\ PS \div D$.

oof-truss of span 64 ft., depth 16 ft., with four
e cut; total load 32 tons, or 4 tons each at the
tons each at A and B, which transmit no strain
re $W = 32$ tons, $P = 4$ tons, $S = 32$ ft., $D = 16$
D. $L \div D = 2.236$, $D \div L = .4472$, $S \div D = 2$,
on the numbered members then are as follows:

tons;	7,	$31.3 - 12 \times .447 = 25.94$	tons.
"	8,	$4 \times .8944 = 3.58$	"
"	9,	$8 \times .8944 = 7.16$	"
"	10,	$2 \times 2 = 4$	"
2 "	11,	$4 \times 2 = 8$	"
2 "	12,	$6 \times 2 = 12$	"

gle.—A structure of tri
supported at a and b. It
nents cc being in compres-
uired the angle θ so that
ture shall be a minimum.
Jan. 17, 1895) gives a solu-
result $\tan\theta = \sqrt{\frac{C+T}{T}}$,
the crushing and the ten-
the material employed.
erial. For $C = T$, $\tan\theta =$
pine), $\tan\theta = 49\frac{1}{4}°$. For $C = 0.8T$ (soft steel),
st iron), $\tan\theta = 69\frac{1}{4}°$.

FIG. 129a.

HEAT.

THERMOMETERS.

The Fahrenheit thermometer is generally used in English-speaking [illegible] tries, and the Centigrade, or French thermometer, in countries that use [illegible] metric system. In many scientific treatises in English, however, the Ce[illegible] grade temperatures are also used, either with or without their Fahren[illegible] equivalents. The Réaumur thermometer is used in Russia, Sweden, Tur[illegible] and Egypt. (Clark.)

In the Fahrenheit thermometer the freezing-point of water is taken at [illegible] and the boiling-point of water at mean atmospheric pressure at the [illegible] level, 14.7 lbs. per sq. in., is taken at 212°, the distance between these [illegible] points being divided into 180°. In the Centigrade and Réaumur thermome[illegible] the freezing-point is taken at 0°. The boiling-point is 100° in the Centigr[illegible] scale, and 80° in the Réaumur.

1 Fahrenheit degree	= 5/9 deg. Centigrade	= 4/9 deg. Réaumur.
1 Centigrade degree	= 9/5 deg. Fahrenheit	= 4/5 deg. Réaumur.
1 Réaumur degree	= 9/4 deg. Fahrenheit	= 5/4 deg. Centigrade.
Temperature Fahrenheit	$= 9/5 \times$ temp. C. + 32°	= 9/4 R. + 32°.
Temperature Centigrade	= 5/9 (temp. F. − 32°)	= 5/4 R.
Temperature Réaumur	= 4/5 temp. C.	= 4/9 (F. − 32°).

Mercurial Thermometer. (Rankine, S. E., p. 234.)—The rate [illegible] expansion of mercury with rise of temperature increases as the temperat[illegible] becomes higher; from which it follows, that if a thermometer showing [illegible] dilatation of mercury simply were made to agree with an air thermome[illegible] at 32° and 212°, the mercurial thermometer would show lower temperatu[illegible] than the air thermometer between those standard points, and higher te[illegible]peratures beyond them.

For example, according to Regnault, when the air thermometer mark[illegible] 350° C. (= 662° F.), the mercurial thermometer would mark 362.16° C. [illegible] 683.89° F.), the error of the latter being in excess 12.16° C. (= 21.89° F.).

Actual mercurial thermometers indicate intervals of temperature pro[illegible]tional to the difference between the expansion of mercury and that of gl[illegible]

The inequalities in the rate of expansion of the glass (which are [illegible] different for different kinds of glass) correct, to a greater or less extent, [illegible] errors arising from the inequalities in the rate of expansion of the merc[illegible]

For practical purposes connected with heat engines, the mercurial th[illegible]mometer made of common glass may be considered as sensibly coinci[illegible] with the air-thermometer at all temperatures not exceeding 500° F.

PYROMETRY.

Principles Used in Various Pyrometers.—Contraction of [illegible] by heat, as in the Wedgwood pyrometer used by potters. Not accurate [illegible] the contraction varies with the quality of the clay.

Expansion of air, as in the air-thermometers, Wiborgh's pyrometer, U[illegible]ling and Steinbart's pyrometer, etc.

Specific heat of solids, as in the copper-ball, platinum-ball, and fire-[illegible] pyrometers.

Relative expansion of two metals or other substances, as copper and [illegible] as in Brown's and Bulkley's pyrometers, etc.

Melting-points of metals, or other substances, as in approximate de[illegible]minations of temperature by melting pieces of zinc, lead, etc.

Measurement of strength of a thermo-electric current produced by h[illegible]ing the junction of two metals, as in Le Chatelier's pyrometer.

Changes in electric resistance of platinum, as in the Siemens pyrom[illegible]

Time required to heat a weighed quantity of water enclosed in a ve[illegible] as in the water pyrometer.

[illegible]meter for Temperatures up to 800° F.—Mercury [illegible] [illegible]rogen in the tube above the mercury. Made by Queen & [illegible]

F.	C.	F.	C.	F.	C.	F.	C.	F.	C.	F.	C.	F.
-40.	26	78.8	92	197.6	158	316.4	224	435.2	290	554	950	1742
-38.2	27	80.6	93	199.4	159	318.2	225	437.	300	572	960	1760
-36.4	28	82.4	94	201.2	160	320.	226	438.8	310	590	970	1778
-34.6	29	84.2	95	203.	161	321.8	227	440.6	320	608	980	1796
-32.8	30	86.	96	204.8	162	323.6	228	442.4	330	626	990	1814
-31.	31	87.8	97	206.6	163	325.4	229	444.2	340	644	1000	1832
-29.2	32	89.6	98	208.4	164	327.2	230	446.	350	662	1010	1850
-27.4	33	91.4	99	210.2	165	329.	231	447.8	360	680	1020	1868
-25.6	34	93.2	100	212.	166	330.8	232	449.6	370	698	1030	1886
-23.8	35	95.	101	213.8	167	332.6	233	451.4	380	716	1040	1904
-22.	36	96.8	102	215.6	168	334.4	234	453.2	390	734	1050	1922
-20.2	37	98.6	103	217.4	169	336.2	235	455.	400	752	1060	1940
-18.4	38	100.4	104	219.2	170	338.	236	456.8	410	770	1070	1958
-16.6	39	102.2	105	221.	171	339.8	237	458.6	420	788	1080	1976
-14.8	40	104.	106	222.8	172	341.6	238	460.4	430	806	1090	1994
-13.	41	105.8	107	224.6	173	343.4	239	462.2	440	824	1100	2012
-11.2	42	107.6	108	226.4	174	345.2	240	464.	450	842	1110	2030
-9.4	43	109.4	109	228.2	175	347.	241	465.8	460	860	1120	2048
-7.6	44	111.2	110	230.	176	348.8	242	467.6	470	878	1130	2066
-5.8	45	113.	111	231.8	177	350.6	243	469.4	480	896	1140	2084
-4.	46	114.8	112	233.6	178	352.4	244	471.2	490	914	1150	2102
-2.2	47	116.6	113	235.4	179	354.2	245	473.	500	932	1160	2120
-0.4	48	118.4	114	237.2	180	356.	246	474.8	510	950	1170	2138
1.4	49	120.2	115	239.	181	357.8	247	476.6	520	968	1180	2156
3.2	50	122.	116	240.8	182	359.6	248	478.4	530	986	1190	2174
5.	51	123.8	117	242.6	183	361.4	249	480.2	540	1004	1200	2192
6.8	52	125.6	118	244.4	184	363.2	250	482.	550	1022	1210	2210
8.6	53	127.4	119	246.2	185	365.	251	483.8	560	1040	1220	2228
0.4	54	129.2	120	248.	186	366.8	252	485.6	570	1058	1230	2246
2.2	55	131.	121	249.8	187	368.6	253	487.4	580	1076	1240	2264
4.	56	132.8	122	251.6	188	370.4	254	489.2	590	1094	1250	2282
5.8	57	134.6	123	253.4	189	372.2	255	491.	600	1112	1260	2300
7.6	58	136.4	124	255.2	190	374.	256	492.8	610	1130	1270	2318
9.4	59	138.2	125	257.	191	375.8	257	494.6	620	1148	1280	2336
1.2	60	140.	126	258.8	192	377.6	258	496.4	630	1166	1290	2354
3.	61	141.8	127	260.6	193	379.4	259	498.2	640	1184	1300	2372
.8	62	143.6	128	262.4	194	381.2	260	500.	650	1202	1310	2390
.6	63	145.4	129	264.2	195	383.	261	501.8	660	1220	1320	2408
.4	64	147.2	130	266.	196	384.8	262	503.6	670	1238	1330	2426
.2	65	149.	131	267.8	197	386.6	263	505.4	680	1256	1340	2444
.	66	150.8	132	269.6	198	388.4	264	507.2	690	1274	1350	2462
.8	67	152.6	133	271.4	199	390.2	265	509.	700	1292	1360	2480
.6	68	154.4	134	273.2	200	392.	266	510.8	710	1310	1370	2498
.4	69	156.2	135	275.	201	393.8	267	512.6	720	1328	1380	2516
2	70	158.	136	276.8	202	395.6	268	514.4	730	1346	1390	2534
	71	159.8	137	278.6	203	397.4	269	516.2	740	1364	1400	2552
8	72	161.6	138	280.4	204	399.2	270	518.	750	1382	1410	2570
6	73	163.4	139	282.2	205	401.	271	519.8	760	1400	1420	2588
4	74	165.2	140	284.	206	402.8	272	521.6	770	1418	1430	2606
2	75	167.	141	285.8	207	404.6	273	523.4	780	1436	1440	2624
	76	168.8	142	287.6	208	406.4	274	525.2	790	1454	1450	2642
	77	170.6	143	289.4	209	408.2	275	527.	800	1472	1460	2660
	78	172.4	144	291.2	210	410.	276	528.8	810	1490	1470	2678
	79	174.2	145	293.	211	411.8	277	530.6	820	1508	1480	2696
	80	176.	146	294.8	212	413.6	278	532.4	830	1526	1490	2714
	81	177.8	147	296.6	213	415.4	279	534.2	840	1544	1500	2732
	82	179.6	148	298.4	214	417.2	280	536.	850	1562	1510	2750
	83	181.4	149	300.2	215	419.	281	537.8	860	1580	1520	2768
	84	183.2	150	302.	216	420.8	282	539.6	870	1598	1530	2786
	85	185.	151	303.8	217	422.6	283	541.4	880	1616	1540	2804
	86	186.8	152	305.6	218	424.4	284	543.2	890	1634	1550	2822
	87	188.6	153	307.4	219	426.2	285	545.	900	1652	1600	2912
	88	190.4	154	309.2	220	428.	286	546.8	910	1670	1650	3002
	89	192.2	155	311.	221	429.8	287	548.6	920	1688	1700	3092
	90	194.	156	312.8	222	431.6	288	550.4	930	1706	1750	[illegible]
	91	195.8	157	314.6	223	433.4	289	552.2	940	1724	1800	[illegible]

F.	C.	F.	C.	F.	C.	F.	C.	F.	C.	F.	C.
-40	-40.	26	-3.3	92	33.3	158	70.	224	106.7	290	143.3
-39	-39.4	27	-2.8	93	33.9	159	70.6	225	107.2	291	143.9
-38	-38.9	28	-2.2	94	34.4	160	71.1	226	107.8	292	144.4
-37	-38.3	29	-1.7	95	35.	161	71.7	227	108.3	293	145.
-36	-37.8	30	-1.1	96	35.6	162	72.2	228	108.9	294	145.6
-35	-37.2	31	-0.6	97	36.1	163	72.8	229	109.4	295	146.1
-34	-36.7	32	0.	98	36.7	164	73.3	230	110.	296	146.7
-33	-36.1	33	+0.6	99	37.2	165	73.9	231	110.6	297	147.2
-32	-35.6	34	1.1	100	37.8	166	74.4	232	111.1	298	147.8
-31	-35.	35	1.7	101	38.3	167	75.	233	111.7	299	148.3
-30	-34.4	36	2.2	102	38.9	168	75.6	234	112.2	300	148.9
-29	-33.9	37	2.8	103	39.4	169	76.1	235	112.8	301	149.4
-28	-33.3	38	3.3	104	40.	170	76.7	236	113.3	302	150.
-27	-32.8	39	3.9	105	40.6	171	77.2	237	113.9	303	150.6
-26	-32.2	40	4.4	106	41.1	172	77.8	238	114.4	304	151.1
-25	-31.7	41	5.	107	41.7	173	78.3	239	115.	305	151.7
-24	-31.1	42	5.6	108	42.2	174	78.9	240	115.6	306	152.2
-23	-30.6	43	6.1	109	42.8	175	79.4	241	116.1	307	152.8
-22	-30.	44	6.7	110	43.3	176	80.	242	116.7	308	153.3
-21	-29.4	45	7.2	111	43.9	177	80.6	243	117.2	309	153.9
-20	-28.9	46	7.8	112	44.4	178	81.1	244	117.8	310	154.4
-19	-28.3	47	8.3	113	45.	179	81.7	245	118.3	311	155.
-18	-27.8	48	8.9	114	45.6	180	82.2	246	118.9	312	155.6
-17	-27.2	49	9.4	115	46.1	181	82.8	247	119.4	313	156.1
-16	-26.7	50	10.	116	46.7	182	83.3	248	120.	314	156.7
-15	-26.1	51	10.6	117	47.2	183	83.9	249	120.6	315	157.2
-14	-25.6	52	11.1	118	47.8	184	84.4	250	121.1	316	157.8
-13	-25.	53	11.7	119	48.3	185	85.	251	121.7	317	158.3
-12	-24.4	54	12.2	120	48.9	186	85.6	252	122.2	318	158.9
-11	-23.9	55	12.8	121	49.4	187	86.1	253	122.8	319	159.4
-10	-23.3	56	13.3	122	50.	188	86.7	254	123.3	320	160.
-9	-22.8	57	13.9	123	50.6	189	87.2	255	123.9	321	160.6
-8	-22.2	58	14.4	124	51.1	190	87.8	256	124.4	322	161.1
-7	-21.7	59	15.	125	51.7	191	88.3	257	125.	323	161.7
-6	-21.1	60	15.6	126	52.2	192	88.9	258	125.6	324	162.2
-5	-20.6	61	16.1	127	52.8	193	89.4	259	126.1	325	162.8
-4	-20.	62	16.7	128	53.3	194	90.	260	126.7	326	163.3
-3	-19.4	63	17.2	129	53.9	195	90.6	261	127.2	327	163.9
-2	-18.9	64	17.8	130	54.4	196	91.1	262	127.8	328	164.4
-1	-18.3	65	18.3	131	55.	197	91.7	263	128.3	329	165.
0	-17.8	66	18.9	132	55.6	198	92.2	264	128.9	330	165.6
+1	-17.2	67	19.4	133	56.1	199	92.8	265	129.4	331	166.1
2	-16.7	68	20.	134	56.7	200	93.3	266	130.	332	166.7
3	-16.1	69	20.6	135	57.2	201	93.9	267	130.6	333	167.2
4	-15.6	70	21.1	136	57.8	202	94.4	268	131.1	334	167.8
5	-15.	71	21.7	137	58.3	203	95.	269	131.7	335	168.3
6	-14.4	72	22.2	138	58.9	204	95.6	270	132.2	336	168.9
7	-13.9	73	22.8	139	59.4	205	96.1	271	132.8	337	169.4
8	-13.3	74	23.3	140	60.	206	96.7	272	133.3	338	170.
9	-12.8	75	23.9	141	60.6	207	97.2	273	133.9	339	170.6
10	-12.2	76	24.4	142	61.1	208	97.8	274	134.4	340	171.1
11	-11.7	77	25.	143	61.7	209	98.3	275	135.	341	171.7
12	-11.1	78	25.6	144	62.2	210	98.9	276	135.6	342	172.2
13	-10.6	79	26.1	145	62.8	211	99.4	277	136.1	343	172.8
14	-10.	80	26.7	146	63.3	212	100.	278	136.7	344	173.3
15	-9.4	81	27.2	147	63.9	213	100.6	279	137.2	345	173.9
16	-8.9	82	27.8	148	64.4	214	101.1	280	137.8	346	174.4
17	-8.3	83	28.3	149	65.	215	101.7	281	138.3	347	175.
18	-7.8	84	28.9	150	65.6	216	102.2	282	138.9	348	175.6
19	-7.2	85	29.4	151	66.1	217	102.8	283	139.4	349	176.1
20	-6.7	86	30.	152	66.7	218	103.3	284	140.	350	176.7
21	[illegible]	87	30.6	153	67.2	219	103.9	285	140.6	351	177.2
			31.1	154	67.8	220	104.4	286	141.1	352	177.8
			31.7	155	68.3	221	105.	287	141.7	353	178.3
			32.2	156	68.9	222	105.6	288	142.2	354	178.9
			32.8	157	69.4	223	106.1	289	142.8	355	179.4

tinum or Copper Ball Pyrometer.—A weighed piece of um, copper, or iron is allowed to remain in the furnace or heated er till it has attained the temperature of its surroundings. It is then ly taken out and dropped into a vessel containing water of a known t and temperature. The water is stirred rapidly and its maximum rature taken. Let W = weight of the water, w the weight of the ball, e original and T the final heat of the water, and S the specific heat of etal; then the temperature of fire may be found from the formula

$$x = \frac{W(T-t)}{wS} + T.$$

mean specific heat of platinum between 32° and 446° F. is .03333 or hat of water, and it increases with the temperature about .000305 for 100° F. For a fuller description, by J. C. Hoadley, see Trans. A. S. M. E., . Compare also Henry M. Howe, Trans. A. I. M. E., xviii. 728.

accuracy corrections are required for variations in the specific heat of ater and of the metal at different temperatures, for loss of heat by tion from the metal during the transfer from the furnace to the water, rom the apparatus during the heating of the water; also for the heat-bing capacity of the vessel containing the water.

-clay or fire-brick may be used instead of the metal ball.

Chatelier's Thermo-electric Pyrometer.—For a very full iption see paper by Joseph Struthers, *School of Mines Quarterly*, vol. 91; also, paper read by Prof. Roberts-Austen before the Iron and Steel ute, May 7, 1891.

principle upon which this pyrometer is constructed is the measure- of a current of electricity produced by heating a couple composed of ires, one platinum and the other platinum with 10% rhodium—the cur- produced being measured by a galvanometer.

composition of the gas which surrounds the couple has no influence e indications.

en temperatures above 2500° F. are to be studied, the wires must have lating support and must be of good length, so that all parts of a fur- can be reached.

a Siemens furnace, about 11½ feet is the general length. The wires pported in an iron tube, ½ inch interior diameter and held in place by nder of refractory clay having two holes bored through, in which the are placed. The shortness of time (five seconds) allows the tempera- o be taken without deteriorating the tube.

s made by this pyrometer in measuring furnace temperatures under t variety of conditions show that the readings of the scale uncorrected ways within 45° F. of the correct temperature, and in the majority of rial measurements this is sufficiently accurate. Le Chatelier's py- er is sold by Queen & Co., of Philadelphia.

duation of Le Chatelier's Pyrometer.—W. C. Roberts- n in his Researches on the Properties of Alloys, Proc. Inst. M. E. 1892, The electromotive force produced by heating the thermo-junction given temperature is measured by the movement of the spot of light scale graduated in millimetres. A formula for converting the divi- f the scale into thermometric degrees is given by M. Le Chatelier; but tter to calibrate the scale by heating the thermo-junction to temper- which have been very carefully determined by the aid of the air- ometer, and then to plot the curve from the data so obtained. Many and boiling-points have been established by concurrent evidence of kinds, and are now very generally accepted. The following table s certain of these:

F.	Deg. C.		Deg. F.	Deg. C.	
	100	Water boils.	1733	945	Silver melts.
	326	Lead melts.	1859	1015	Potassium sulphate melts.
	358	Mercury boils.	1913	1045	Gold melts.
	415	Zinc melts.	1929	1054	Copper melts.
	448	Sulphur boils.	2732	1500	Palladium melts.
	625	Aluminum melts.	3227	1775	Platinum melts.
	665	Selenium boils.			

Temperatures Developed in Industrial Furnaces.— Chatelier states that by means of his pyrometer he has discovere e temperatures which occur in melting steel and in other industri ons have been hitherto overestimated.

M. Le Chatelier finds the melting heat of white cast iron 1135° (2
and that of gray cast iron 1220° (2228° F.). Mild steel melts at 147
F.), semi-mild at 1455° (2651° F.), and hard steel at 1410° (2570° F
furnace for hard porcelain at the end of the baking has a heat
(2498° F.). The heat of a normal incandescent lamp is 1800° (3272°
it may be pushed to beyond 2100° (3812° F.).

Prof. Roberts-Austen (Recent Advances in Pyrometry, Trans. A.
Chicago Meeting, 1893) gives an excellent description of modern
pyrometers. The following are some of his temperature determina

GOLD-MELTING, ROYAL MINT.

	Degrees. Centigrade.
Temperature of standard alloy, pouring into moulds.	1180
Temperature of standard alloy, pouring into moulds (on a previous occasion, by thermo-couple)	1147
Annealing blanks for coinage, temperature of chamber..	800

SILVER-MELTING, ROYAL MINT.

Temperature of standard alloy, pouring into mould......	980

TEN-TON OPEN-HEARTH FURNACE, WOOLWICH ARSENAL.

Temperature of steel, 0.3% carbon, pouring into ladle.....	1645
Temperature of steel, 0.3% carbon, pouring into large mould	1580
Reheating furnace, Woolwich Arsenal, temperature of interior	930
Cupola furnace, temperature of No. 2, cast-iron pouring into ladle	1600

The following determinations have been effected by M. Le Chatel

BESSEMER PROCESS.

Six-ton Converter.

	Degrees. Centigrade
A. Bath of slag	1580
B. Metal in ladle	1640
C. Metal in ingot mould	1580
D. Ingot in reheating furnace	1200
E. Ingot under the hammer	1080

OPEN-HEARTH FURNACE (Siemens).

Semi-Mild Steel.

A. Fuel gas near gas generator	720
B. Fuel gas entering into bottom of regenerator chamber	400
C. Fuel gas issuing from regenerator chamber	1200
Air issuing from regenerator chamber	1000

CHIMNEY GASES.

Furnace in perfect condition	300

OPEN-HEARTH FURNACE.

End of the melting of pig charge	1420
Completion of conversion	1500

MOLTEN STEEL.

In the ladle—Commencement of casting	1580
End of casting	1490
In the moulds	1520

For very mild (soft) steel the temperatures are higher by 50° C.

SIEMENS CRUCIBLE OR POT FURNACE.

1600° C., 2912° F.

ROTARY PUDDLING FURNACE.

	Degrees C
Furnace	1340-12
Puddled ball—End of operation	13

ir-volume V' to the air-volume V, can be measured by a ma-
t this pressure is of course a function of the temperature T.
troduction of V', we have the two separate air-volumes, V at
ure T and V' at the temperature t, both under the atmospheric
After the forcing in of V' into the globe, we have, on the
y the volume V of the temperature T, but under the pressure

h Air-pyrometer is adapted for use at blast-furnaces, smelting-
ning and tempering furnaces, etc., where determinations of
from 0° to 2400° F. are required.

Fire-clay Pyrometer. (H. M. Howe, *Eng. and Mining*
7, 1890.)—Professor Seger uses a series of slender triangular
amids, about 3 inches high and ⅝ inch wide at the base, and
ess fusible than the next: these he calls "normal pyramids"
gel"). When the series is placed in a furnace whose temper-
ually raised, one after another will bend over as its range of
eached; and the temperature at which it has bent, or "wept,"
apex touches the hearth of the furnace or other level surface
standing, is selected as a point on Seger's scale. These points
rately determined by some absolute method, or they may
to give comparative results. Unfortunately, these pyramids
ications when the temperature is stationary or falling.

nd Nouel's Pyrometric Telescope. (*Ibid.*)—Mesuré
yrometric telescope gives us an immediate determination of
ture of incandescent bodies, and is therefore much better
ses where a great number of observations are to be made, and
vals, than Seger's. Such cases arise in the careful heating of
tle telescope, carried in the pocket or hung from the neck, can
reman or heater at any moment.

n the fact that a plate of quartz, cut at right angles to the
he plane of polarization of polarized light to a degree nearly
portional to the square of the length of the waves; and,
e fact that while a body at dull redness merely emits red
emperature rises, the orange, yellow, green, and blue waves
ppear.

h a plate of quartz is placed between two Nicol prisms at
"a ray of monochromatic light which passes the first, or

and the suction is obtained by an aspirator and regulated
water of constant height.

The tension in the chamber between the apertures is
manometer.

The Air-thermometer. (Prof. R. C. Carpenter, *Eng*
1893.)—Air is a perfect thermometric substance, and if a gi
be considered, the product of its pressure and volume
absolute temperature is in every case constant. If the
remain constant, the temperature will vary with the pr
pressure remain constant the temperature will vary with th
the former condition is more easily attained air-thermomete
constructed of constant volume, in which case the absolut
will vary with the pressure.

If we denote pressure by p and p', the corresponding ab
atures by T and T', we should have

$$p : p' :: T : T' \quad \text{and} \quad T' = p'\frac{T}{p}.$$

The absolute temperature T is to be considered in every
than the thermometer-reading expressed in Fahrenheit degr
form of the above equation, if the pressure p correspondi
absolute temperature T be known, T' can be found. The qu
constant which may be used in all determinations with the in
pressure on the instrument can be expressed in inches of m
evidently the atmospheric pressure b as shown by a baro
minus an additional amount h shown by a manometer atta
thermometer.

That is, in general, $p = b \times h$.

The temperature of 32° F. is fixed as the point of melting
case $T = 460 \times 32 = 492°$ F. This temperature can be pr
rounding the bulb in melting ice and leaving several minut
temperature of the confined air shall acquire that of the su
When the air is at that temperature, note the reading o
manometer h, and that of a barometer; the sum will be the v
responding to the absolute temperature of 492° F. The c
instrument, $K = 492 - p$, once obtained, can be used in all fut

ar of iron, slowly heated in contact with air, assumes the fol- at annexed temperatures (Claudel):

	Cent.	Fahr.		Cent.	Fahr.
....	225	437	Indigo at..............	288	550
....	243	473	Blue at..............	293	559
....	265	509	Green at.............	332	630
....	277	531	"Oxide-gray"........	400	752

NG POINTS AT ATMOSPHERIC PRESSURE.

14.7 lbs. per square inch.

uric..............	100° F.	Average sea-water..........	213.2° F.
phide..........	118	Saturated brine............	226
....	140	Nitric acid..................	248
....	140	Oil of turpentine...........	315
....	145	Phosphorus..................	554
....	150	Sulphur......................	570
....	173	Sulphuric acid..............	590
....	176	Linseed oil.................	597
....	212	Mercury.....................	676

points of liquids increase as the pressure increases. The boil- water at any given pressure is the same as the temperature of am of the same pressure. (See Steam.)

TING-POINTS OF VARIOUS SUBSTANCES.

ing figures are given by Clark (on the authority of Pouillet, Wilson), except those marked *, which are given by Prof. Rob- n his description of the Le Chatelier pyrometer. These latter the most reliable figures.

acid	− 148° F.	Alloy, 1 tin, 1 lead..	370 to 466° F.
d............	− 108	Tin	442 to 446
....	− 39	Cadmium..................	442
....	+ 9.5	Bismuth............	504 to 507
....	14	Lead................	608 to 618*
acid..............	16	Zinc	680 to 779*
....	32	Antimony	810 to 1150
ne..............	45	Aluminum..................	1157*
....	92	Magnesium..................	1200
....	112	Calcium	Full red heat.
....	113	Bronze	1692
....	109 to 120	Silver..............	1733* to 1873
....	120	Potassium sulphate........	1859*
d	131 to 140	Gold	1913* to 2282
....	136 to 144	Copper	1929* to 1996
....	142 to 154	Cast iron, white...	1922 to 2075*
....	158	" gray	2012 to 2786 2228*
....	194 to 208	Steel..............	2372 to 2532
2 tin, 5 bismuth	199	" hard	2570*; mild, 2687*
....	225	Wrought iron......	2732 to 2912
....	239	Palladium..................	2732*
, 1 lead..........	334	Platinum..................	3227*

g-point of fusible alloys, see Alloys.
kel, and manganese, fusible in highest heat of a forge. Tung- omium, not fusible in forge, but soften and agglomerate. Plati- lium, fusible only before the oxyhydrogen blowpipe.

NTITATIVE MEASUREMENT OF HEAT.

Heat.—The British unit of heat, or British thermal unit that quantity of heat which is required to raise the temperature re water 1° Fahr., at or near 39°.1 F., the temperature of maxi- of water.

thermal unit, or calorie, is that quantity of heat which is re- *e the temperature of 1 kilogramme* of pure water 1° Cent., at or *hich is equivalent to 39°.1 F.*

lorie = 3.968 British thermal units; 1 B. T. U. = .252 calorie. *lorie " is sometimes used* by English writers; it is the quan-

tity of heat required to raise the temperature of 1 lb. of water 1° C
calorie = 9/5 B.T.U. = 0.4536 calorie. The heat of combustion of carb
CO_2, is said to be 8080 calories. This figure is used either for French calor
for pound calories, as it is the number of pounds of water that can be r
1° C. by the complete combustion of 1 lb. of carbon, or the numb
kilogrammes of water that can be raised 1° C. by the combustion of 1
of carbon; assuming in each case that all the heat generated is transf
to the water.

The Mechanical Equivalent of Heat is the number of
pounds of mechanical energy equivalent to one British thermal unit,
and mechanical energy being mutually convertible. Joule's experim
1843–50, gave the figure 772, which is known as Joule's equivalent. Mo
cent experiments by Prof. Rowland (*Proc. Am. Acad. Arts and Sci*
1880; see also *Wood's Thermodynamics*) give higher figures, and the
probable average is now considered to be 778.

1 heat-unit is equivalent to 778 ft.-lbs. of energy. 1 ft. lb. = 1/778 = .0
heat-units. 1 horse-power = 33,000 ft.-lbs. per minute = 2545 heat-unit
hour = 42.416 + per minute = .70694 per second. 1 lb. carbon burned to
= 14,544 heat-units. 1 lb. C. per H.P. per hour = 2545 ÷ 14544 = 17½% effic
(.174986).

Heat of Combustion of Various Substances in Oxyg

	Heat-units.		Authority.
	Cent.	Fahr.	
Hydrogen to liquid water at 0° C....	34,462	62,032	Favre and Silberm
	33,808	60,854	Andrews.
	34,342	61,816	Thomsen.
" to steam at 100° C.........	28,732	51,717	Favre and Silberm
Carbon (wood charcoal) to carbonic acid, CO_2; ordinary temperatures.	8,080	14,544	"
	7,900	14,220	Andrews.
	8,137	14,647	Berthelot.
Carbon, diamond to CO_2	7,859	14,146	"
" black diamond to CO_2.	7,861	14,150	"
" graphite to CO_2	7,901	14,222	"
Carbon to carbonic oxide, CO.........	2,473	4,451	Favre and Silberm
Carbonic oxide to CO_2, per unit of CO	2,403	4,325	"
	2,431	4,376	Andrews.
	2,385	4,293	Thomsen.
CO to CO_2 per unit of C = 2⅓ × 2403	5,607	10,093	Favre and Silberm
Marsh-gas, Methane, CH_4 to water and CO_2....	13,120	23,616	Thomsen.
	13,108	23,594	Andrews.
	13,063	23,513	Favre and Silberm
Olefiant gas, Ethylene, C_2H_4 to water and CO_2	11,858	21,344	"
	11,942	21,496	Andrews.
	11,957	21,523	Thomsen.
Benzole gas, C_6H_6 to water and CO_2	10,102	18,184	"
	9,915	17,847	Favre and Silberm

In burning 1 pound of hydrogen with 8 pounds of oxygen to form 9 p
of water, the units of heat evolved are 62,032 (Favre and S.); but t
resulting product is not cooled to the initial temperature of the
part of the heat is rendered latent in the steam. The total heat of
of steam at 212° F. is 1146.1 heat-units above that of water at 32°
9 × 1146.1 = 10,315 heat-units, which deducted from 62,032 gives 51,717
heat evolved by the combustion of 1 lb. of hydrogen and 8 lbs. of oxy
32° F. to form steam at 212° F.

By the decomposition of a chemical compound as much heat is abs
or rendered latent as was evolved when the compound was formed.
of carbon is burned to CO_2, generating 14,544 B.T.U., and the CO_2 thus f
ly reduced to CO in the presence of glowing carbon.
C = 2CO, the result is the same as if the 2 lbs. C ha
to 2CO, generating 2 × 4451 = 8902 heat-units; conseq
542 heat-units have disappeared or become latent, a

nburning" of CO_2 to CO is thus a cooling operation. (For heats of com-
ation of various fuels, see Fuel.)

SPECIFIC HEAT.

Thermal Capacity.—The thermal capacity of a body is the quantity heat required to raise its temperature one degree. The ratio of the heat quired to raise the temperature of a given substance one degree to that quired to raise the temperature of water one degree from the temperature maximum density 39.1 is commonly called the *specific heat* of the substance. Some writers object to the term as being an inaccurate use of the rds "specific" and "heat." A more correct name would be "coefficient thermal capacity."

Determination of Specific Heat.—*Method by Mixture.*—The ly whose specific heat is to be determined is raised to a known temperae, and is then immersed in a mass of liquid of which the weight, specific at, and temperature are known. When both the body and the liquid e attained the same temperature, this is carefully ascertained.

ow the quantity of heat lost by the body is the same as the quantity of at absorbed by the liquid.

et c, w, and t be the specific heat, weight, and temperature of the hot dy, and c', w', and t' of the liquid. Let T be the temperature the mixe assumes.

Then, by the definition of specific heat, $c \times w \times (t - T) =$ heat-units lost the hot body, and $c' \times w' \times (T - t') =$ heat-units gained by the cold uid. If there is no heat lost by radiation or conduction, these must be ual, and

$$cw(t - T) = c'w'(T - t') \quad \text{or} \quad c = \frac{c'w'(T - t')}{w(t - T)}.$$

Specific Heats of Various Substances.

The specific heats of substances, as given by different authorities, show siderable lack of agreement, especially in the case of gases.

he following tables give the mean specific heats of the substances named ording to Regnault. (From Rontgen's Thermodynamics, p. 134.) These cific heats are average values, taken at temperatures which usually come der observation in technical application. The actual specific heats of all stances, in the solid or liquid state, increase slowly as the body expands as the temperature rises. It is probable that the specific heat of a body en liquid is greater than when solid. For many bodies this has been ified by experiment.

SOLIDS.

imony	0.0508	Steel (soft)	0.1165
per	0.0951	Steel (hard)	0.1175
d	0.0324	Zinc	0.0956
ought iron	0.1138	Brass	0.0939
ss	0.1937	Ice	0.5040
t iron	0.1298	Sulphur	0.2026
d	0.0314	Charcoal	0.2410
inum	0.0324	Alumina	0.1970
er	0.0570	Phosphorus	0.1887
	0.0562		

LIQUIDS.

er	1.0000	Mercury	0.0333
d (melted)	0.0402	Alcohol (absolute)	0.7000
hur "	0.2340	Fusel oil	0.5640
nuth "	0.0308	Benzine	0.4500
"	0.0637	Ether	0.5034
huric acid	0.8350		

GASES.

	Constant Pressure.	Constant Volum[e]
Air	0.23751	0.16847
Oxygen	0.21751	0.15507
Hydrogen	3.40900	2.41226
Nitrogen	0.24380	0.17273
Superheated steam	0.4805	0.346
Carbonic acid	0.217	0.1535
Olefiant Gas (CH_2)	0.404	0.173
Carbonic oxide	0.2479	0.1758
Ammonia	0.508	0.299
Ether	0.4797	0.3411
Alcohol	0.4534	0.3200
Acetic acid	0.4125	......
Chloroform	0.1567	......

In addition to the above, the following are given by other author[s] (Selected from various sources.)

METALS.

Platinum, 32° to 446° F. (increased .000305 for each 100° F.)	.0333	Wrought iron (Petit & Dulong).	
Cadmium	.0567	" 32° to 212°	
Brass	.0939	" 32° to 392°	
Copper, 32° to 212° F.	.094	" 32° to 572°	
" 32° to 572° F.	.1013	" 32° to 662°	
Zinc 32° to 212° F.	.0927	Wrought iron (J. C. Hoadley, A. S. M. E., vi. 713),	
" 32° to 572° F.	.1015	Wrought iron, 32° to 200°	
Nickel	.1086	" 32° to 600°	
Aluminum, 0° F. to melting-point (A. E. Hunt)	0.2185	" 32° to 2000°	

OTHER SOLIDS.

Brickwork and masonry, about	.20	Coal	.20 [to]
Marble	.210	Coke	
Chalk	.215	Graphite	
Quicklime	.217	Sulphate of lime	
Magnesian limestone	.217	Magnesia	
Silica	.191	Soda	
Corundum	.198	Quartz	
Stones generally	.2 to .22	River sand	

WOODS.

Pine (turpentine)	.467	Oak	
Fir	.650	Pear	

LIQUIDS.

Alcohol, density .793	.622	Olive oil	
Sulphuric acid, density 1.87	.335	Benzine	
" " 1.30	.661	Turpentine, density .872	
Hydrochloric acid	.600	Bromine	

GASES.

	At Constant Pressure.	At Cons[tant] Volume.
Sulphurous acid	.1553	.1246
Light carburetted hydrogen, marsh gas (CH_4)	.5929	.4683
Blast-furnace gases	.2277	.....

Specific Heat of Salt Solution. (Schuller.)

Per cent salt in solution	5	10	15	20
Specific heat	.9306	.8909	.8606	.8490

Specific Heat of Air.—Regnault gives for the mean value

Between — 30° C. and + 10° C.	0.2377
" 0° C. " 100° C.	0.2374
" 0° C. " 200° C.	0.2375

Ha[illegible] *0.1686* for the specific heat of air at constant volume[.] value [illegible]nstant has never been found to any degree of accura[cy] [illegible]t. Prof. Wood gives 0.2375 ÷ 1.406 = 0.1689. The

: heat of a fixed gas at constant pressure to the sp. ht. at con-
ne is given as follows by different writers (*Eng'g*, July 12, 1889):
1.3953; Moll and Beck, 1.4085; Szathmari, 1.4027; J. Macfarlane
The first three are obtained from the velocity of sound in air. The
erived from theory. Prof. Wood says: The value of the ratio for
ind in the days of La Place, was 1.41, and we have 0.2377 ÷ 1.41
ie value used by Clausius, Hanssen, and many others. But this
definitely known. Rankine in his later writings used 1.408, and
recent work gives 1.404, while some experiments gives less than
ers more than 1.41. Prof. Wood uses 1.406.

: **Heat of Gases.**—Experiments by Mallard and Le Chatelier
continuous increase in the specific heat at constant volume of
, and even of the perfect gases, with rise of temperature. The
inappreciable at 100° C., but increases rapidly at the high tem-
of the gas-engine cylinder. (Robinson's Gas and Petroleum

Heat and Latent Heat of Fusion of Iron and Steel. (H. H. Campbell, Trans. A. I. M. E., xix. 181.)

		Åkerman.	Troilius.
fic heat pig iron,	0 to 1200° C.	0.16	....
" "	1200 to 1800° C.	0.21	....
" "	0 to 1500° C.	....	0.18
" "	1500 to 1800° C.	....	0.20

ng by both sets of data we have :

	Åkerman.	Troilius.
ng from 0 to 1800° C.	318	330 calories per kilo.

probable value is about.......... 325 calories per kilo.

fic heat, steel (probably high carbon)	(Troilius)	.1175
" soft iron	"	.1081
probable value solid rail steel	...	.1125
" " melted rail steel		.1275

	Åkerman.	Troilius.
t heat of fusion, pig iron, calories per kilo.	46	..
" " gray pig	..	33
" " white pig	..	23

we may assume that the truth is about: Steel, 20; pig iron, 30.

EXPANSION BY HEAT.

tigrade scale the coefficient of expansion of air per degree is
/273; that is, the pressure being constant, the volume of a perfect
es 1/273 of its volume at 0° C. for every increase in temperature
Fahrenheit units it increases 1/491.2 = .002036 of its volume at
very increase of 1° F.

on of Gases by Heat from 32° to 212° F. (Regnault.)

	Increase in Volume, Pressure Constant. Volume at 32° Fahr. = 1.0, for		Increase in Pressure, Volume Constant. Pressure at 32° Fahr. = 1.0, for	
	100° C.	1° F.	100° C.	1° F.
....	0.3661	0.002034	0.3667	0.002037
ic air....	0.3670	0.002039	0.3665	0.002036
....	0.3670	0.002039	0.3668	0.002039
xide.................	0.3669	0.002038	0.3667	0.002037
cid	0.3710	0.002061	0.3688	0.002039
acid	0.3903	0.002168	0.3845	0.002136

ume is kept *constant*, the pressure varies directly as the
e.

Lineal Expansion of Solids at Ordinary Temperatures.

(British Board of Trade; from CLARK.)

	For 1° Fahr.	For 1° Cent.	Coefficient of Expansion from 32° to 212° F.	According to Other Authorities.
	Length=1	Length=1		
Aluminum (cast)	.00001234	.00002221	.002221	....
Antimony (cryst.)	.00000627	.00001129	.001129	.001083
Brass, cast	.00000957	.00001722	.001722	.001868
" plate	.00001052	.00001894	.001894	....
Brick	.00000306	.00000550	.000550	....
Bronze (Copper, 17; Tin, 2½; Zinc 1)	.00000986	.00001774	.001774	....
Bismuth	.00000975	.00001755	.001755	.001392
Cement, Portland (mixed), pure	.00000594	.00001070	.001070	....
Concrete: cement, mortar, and pebbles	.00000795	.00001430	.001430	....
Copper	.00000887	.00001596	.001596	.001718
Ebonite	.00004278	.00007700	007700	....
Glass, English flint	.00000451	.00000812	.000812	....
" thermometer	.00000499	.00000897	.000897	....
" hard	.00000397	.00000714	.000714	....
Granite, gray, dry	.00000438	.00000789	.000789	....
" red, dry	.00000498	.00000897	.000897	....
Gold, pure	.00000786	.00001415	.001415	....
Iridium, pure	.00000356	.00000641	.000641	....
Iron, wrought	.00000648	.00001166	.001166	.001235
" cast	.00000556	.00001001	.001001	.001110
Lead	.00001571	.00002828	.002828	....
Magnesium	....	....	....	.002694
Marbles, various { from	.00000308	.00000554	000554	....
Marbles, various { to	.00000786	.00001415	.001415	....
Masonry, brick { from	.00000256	.00000460	.000460	....
Masonry, brick { to	.00000494	.00000890	.000890	....
Mercury (cubic expansion)	.00009984	.00017971	.017971	.018018
Nickel	.00000695	.00001251	.001251	.001279
Pewter	.00001129	.00002033	.002033	....
Plaster, white	.00000922	.00001660	.001660	....
Platinum	.00000479	.00000863	.000863	....
Platinum, 85 per cent } Iridium, 15 " " }	.00000453	.00000815	.000815	.000884
Porcelain	.00000200	.00000360	.000360	....
Quartz, parallel to major axis, t 0° to 40° C.	.00000434	.00000781	.000781	....
Quartz, perpendicular to major axis, t 0° to 40° C.	.00000788	.00001419	.001419	....
Silver, pure	.00001079	.00001943	.001943	.001908
Slate	.00000577	.00001038	.001038	....
Steel, cast	.00000636	.00001144	.001144	.001079
" tempered	.00000689	.00001240	.001240	....
Stone (sandstone), dry	.00000652	.00001174	.001174	....
" " Rauville	.00000417	.00000750	.000750	....
Tin	.00001163	.00002094	.002094	.001938
Wedgwood ware	.00000489	.00000881	.000881	....
Wood, pine	.00000276	.00000496	.000496	....
Zinc	.00001407	.00002532	.002532	.0029[illegible]
Zinc, 8 } *Tin, 1* }	.00001496	.00002692	.002692	....

Cubical expansion, or expansion of volume = linear expansion × 3.

e Temperature—Absolute Zero.—The absolute zero of a retical consequence of the law of expansion by heat, assuming sible to continue the cooling of a perfect gas until its volume is o nothing.

me of a perfect gas increases 1/273 of its volume at 0° C. for se of temperature of 1° C., and decreases 1/273 of its volume for se of temperature of 1° C., then at − 273° C. the volume of the as would be reduced to nothing. This point − 273° C., or 491.2° e melting-point of ice on the air thermometer, or 492.66° F. be- rfect gas thermometer = − 459.2° F. (or − 460.66°), is called the o; and absolute temperatures are temperatures measured, on ahrenheit or centigrade scale, from this zero. The freezing corresponds to 491.2° F. absolute. If p_0 be the pressure and e of a gas at the temperature of 32° F. = 491.2° on the absolute nd p the pressure, and v the volume of the same quantity of ther absolute temperature T, then

$$\frac{pv}{p_0v_0} = \frac{T}{T_0} = \frac{t + 459.2}{491.2}; \quad \frac{pv}{T} = \frac{p_0v_0}{T_0}.$$

of $p_0v_0 \div T_0$ for air is 53.37, and $pv = 53.37T$, calculated as fol- . Wood:

ot of dry air at 32° F. at the sea-level weighs 0.080728 lb. The ne pound is $v_0 = \frac{1}{.080728} = 12.387$ cubic feet. The pressure per s 2116.2 lbs.

$$\frac{p_0v_0}{T_0} = \frac{2116.2 \times 12.387}{491.13} = \frac{26214}{491.13} = 53.37.$$

491.13 is the number of degrees that the absolute zero is below point of ice, by the air thermometer. On the absolute scale, ons would be indicated by a perfect gas thermometer, the cal- e approximately is 492.66, which would make $pv = 53.21T$. Prof. nsiders that − 273.1° C., = − 459.4° F., is the most probable value te zero. See *Heat* in *Ency. Brit.*

ion of Liquids from 32° to 212° F.—Apparent ex- lass (Clark). Volume at 212°, volume at 32° being 1:

....	1.0466	Nitric acid....................	1.11
ated with salt.....	1.05	Olive and linseed oils...........	1.08
....	1.0182	Turpentine and ether	1.07
....	1.11	Hydrochlor. and sulphuric acids	1.06

at various temperatures, see Water.
various temperatures, see Air.

T HEATS OF FUSION AND EVAPORATION.

Heat means a quantity of heat which has disappeared, having ed to produce some change other than elevation of temperature. reversing that change, the quantity of heat which has dis- reproduced. Maxwell defines it as the quantity of heat which mmunicated to a body in a given state in order to convert it into e without changing its temperature.

Heat of Fusion.—When a body passes from the solid to the its temperature remains stationary, or nearly stationary, at a ing point during the whole operation of melting; and in order t operation go on, a quantity of heat must be transferred to the nelted, being a certain amount for each unit of weight of the This quantity is called the latent heat of fusion.

body passes from the liquid to the solid state, its temperature tionary or nearly stationary during the whole operation of freez- tity of heat equal to the latent heat of fusion is produced in the jected into the atmosphere or other surrounding bodies.

ving are examples in British thermal units per pound, as given :

Substances.	Melting Points.	Latent Heat of Fusion.
according to Person)............	32	142.65
maceti...........................	56	148
wax..............................	140	175
sphorus..........................	177	9.06
hur..............................	405	16.86
....	426	500

corresponding to the pressure of the vapor: a quantity of h...
latent heat of evaporation at that temperature is produced i...
in order that the operation of condensation may go on, tha...
transferred from the body condensed to some other body.

The following are examples of the latent heat of evapor...
thermal units, of one pound of certain substances, when the...
vapor is one atmosphere of 14.7 lbs. on the square inch:

Substance.	Boiling-point under one atm. Fahr.	L...
Water	212.0	9...
Alcohol	172.2	3...
Ether	95.0	1...
Bisulphide of carbon	114.8	1...

The latent heat of evaporation of water at a series of b...
tending from a few degrees below its freezing-point up to a...
Fahrenheit has been determined experimentally by M. Re...
sults of those experiments are represented approximately...
in British thermal units per pound,

$$l \text{ nearly} = 1091.7 - 0.7(t - 32°) = 965.7 - 0.7(t - ...$$

The Total Heat of Evaporation is the sum of...
disappears in evaporating one pound of a given substance...
perature (or latent heat of evaporation) and of the heat req...
temperature, before evaporation, from some fixed temper...
temperature of evaporation. The latter part of the total h...
sensible heat.

In the case of water, the experiments of M. Regnault sho...
heat of steam from the temperature of melting ice increas...
rate as the temperature of evaporation rises. The followin...
in British thermal units per pound:

$$h = 1091.7 + 0.305(t - 32°).$$

For the total heat, latent heat, etc., of steam at differen...
table of the Properties of Saturated Steam. For tables of t...
heat, and other properties of steams of ether, alcohol, acet...
chloride of carbon, and bisulphide of carbon, see Rontgen's...
ics (Dubois's translation.) For ammonia and sulphur diox...
Thermodynamics; also, tables under Refrigerating Machine...

EVAPORATION AND DRYING.

ation of Water in Reservoirs.—Experiments at the Reservoir, Rochester, N. Y., in 1891, gave the following results:

	July.	Aug.	Sept.	Oct.
rature of air in shade...	70.5	70.3	68.7	53.3
" water in reservoir........	68.2	70.2	66.1	54.4
ity of air, per cent...	67.0	74.6	75.2	74.7
in inches during month...........	5.59	4.93	4.05	3.23
ches during month...............	3.44	2.95	1.44	2.16

tion of Water from Open Channels. (Flynn's nals and Flow of Water.)—Experiments from 1881 to 1885 in y, California, showed an evaporation from a pan in the river verage depth of one eighth of an inch per day throughout the

pan was in the air the average evaporation was less than 3/16 r day. The average for the month of August was 1/3 inch per March and April 1/12 of an inch per day. Experiments in w that evaporation ranges from .088 to .16 of an inch per day rigating season.

Italy the evaporation was from 1/12 to 1/9 inch per day, while under the influence of hot winds, it was from 1/6 to 1/5 inch

season in Northern India, with a decidedly hot wind blowing, evaporation was ½ inch per day. The evaporation increases perature of the water.

tion by the Multiple System.—A multiple effect is a porating vessels each having a steam chamber, so connected of the steam or vapor produced in the first vessel heats the apor or steam produced in the second heats the third, and so or from the last vessel is condensed in a condenser. Three nerally used, in which case the apparatus is called a *Triple* aporating in a triple effect the vacuum is graduated so that the d at a constant and low temperature.

ce to Boiling.—Brine. (Rankine.)—The presence in a stance dissolved in it (as salt in water) resists ebullition, and perature at which the liquid boils, under a given pressure; but solved substance enters into the composition of the vapor, the een the temperature and pressure of saturation of the vapor anged. A resistance to ebullition is also offered by a vessel of ich attracts the liquid (as when water boils in a glass vessel). g take place by starts. To avoid the errors which causes of luce in the measurement of boiling-points, it is advisable to mometer, not in the liquid, but in the vapor, which shows the oint, freed from the disturbing effect of the attractive nature The boiling-point of saturated brine under one atmosphere and that of weaker brine is higher than the boiling-point of y 1.2° Fahr., for each 1/32 of salt that the water contains. ater contains 1/32; and the brine in marine boilers is not suf- in more than from 2/32 to 3/32.

of Evaporation Employed in the Manufacture E. Engelhardt, Chemist Onondaga Salt Springs; Report for r heat—solar evaporation. 2. Direct fire, applied to the heat- the vessels containing brine—kettle and pan methods. 3. The system—steam-pans, steam-kettles, etc. 4. Use of steam and f the atmospheric pressure over the boiling brine—vacuum

urated salt solution boils, it is immaterial whether it is done y atmospheric pressure at 228° F., or under four atmospheres ature of 320° F., or in a vacuum under 1/10 atmosphere, the ays be a fine-grained salt.

sumption is stated to be as follows: By the kettle method, 40 t evaporated per ton of fuel, anthracite dust burned on per- ; evaporation, 5.53 lbs. of water per pound of coal. By the 0 to 75 bu. per ton of fuel. By vacuum pans, single effect, 86 f anthracite dust (2000 lbs.). With a double effect nearly ount can be produced.

Solubility of Common Salt in Pure Water. (And

Temp. of brine, F.	32	50	86	104	140
100 parts water dissolve parts	35.63	35.69	36.03	36.32	37.0
100 parts brine contain salt	26.27	26.30	26.49	26.64	27.0

According to Poggial, 100 parts of water dissolve at 229.66° F., 4 of salt, or in per cent of brine, 28.749. Gay Lussac found that at 2 100 parts of pure water would dissolve 40.38 parts of salt, in pe brine, 28.764 parts.

The solubility of salt at 229° F. is only 2.5% greater than at 32°. I cannot, as in the case of alum, separate the salt from the water by a saturated solution at the boiling point to cool to a lower temper

Solubility of Sulphate of Lime in Pure Water. (M

Temperature F. degrees.	32	64.5	89.6	100.4	105.8	127.4
Parts water to dissolve 1 part gypsum	415	386	371	368	370	375
Parts water to dissolve 1 part anhydrous $CaSO_4$	525	488	470	466	468	474

In salt brine sulphate of lime is much more soluble than in pu In the evaporation of salt brine the accumulation of sulphate of li to stop the operation, and it must be removed from the pans to av of fuel.

The average strength of brine in the New York salt districts ir 69.38 degrees of the salinometer.

Strength of Salt Brines.—The following table is conden one given in U. S. Mineral Resources for 1888, on the authori Englehardt.

Relations between Salinometer Strength, Specific G Solid Contents, etc., of Brines of Different Stren

Salinometer, degrees.	Baumé, degrees.	Specific gravity.	Per cent of salt.	Weight of a gallon of this brine in pounds.	Pounds of salt in a gallon of brine of 231 cubic inches.	Gallons of brine required for a bushel of salt.	Pounds of water to be evaporated to produce a bushel of salt.	Lbs. of coal required to produce a bushel of salt, 1 lb. coal evaporating 6 lbs. of water
1	.26	1.002	.265	8.347	.022	2,531	21,076	3,513
2	.52	1.003	.530	8.356	.044	1,264	10,510	1,752
4	1.04	1.007	1.060	8.389	.088	629.7	5,227	871.2
6	1.56	1.010	1.590	8.414	.133	418.6	3,466	577.7
8	2.08	1.014	2.120	8.447	.179	312.7	2,585	430.9
10	2.60	1.017	2 650	8.472	.224	249.4	2,057	342.9
12	3.12	1.021	3.180	8.506	.270	207.0	1,705	284.2
14	3.64	1.025	3.710	8.539	.316	176.8	1,453	242.2
16	4.16	1 028	4.240	8.564	.364	154.2	1,265	210.8
18	4.68	1.032	4.770	8.597	.410	136.5	1,118	186.3
20	5.20	1.035	5.300	8.622	.457	122.5	1,001	176.8
30	7.80	1.054	7.950	8.781	.698	80.21	648.4	108.1
40	10.40	1.073	10.600	8.939	.947	59.09	472.3	78.7
50	13.00	1.093	13.250	9.105	1.206	46.41	366.6	61.1
60	15 60	1.114	15.900	9.280	1.475	37.94	296.2	49.3
70	18.20	1.136	18.550	9.464	1.755	31.89	245.9	40.9
80	20.80	1.158	21.200	9.647	2.045	27.38	208.1	34.6
90	[illegible]	1.182	23.850	9.847	2.348	23.84	178.8	29.8
100	[illegible]	1.205	26.500	10.089	2.660	21.04	155.3	25.8

n of Sugar Solutions.* (From "Heating and Con- by Steam," by John G. Hudson; *The Engineer*, June 13, stages of the process, when the liquor is of low density, the will be high, say two to three (British) gallons per square face with 10 lbs. steam pressure, but will gradually fall to amount as the final stage is approached. As a generally gning, Mr. Hudson takes an evaporation of one gallon per re foot of gross heating surface, with steam of the pres- s.

the evaporative duty of a vacuum pan when performing of concentration, during which all the heating surface he gives the following:

AN.—4¾ in. copper coils. 528 square feet of surface; bs.; temperature in pan, 141° to 148°; density of feed, 25° entrated to 31° Beaumé.

aporation at the rate of 2000 gallons per hour = 3.8 gallons transmission, 376 units per degree of difference of tem-

Evaporation at the rate of 1503 gallons per hour = 2.8 gal- ot; transmission, 265 units per degree.

otal time needed to work up a charge of massecuite from density, the following figures, obtained by plotting the ge number of pans, form a guide to practical working. of the coil type, some with and some without jackets, surface probably averaging, and not greatly differing ot per gallon capacity, and the steam pressure 10 lbs. per h plantation and refining pans are included, making sugar:

	Density of Feed (degs. Beaumé).				
	10°	15°	20°	25°	30°
red per gallon masse-	6.123	3.6	2.26	1.5	.97
hours required per	12.	9.	6½	5.	4.
e evaporation per hour of gross surface, as- . per gallon capacity..	2.04	1.6	1.39	1.2	.97
hours required per	8.5	5.5	3.8	2.75	2.0
ge evaporation per foot..................	2.88	2.6	2.38	2.18	1.9

heating steam needed is practically the same in vacuum The advantages proper to the vacuum system are pri- l temperature of boiling, and incidentally the possibility eam of low pressure.

ugar in water, each pound of sugar adds to the volume e extent of .061 gallon at a low density to .0638 gallon at

Evaporating by Exhaust Steam is described in Trans. A. S. M. E., vol. viii. A pan 17′ 6″ × 11′ × 1′ 6″, n condensing pipes of about 250 sq. ft. of surface, evapo- er hour from clear water, condensing only about one half lied by a plain slide-valve engine of 14″ × 32″ cylinder, r min., cutting off about two thirds stroke, with steam at ure.

t keeping the pan-room warm and letting only sufficient vapor up out of a ventilator adds to its efficiency, as the re of the water in the pan was only about 165° F.

re made with coils of pipe in a small pan, first with no h one having straight blades, and lastly with troughed ative results being about the proportions of one, two, and

iquors whose boiling point is 220° F., or much above that d that exhaust steam can do but little more than bring tion strength, but on weak liquors, syrups, glues ... is ful.

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er sugar data see Bagasse as Fuel, under Fu

Drying in Vacuum.—An apparatus for drying grain ar
stances in vacuum is described by Mr. Emil Passburg in Pro.
Engrs., 1889. The three essential requirements for a success
nomical process of drying are: 1. Cheap evaporation of th
2. Quick drying at a low temperature; 3. Large capacity of th
employed.

The removal of the moisture can be effected in either of two
by slow evaporation, or by quick evaporation—that is, by boili

Slow Evaporation.—The principal idea carried into practice
acting by slow evaporation is to bring the wet substance re
contact with the inner surfaces of the apparatus, which ar
steam, while at the same time a current of hot air is also pass
the substances for carrying off the moisture. This method re
heat, because the hot-air current has to move at a considera
order to shorten the drying process as much as possible; co
great quantity of heated air passes through and escapes unuse
rier of moisture hot air cannot in practice be charged beyond
saturation; and it is in fact considered a satisfactory result
proportion be attained. A great amount of heat is here produ
not used; while, with scarcely half the cost for fuel, a much
moval of the water is obtained by heating it to the boiling poin

Quick Evaporation by Boiling.—This does not take place un
is brought up to the boiling point and kept there, namely, 2
atmospheric pressure. The vapor generated then escapes fre
are easily evaporated in this way, because by their motion co
boiling the heat is continuously conveyed from the heating surf
the liquid, but it is different with solid substances, and many n
ties have to be overcome, because convection of the heat cease
solids. The substance remains motionless, and consequent
greater quantity of heat is required than with liquids for o
same results.

Evaporation in Vacuum.—All the foregoing disadvantages a
the boiling-point of water is lowered, that is, if the evaporatic
out under vacuum.

This plan has been successfully applied in Mr. Passburg's vac
apparatus, which is designed to evaporate large quantities of
tained in solid substances.

The drying apparatus consists of a top horizontal cylinder,
by a charging vessel at one end, and a bottom horizontal cyl
discharging vessel beneath it at the same end. Both cylinders
in steam-jackets heated by exhaust steam. In the top cylinder
volving cast-iron screw with hollow blades, which is also heated
steam. The bottom cylinder contains a revolving drum of tube
of one large central tube surrounded by 24 smaller ones, all f
plates at both ends; this drum is heated by live steam direct fro
The substance to be dried is fed into the charging vessel throu
holes, and is carried along the top cylinder by the screw creeper
end, where it drops through a valve into the bottom cylinder, i
lifted by blades attached to the drum and travels forwards i
direction; from the front end of the bottom cylinder it falls int
ing vessel through another valve, having by this time become
vapor arising during the process is carried off by an air-pum
dome and air-valve on the top of the upper cylinder, and a
a throttle-valve on the top of the lower cylinder; both of thes
supplied with strainers.

As soon as the discharging vessel is filled with dried materi
connecting it with the bottom cylinder is shut, and the dried c
out without impairing the vacuum in the apparatus. When t
vessel requires replenishing, the intermediate valve between t
ders is shut, and the charging vessel filled with a fresh supply
rial; the vacuum still remains unimpaired in the bottom cylin
to be restored only in the top cylinder after the charging ves
closed again.

In this vacuum the boiling-point of the water contained in th
rial is brought down as low as 110° F. The difference between t
ture and that of the heating surfaces is amply sufficient for ob
results from the employment of exhaust steam for heating all
except the ...ing drum of tubes. The water contained in t
stance ... evaporates as soon as the latter is heated to

there is any moisture to be removed the solid substance is e this temperature.

om a brewery or distillery, containing from 75% to 78% of his drying process been converted in some localities from mbrance into a valuable food-stuff. The water is removed only, no previous mechanical pressing being resorted to. nness's brewery in Dublin two of these machines are em- of these the top cylinder is 20′ 4″ long and 2′ 8″ diam., and ng inside it makes 7 revs. per min.; the bottom cylinder is ′ 4″ diam., and the drum of the tubes inside it makes 5 revs. drying surfaces of the two cylinders amount together to a ut 1000 sq. ft., of which about 40% is heated by exhaust steam boiler. There is only one air-pump, which is made large e machines; it is horizontal, and has only one air-cylinder, acting, 17¾ in. diam. and 17¾ in. stroke; and it is driven at r min. As the result of about eight months' experience, the ave been drying the wet grains from about 500 cwt. of malt urs.

ing, 3 cwt. of malt gave 4 cwt. of wet grains, and the latter ried grains; 500 cwt. of malt will therefore yield about 670 ns, or 335 cwt. per machine. The quantity of water to be n the wet grains is from 75% to 78% of their total weight, or t. altogether, being 256 cwt. per machine.

RADIATION OF HEAT.

heat takes place between bodies at all distances apart, and for the radiation of light.

s proceed in straight lines, and the intensity of the rays by one source varies inversely as the square of their distance

t has been erroneously interpreted by some writers, who rom it that a boiler placed two feet above a fire would re- n only one fourth as much heat as if it were only one foot ase of boiler furnaces the side walls reflect those rays that n angle—following the law of optics, that the angle of inci- the angle of reflection,—with the result that the intensity above the fire is practically the same as at one foot above, ne-fourth as much.

ich a hotter body radiates heat, and a colder body absorbs on the state of the surfaces of the bodies as well as on their The rate of radiation and of absorption are increased by ughness of the surfaces of the bodies, and diminished by polish. For this reason the covering of steam pipes and smooth and of a light color; uncovered pipes and steam- should be polished.

of heat radiated by a body is also a measure of its heat- , under the same circumstances. When a polished body is of heat, it absorbs part of the heat and reflects the rest. ower of a body is therefore the complement of its absorbing ter is the same as its radiating power.

diating and reflecting power of different bodies has been xperiment, as shown in the table below, but as far as quan- e concerned, says Prof. Trowbridge (Johnson's Cyclopædia, doubtful whether anything further than the said relative can, in the present state of our knowledge, be depended or absolute quantities for different temperatures being still authorities do not even agree on the relative radiating Leslie gives for tin plate, gold, silver, and copper the figure considerably from the figures in the table below, given by be on the authority of Leslie, De La Provostaye and De- ni.

Relative Radiating and Reflecting Power of Differe[...] Substances.

	Radiating or Absorbing Power.	Reflecting Power.		Radiating or Absorbing Power.	Reflecting Power.
Lampblack	100	0	Zinc, polished	19	[illegible]
Water	100	0	Steel, polished	17	[illegible]
Carbonate of lead	100	0	Platinum, polished	24	[illegible]
Writing-paper	98	2	" in sheet	17	[illegible]
Ivory, jet, marble	93 to 98	7 to 2	Tin	15	[illegible]
Ordinary glass	90	10	Brass, cast, dead polished	11	[illegible]
Ice	85	15	Brass, bright polished	7	[illegible]
Gum lac	72	28	Copper, varnished	14	[illegible]
Silver-leaf on glass	27	73	" hammered	7	[illegible]
Cast iron, bright polished	25	75	Gold, plated	5	[illegible]
Mercury, about	23	77	" on polished steel	3	[illegible]
Wrought iron, polished	23	77	Silver, polished bright	3	[illegible]

Experiments of Dr. A. M. Mayer give the following: The relative ra[...] tions from a cube of cast iron, having faces rough, as from the found[...] planed, "drawfiled," and polished, and from the same surfaces oiled, ar[...] below (Prof. Thurston, in Trans. A. S. M. E., vol. xvi.):

Surface.	Oiled.	Dry.
Rough	100	100
Planed	60	32
Drawfiled	49	20
Polished	45	18

It here appears that the oiling of smoothly polished castings, as of cy[...] der-heads of steam-engines, more than doubles the loss of heat by radiati[...] while it does not seriously affect rough castings.

CONDUCTION AND CONVECTION OF HEAT.

Conduction is the transfer of heat between two bodies or parts o[...] body which touch each other. Internal conduction takes place between[...] parts of one continuous body, and external conduction through the surf[...] of contact of a pair of distinct bodies.

The rate at which conduction, whether internal or external, goes [...] being proportional to the area of the section or surface through which [...] takes place, may be expressed in thermal units per square foot of area [...] hour.

Internal Conduction varies with the *heat conductivity*, which [...] pends upon the nature of the substance, and is directly proportional to [...] difference between the temperatures of the two faces of a layer, and [...] versely as its thickness. The reciprocal of the conductivity is called [...] *internal thermal resistance* of the substance. If r represents this resistan[...] x the thickness of the layer in inches, T' and T the temperatures on the [...] faces, and q the quantity in thermal units transmitted per hour per squ[...]

foot of area, $q = \frac{T' - T}{rx}$. (Rankine.)

Péclet gives the following values of r:

Gold, platinum, silver	0.0016	Lead	[illegible]
Copper	0.0018	Marble	[illegible]
Iron	0.0043	Brick	[illegible]
Zinc	0.0045		

ative Heat-conducting Power of Metals.

(* Calvert & Johnson ; † Weidemann & Franz)

Silver = 1000.

.	*C. & J.	†W. & F.	Metals.	*C. & J.	†W. & F.
....	1000	1000	Cadmium............	577	....
....	981	532	Wrought iron	436	119
1% of			Tin..................	422	145
....	840	....	Steel................	397	116
d.......	845	736	Platinum	380	84
....	811	....	Sodium..............	365	....
....	677	....	Cast iron............	359	....
h 1.25%			Lead	287	85
....	412	....	Antimony:		
....	665	....	cast horizontally...	215	....
....	641	....	cast vertically. ...	192	....
		....	Bismuth..............	61	18
lly....	628				
tally..	608	....			

E OF A NON-METALLIC SUBSTANCE IN COMBINATION ON THE CONDUCTING POWER OF A METAL.

arbon on iron :		Influence of arsenic on copper :	
on..................	436	Cast copper..........	811
....	397	Copper with 1% of arsenic......	570
....	359	" with .5% of arsenic.....	669
		" with .25% of arsenic....	771

Steam-pipe Coverings.

by Prof. Ordway, Trans. A. S. M. E., v, 73; also Circular No. 27 of Boston Mfrs. Mutual Fire Ins. Co., 1890.)

bserved that several of the incombustible materials are nearly wool, cotton, and feathers, with which they may be compared ing table. The materials which may be considered wholly e danger of being carbonized or ignited by slow contact with rs are printed in Roman type. Those which are more or less rbonized are printed in italics.

TABLE I.

inch thick. Heat applied, 310° F.	Pounds of Water heated 10° F., per hour, through 1 square foot.	Solid Matter in 1 square foot 1 inch thick, parts in 1000.	Air Included, parts in 1000.
l..............................	8.1	56	944
feathers......................	9.6	50	950
tton wool.........	10.4	20	980
....	10.3	185	815
pblack........................	9.8	56	944
d lampblack.............. ..	10.6	244	756
coal........	11.9	53	947
e charcoal	13.9	119	881
e-coal powder.................	35.7	506	494
ined magnesia..................	12.4	23	977
ed calcined magnesia.........	42.6	285	715
onate of magnesia.............	13.7	60	940
ed carbonate of magnesia.....	15.4	150	850
il-meal...........................	14.5	60	940
ossil-meal.........	15.7	112	888
alk (Paris white).............	20.6	253	[illegible]
r of Paris....................... ..	30.9	368	[illegible]
tos...........	49.0	[illegible]	[illegible]
....	48.0	[illegible]	[illegible]
....	62.1	5[illegible]	[illegible]

TABLE II.

Covering.	Pounds of heated 1 per hou 1 square
21. Best slag-wool	13.
22. *Paper*	14.
23. *Blotting-paper wound tight*	21.
24. *Asbestos paper wound tight*	21.
25. *Cork strips bound on*	14.
26. *Straw rope wound spirally*	18.
27. *Loose rice chaff*	18.
28. Paste of fossil-meal with hair	16.
29. Paste of fossil-meal with asbestos	22.
30. Loose bituminous-coal ashes	21.
31. Loose anthracite-coal ashes	27.
32. Paste of clay and vegetable fibre	30.

Professor Ordway's report says: Careful experiments have been with various non-conductors, each used in a mass one inch thick, pla a flat surface of iron kept heated by steam to 310° Fahr. Table I giv amount of heat transmitted per hour through each kind of non-con one inch thick, reckoned in pounds of water heated 10° Fahr., the unit being one square foot of covering.

The substances given in Table II were actually tried as coverin two-inch steam-pipe, but for convenience of comparison the result been reduced by calculation to the same terms as in Table I.

Later experiments have given results for still air which differ littl those of Nos. 3, 4, and 6. In fact the bulk of matter in the best non-c tors is relatively too small to have any specific effect, except to entr air and keep it stagnant. These substances keep the air still by vir the roughness of their fibres or particles. The asbestos, No. 18, had fibres, which could not prevent the air from moving about.

Later trials with an asbestos of exceedingly fine fibre have made a what better showing, but asbestos is really one of the poorest non-c tors. By reason of its fibrous character it may be used advantag to hold together other incombustible substances, but the less the We have made trials of two samples of a "magnesia covering," con of carbonate of magnesia with a small percentage of good asbestos One transmitted heat which, reduced to the terms of Table I, would a to 15 lbs.: the denser one gave 20 lbs. The former contained of solid matter; the latter 396/1000.

Any suitable substance which is used to prevent the escape of heat should not be less than one inch thick.

Any covering should be kept perfectly dry, for not only is water a carrier of heat, but it has been found that still water conducts heat eight times as rapidly as still air.

Heat-conducting Power of Covering Materials.

(J. J. Coleman, *Eng'g*, Sept. 5, 1884, p. 237.)

Experiments were made by filling a 10-in. cube with ice, surround with the different materials to be tested, and noting the quantity melted per hour with each insulator.

The relative results were as follows:

Silicate cotton (mineral wool)	100	Charcoal	
Hair felt	117	Sawdust	
Cotton wool	122	Gas-works breeze	
Sheep's wool	136	Wood and air-space	
Infusorial earth	136		

The Rate of External Conduction through the bound face between a solid body and a fluid is approximately proportional difference of temperature, when that is small; but when that differ nsiderable the rate of conduction increases faster than the simpl hat difference. (Rankine.)

If r, as before, is the coefficient of internal thermal resistance, e and e' the coefficient of external resistance of the two surfaces, x the thickness of the plate, and T' and T the temperatures of the two fluids in contact with the two surfaces, the total thermal resistance is $q = \frac{T' - T}{e + e' + rx}$. According to Peclet, $e + e' = \frac{1}{A[1 + B(T' - T)]}$, in which the constants A and B have the following values:

B for polished metallic surfaces	.0028
B for rough metallic surfaces and for non-metallic surfaces	.0037
A for polished metals, about	.90
A for glassy and varnished surfaces	1.34
A for dull metallic surfaces	1.58
A for lamp-black	1.78

When a metal plate has a liquid at each side of it, it appears from experiments by Peclet that $B = .058$, $A = 8.8$.

The results of experiments on the evaporative power of boilers agree very well with the following approximate formula for the thermal resistance of boiler plates and tubes:

$$e + e' = \frac{a}{(T' - T)},$$

which gives for the rate of conduction, per square foot of surface per hour,

$$q = \frac{(T' - T)^2}{a}.$$

This formula is proposed by Rankine as a rough approximation, near enough to the truth for its purpose. The value of a lies between 160 and 200.

Convection, or carrying of heat, means the transfer and diffusion of the heat in a fluid mass by means of the motion of the particles of that mass.

The conduction, properly so called, of heat through a stagnant mass of fluid is very slow in liquids, and almost, if not wholly, inappreciable in gases. It is only by the continual circulation and mixture of the particles of the fluid that uniformity of temperature can be maintained in the fluid mass, or heat transferred between the fluid mass and a solid body.

The free circulation of each of the fluids which touch the side of a solid plate is a necessary condition of the correctness of Rankine's formulæ for the conduction of heat through that plate; and in these formulæ it is implied that the circulation of each of the fluids by currents and eddies is such as to prevent any considerable difference of temperature between the fluid particles in contact with one side of the solid plate and those at considerable distances from it.

When heat is to be transferred by convection from one fluid to another, through an intervening layer of metal, the motions of the two fluid masses should, if possible, be in opposite directions, in order that the hottest particles of each fluid may be in communication with the hottest particles of the other, and that the minimum difference of temperature between the adjacent particles of the two fluids may be the greatest possible.

Thus, in the surface condensation of steam, by passing it through metal tubes immersed in a current of cold water or air, the cooling fluid should be made to move in the opposite direction to the condensing steam.

Transmission of Heat, through Solid Plates, from Water to Water. (Clark, S.E.).—M. Péclet found, from experiments made with plates of wrought iron, cast iron, copper, lead, zinc, and tin, that when the fluid in contact with the surface of the plate was not circulated by artificial means, the rate of conduction was the same for different metals and for plates of the same metal of different thicknesses. But when the water was thoroughly circulated over the surfaces, and when these were perfectly clean, the quantity of transmitted heat was inversely proportional to the thickness, and directly as the difference in temperature of the two faces of the plate. When the metal surface became dull, the rate of transmission of heat through all the metals was very nearly the same.

It follows, says Clark, that the absorption of heat through metal plates is more active whilst evaporation is in progress—when the circulation of the water is more active—than while the water is being heated up to the boiling point.

Transmission from Steam to Water.—M. Péclet's prin supported by the results of experiments made in 1867 by Mr. Isherw the conductivity of different metals. Cylindrical pots, 10 inches in di 21¼ inches deep inside, and ⅛ inch, ¼ inch, and ⅜ inch thick, tur bored, were formed of pure copper, brass (60 copper and 40 zinc wrought iron, and remelted cast iron. They were immersed in bath, which was varied from 220° to 320° F. Water at 212° was sup the pots, which were kept filled. It was ascertained that the rate of tion was in the direct ratio of the difference of the temperatures in outside of the pots; that is, that the rate of evaporation per d difference of temperatures was the same for all temperatures; and rate of evaporation was exactly the same for different thickness metal. The respective rates of conductivity of the several metals follows, expressed in weight of water evaporated from and at 21 square foot of the interior surface of the pots per degree of diff temperature per hour, together with the equivalent quantities of h

	Water at 212°.	Heat-units.	Ra
Copper......................	.665 lb.	642.5	1
Brass........................	.577 "	556.8	
Wrought iron........... ...	.387 "	373.6	
Cast iron..................	.327 "	315.7	

Whitham, "Steam Engine Design," p. 283, also Trans. A. S. M. E. using these data in deriving a formula for surface condensers c figures those of perfect conductivity, and multiplies them by a c C, which he takes at 0.323, to obtain the efficiency of condenser i ordinary use, i.e., coated with saline and greasy deposits.

Transmission of Heat from Steam to Water t Coils of Iron Pipe.—H. G. C. Kopp and F. J. Meystre (*Stea cator*, Jan., 1894), give an account of some experiments on transm heat through coils of pipe. They collate the results of earlier exp as follows, for comparison:

Experimenter.	Character of Surface.	Steam Condensed per Square foot per degree difference of temperature per hour. Heating, pounds.	Evaporating, pounds.	Heat transmitted per square foot per degree difference of temperature per hour. Heating, B. T. U.	Evaporating B. T. U.	Rem
Laurens	Copper coils...	.292	.981	315	974	
"	2 Copper coils.	....	1.20	....	1120	
Havrez..	Copper coil....	.268	1.26	280	1200	
Perkins.	Iron coil......	....	.24	....	215	Steam = 10
"	" "	....	.22	....	208.2	Steam = 10.
Box.....	Iron tube	.235	......	230	......	
"	" "	.196	......	207	......	
"	" "	.206	......	210	......	
Havrez..	Cast-iron boiler	.077	.105	82	100	

From the above it would appear that the efficiency of iron surfa than that of copper coils, plate surfaces being far inferior.

In all experiments made up to the present time, it appears that ...rature of the condensing water was allowed to rise, a mean be ...al and final temperatures being accepted as the effective tem as ... mes warmer it circulates more rapidly, thereb ...g the coil to become agitated and replaced ...ore heat to be transmitted.

Again, in accepting the mean temperature as that of the condensing medium, the assumption is made that the rate of condensation is in direct proportion to the temperature of the condensing water.

In order to correct and avoid any error arising from these assumptions and approximations, experiments were undertaken, in which all the conditions were constant during each test.

The pressure was maintained uniform throughout the coil, and provision was made for the free outflow of the condensed steam, in order to obtain at all times the full efficiency of the condensing surface. The condensing water was continually stirred to secure uniformity of temperature, which was regulated by means of a steam-pipe and a cold-water pipe entering the tank in which the coil was placed.

The following is a condensed statement of the results

HEAT TRANSMITTED PER SQUARE FOOT OF COOLING SURFACE, PER DEGREE OF DIFFERENCE OF TEMPERATURE. (British Thermal Units.)

Temperature of Condensing Water.	1-in. Iron Pipe; Steam inside, 60 lbs. Gauge Pressure.	1½ in. Pipe; Steam inside, 10 lbs. Pressure.	1½ in. Pipe; Steam inside, 10 lbs. Pressure.	1½ in. Pipe; Steam inside, 60 lbs. Pressure.
80	265	128	200	....
100	269	130	230	239
120	272	137	260	247
140	277	145	267	276
160	281	158	271	306
180	299	174	270	349
200	313	...	....	419

The results indicate that the heat transmitted per degree of difference of temperature in general increases as the temperature of the condensing water is increased.

The amount transmitted is much larger with the steam on the outside of the coil than with the steam inside the coil. This may be explained in part by the fact that the condensing water when inside the coil flows over the surface of conduction very rapidly, and is more efficient for cooling than when contained in a tank outside of the coil.

This result is in accordance with that found by Mr. Thomas Craddock, which indicated that the rate of cooling by transmission of heat through metallic surfaces was almost wholly dependent on the rate of circulation of the cooling medium over the surface to be cooled.

Transmission of Heat in Condenser Tubes. (*Eng'g*, Dec. 10, 1875, p. 449.).—In 1874 B. C. Nichol made experiments for determining the rate at which heat was transmitted through a condenser tube. The results went to show that the amount of heat transmitted through the walls of the tube per estimated degree of mean difference of temperature increased considerably with this difference. For example:

	Vertical Tube.			Horizontal Tube		
Estimated mean difference of temperature between inside and outside of tube, degrees Fahr. .	128	151.9	152.9	111.6	146.2	150.4
Heat-units transmitted per hour per square foot of surface per degree of mean diff. of temp....	422	531	561	610	737	823

These results seem to throw doubt upon Mr. Isherwood's statement that the rate of evaporation per degree of difference of temperature is the same for all temperatures.

Mr. Thomas Craddock found that water was enormously more efficient than air for the abstraction of heat through metallic surfaces in the process of cooling. He proved that the rate of cooling by transmission of heat through metallic surfaces depends upon the rate of circulation of the cooling medium over the surface to be cooled. A tube filled with hot water, moved by rapid rotation at the rate of 59 ft. per second, through air, lost as much heat in one minute as it did in still air in 12 minutes. In water, at a velocity of 3 ft. per second, as much heat was abstracted in half a minute as was abstracted in one minute when it was at rest in the water. Mr. Craddock concluded, further, that the circulation of the cooling fluid became greater importance as the difference of temperature on the two sid plate became less. (Clark, R. T. D., p. 461.)

Heat Transmission through Cast-iron Plates Pic Nitric Acid.—Experiments by R. C. Carpenter (Trans. A. S. 179) show a marked change in the conducting power of the pla steam to water), due to prolonged treatment with dilute nitric acid

The action of the nitric acid, by dissolving the free iron and not the carbon, forms a protecting surface to the iron, which is larg posed of carbon. The following is a summary of results:

Character of Plates, each plate 8.4 in. by 5.4 in., exposed surface 27 sq. ft.	Increase in Temperature of 3.125 lbs. of Water each Minute.	Proportionate Thermal Units Transmitted fo each Degree of Difference of Temperature pe Square Foot pe Hour.
Cast iron—untreated skin on, but clean, free from rust.	13.90	113.2
Cast iron—nitric acid, 1% sol., 9 days	11.5	97.7
" " 1% sol., 18 days.	9.7	80.08
" " 1% sol., 40 days.	9.6	77.8
" " 5% sol., 9 days	9.93	87.0
" " 5% sol., 40 days.	10.6	77.4
Plate of pine wood, same dimensions as the plate of cast iron	0.33	1.9

The effect of covering cast-iron surfaces with varnish has bee gated by P. M. Chamberlain. He subjected the plate to the action acid for a few hours, and then applied a non-conducting varnish. face only was treated. Some of his results are as follows:

Heat units per sq. ft. per hour, for each degree, $T - T'$.

- 170. As finished—greasy.
- 152. " " washed with benzine and dried.
- 169. Oiled with lubricating oil.
- 162. After exposure to nitric acid sixteen hours, then seed oil.)
- 166 After exposure to hydrochloric acid twelve hours, (linseed oil.)
- 113. } After exposure to sulphuric acid 1, water 2, for
- 117. } then oiled, varnished, and allowed to dry for 24

Transmission of Heat through Solid Plates fr or other Dry Gases to Water. (From Clark on the Stean —The law of the transmission of heat from hot air or other gases through metallic plates, has not been exactly determined by ex The general results of experiments on the evaporative action of portions of the heating surface of a steam-boiler point to the ge that the quantity of heat transmitted per degree difference of ten is practically uniform for various differences of temperature.

The communication of heat from the gas to the plate surface accelerated by mechanical impingement of the gaseous products surface.

Clark says that when the surfaces are perfectly clean, the rate mission of heat through plates of metal from air or gas to water for copper, next for brass, and next for wrought iron. But whe faces are dimmed or coated, the rate is the same for the different

With respect to the influence of the conductivity of metals a thickness of the plate on the transmission of heat from burn water, Mr. Napier made experiments with small boilers of iron a placed over a gas-flame. The vessels were 5 inches in diamete inches deep. From three vessels, one of iron, one of copper, and o sides and copper bottom, each of them 1/30 inch in thickness, equ 'es of w aporated to dryness, in the times as follows:

Iron Vessel.	Copper Vessel.	Iron and Copper Vessel.
19 minutes	18.5 minutes	......
33 "	30.75 "	......
50 "	44 "	......
35.7 "	.....	36.83 minutes.

essels of iron sides 1/30 inch thick, one having a ¼-inch copper he other a ¼-inch lead bottom, were tested against the iron essel, 1/30 inch thick. Equal quantities of water were evapo- , and 53½ minutes respectively. Taken generally, the results riments show that there are practically but slight differences copper, and lead in evaporative activity, and that the activity by the thickness of the bottom.

hnson formed a like conclusion from the results of his obser- o boilers of 160 horse-power each, made exactly alike, ex- had iron flue-tubes and the other copper flue-tubes. No dif- be detected between the performances of these boilers.

between the results of different experimenters are attribut- to the difference of conditions under which the heat was s between water or steam and water, and between gaseous ater. On one point the divergence is extreme: the rate of f heat per degree of difference of temperature. Whilst from s of heat are transmitted from water to water through iron gree of difference per square foot per hour, the quantity of ted between water and air, or other dry gas, is only about ts, according as the surrounding air is at rest or in movement. e boiler, where radiant heat was brought into play, 17 units ransmitted through the plates of the fire-box per degree of emperature per square foot per hour.

sion of Heat through Plates and Tubes from lot Water to Air.—The transfer of heat from steam or a plate or tube into the surrounding air is a complex opera- the internal and external conductivity of the metal, the radi- f the surface, and the convection of heat in the surrounding cerned. Since the quantity of heat radiated from a surface condition of the surface and with the surroundings, according t determined, and since the heat carried away by convection e rate of the flow of the air over the surface, it is evident that can be laid down for the total quantity of heat emitted.

g is condensed from an article on Loss of Heat from Steam- *ocomotive*, Sept. and Oct., 1892.

pipe is radiating heat constantly off into space, but at the cooling also by convection. Experimental data on which to ns of the heat radiated and otherwise lost by steam-pipes are ous nor satisfactory.

ctical Treatise on Heat a number of results are given for the at radiated by different substances when the temperature of ahr. lower than the temperature of the radiating body. A s table is given below. It is said to be based on Péclet's ex-

RADIATED PER HOUR, PER SQUARE FOOT OF SURFACE, FOR 1° FAHRENHEIT EXCESS IN TEMPERATURE.

ned	.0327	Sheet-iron, ordinary	.5662
	.0440	Glass	.5948
, polished	.0491	Cast iron, new	.6480
olished	.0858	Common steam-pipe, inferred	.6400
lished	.0920	Cast and sheet iron, rusted	.6868
	.1329	Wood, building stone, and brick	.7358

mperature of the air is about 50° or 60° Fahr., and the radiat- t more than about 30° hotter than the air, we may calculate of a given surface by assuming the amount of heat given off time to be proportional *to the difference in temperature be- ating body and the air.* This is "Newton's law of cooling." difference in temperature is great, Newton's law does not hold ation is no longer proportional to the difference in tempera- be calculated *by a complex formula* established experiment. *and Petit. Box has* computed a table from this for *acilitates its application,* and which is given below :

FACTORS FOR REDUCTION TO DULONG'S LAW OF RADIATIO

Differences in Temperature between Radiating Body and the Air.	Temperature of the Air on the Fahrenhei									
	32°	50°	59°	68°	86°	104°	122°	140°	158°	176
Deg. Fahr.										
18	1.00	1.07	1.12	1.16	1.25	1.36	1.47	1.58	1.70	1.8
36	1.03	1.08	1.16	1.21	1.30	1.40	1.52	1.68	1.76	1.9
54	1.07	1.16	1.20	1.25	1.35	1.45	1.58	1.70	1.83	1.9
72	1.12	1.20	1.25	1.30	1.40	1.52	1.64	1.76	1.90	2.0
90	1.16	1.25	1.31	1.36	1.46	1.58	1.71	1.84	1.98	2.1
108	1.21	1.31	1.36	1.42	1.52	1.65	1.78	1.92	2.07	2.2
126	1.26	1.36	1.42	1.48	1.50	1.72	1.86	2.00	2.16	2.3
144	1.33	1.42	1.48	1.54	1.65	1.79	1.94	2.08	2.24	2.4
162	1.37	1.48	1.54	1.60	1.73	1.86	2.02	2.17	2.34	2.5
180	1.44	1.55	1.61	1.68	1.81	1.95	2.11	2.27	2.46	2.6
198	1.50	1.62	1.69	1.75	1.89	2.04	2.21	2.38	2.56	2.7
216	1.58	1.69	1.76	1.83	1.97	2.13	2.32	2.48	2.68	2.9
234	1.64	1.77	1.84	1.90	2.06	2.28	2.43	2.52	2.80	3.0
252	1.71	1.85	1.92	2.00	2.15	2.33	2.52	2.71	2.92	3.1
270	1.79	1.93	2.01	2.09	2.22	2.44	2.64	2.84	3.06	3.3
288	1.89	2.03	2.12	2.20	2.37	2.56	2.78	2.99	3.22	3.5
306	1.98	2.13	2.22	2.31	2.49	2.69	2.90	3.12	3.37	3.6
324	2.07	2.23	2.33	2.42	2.62	2.81	3.04	3.28	3.53	3.8
342	2.17	2.34	2.44	2.54	2.73	2.95	3.19	3.44	3.70	4.0
360	2.27	2.45	2.56	2.66	2.86	3.09	3.35	3.60	3.88	4.2
378	2.39	2.57	2.68	2.79	3.00	3.24	3.51	3.78	4.08	4.4
396	2.50	2.70	2.81	2.93	3.15	3.40	3.68	3.97	4.28	4.6
414	2.63	2.84	2.95	3.07	3.31	3.51	3.87	4.12	4.48	4.8
432	2.76	2.98	3.10	3.23	3.47	3.76	4.10	4.32	4.61	5.1

The loss of heat by *convection* appears to be independent of the the surface, that is, it is the same for iron, stone, wood, and other It is different for bodies of different shape, however, and it varie position of the body. Thus a vertical steam-pipe will not lose so by convection as a horizontal one will; for the air heated at the of the vertical pipe will rise along the surface of the pipe, prote some extent from the chilling action of the surrounding cooler a similar reason the shape of a body has an important influence on those bodies losing most heat whose forms are such as to allow t free access to every part of their surface. The following table gives the number of heat units that horizontal cylinders or pi convection per square foot of surface per hour, for one degree di temperature between the pipe and the air.

HEAT UNITS LOST BY CONVECTION FROM HORIZONTAL PIPES, PE FOOT OF SURFACE PER HOUR, FOR A TEMPERATURE DIFFERENCE OF 1° FAHR.

External Diameter of Pipe in inches.	Heat Units Lost.	External Diameter of Pipe in inches.	Heat Units Lost.	External Diameter of Pipe in inches.	H
2	0.728	7	0.509	18	
3	0.626	8	0.498	24	
4	0.574	9	0.489	36	
5	0.544	10	0.482	48	
6	0.523	12	0.472	..	

onvection is nearly proportional to the di the hot body and the air; but the exp and Péclet show that this is not exactly true, and we may here also a table of factors for correcting the results obtained by simple on.

FACTORS FOR REDUCTION TO DULONG'S LAW OF CONVECTION.

nce mp. Hot and	Factor.	Difference in Temp. between Hot Body and Air.	Factor.	Difference in Temp. between Hot Body and Air.	Factor.
F.	0.94	180° F.	1.62	342° F.	1.87
	1.11	198°	1.65	360°	1.90
	1.22	216°	1.68	378°	1.92
	1.30	234°	1.72	396°	1.94
	1.37	252°	1.74	414°	1.96
	1.43	270°	1.77	432°	1.98
	1.49	288°	1.80	450°	2.00
	1.53	306°	1.83	468°	2.02
	1.58	324°	1.85	....	....

LE IN THE USE OF THE TABLES.—Required the total loss of heat by iation and convection, per foot of length of a steam-pipe 2 11/32 nal diameter, steam pressure 60 lbs., temperature of the air in the Fahr.

rature corresponding to 60 lbs. equals 307°; temperature difference 8 = 239°.

f one foot length of steam-pipe = 2 11/32 × 3.1416 ÷ 12 = 0.614 sq.

adiated per hour per square foot per degree of difference, from 4.

ion loss per hour by Newton's law = 239° × .614 ft. × .64 = 93.9 s. Same reduced to conform with Dulong's law of radiation: factor le for temperature difference of 239° and temperature of air 68° = 9 × 1.93 = 181.2 heat units, total loss by radiation.

tion loss per square foot per hour from a 2 11/32-inch pipe: by in- on from table, 2″ = .728. 3″ = .626, 2 11/32″ = .693.

614 × .693 × 239° = 101.7 heat units. Same reduced to conform with law of convection: 101.7 × 1.73 (from table) = 175.9 heat units per otal loss by radiation and convection = 181.2 + 175.9 = 357.1 heat r hour. Loss per degree of difference of temperature per linear ipe per hour = 357.1 ÷ 239 = 1.494 heat units = 2.433 per sq. ft.

ot claimed, says *The Locomotive*, that the results obtained by this of calculation are strictly accurate. The experimental data are not t to allow us to compute the heat-loss from steam-pipes with any gree of refinement; yet it is believed that the results obtained as d above will be sufficiently near the truth for most purposes. An ent by Prof. Ordway, in a pipe 2 11/32 in. diam. under the above ns (Trans. A. S. M. E., v. 73), showed a condensation of steam of 181 s per hour, which is equivalent to a loss of heat of 358.7 heat units , or within half of one per cent of that given by the above calcula-

ling to different authorities, the quantity of heat given off by steam water radiators in ordinary practice of heating of buildings by diation varies from 1.8 to about 3 heat units per hour per square degree of difference of temperature.

west figure is calculated from the following statement by Rob n his paper on "American Practice in Warming Buildings (Proc. Inst. C. E., 1882, vol. lxxi): "Each 100 sq. ft. of radiat will give off 3 Fahr. heat units per minute for each degree F. of n temperature between the radiating surface and the air in wh sed."

ure 2 1/2 heat units is given by the Nason Manufactu[illegible] atalogue, and 2 to 2 1/4 are given by many recent ordinary temperature difference in low-pressur – 70° = 142° F., 1 lb. steam condensed from 21 same temperature gives up 965.7 heat units. A loss of 2 heat units per ft. per hour per degree of difference, under these conditions, is equival to $2 \times 142 \div 965 = 0.3$ lbs. of steam condensed per hour per sq. ft. of heat surface. (See also Heating and Ventilation.)

Transmission of Heat through Walls, etc., of Buildi (Nason Manufacturing Co.). (See also Heating and Ventilation.)—H has the remarkable property of passing through moderate thicknesses of and gases without appreciable loss, so that air is not warmed by radi heat, but by contact with surfaces that have absorbed the radiation.

POWERS OF DIFFERENT SUBSTANCES FOR TRANSMITTING HEAT.

Window-glass	1000	Bricks, rough	200 to
Oak or walnut	66	Bricks, whitewashed	
White pine	80	Granite or slate	
Pitch-pine	100	Sheet iron	1030 to 1
Lath or plaster	75 to 100		

A square foot of glass will cool 1.279 cubic feet of air from the tempe ture inside to that outside per minute, and outside wall surface is gener estimated at one fifth of the rate of glass in cooling effect.

Box, in his "Practical Treatise on Heat," gives a table of the conduct powers of materials prepared from the experiments of Péclet. It gives quantity of heat in units transmitted per square foot per hour by a plat inch in thickness, the two surfaces differing in temperature 1 degree:

Fine-grained gray marble	28.00
Coarse-grained white marble	22.4
Stone, calcareous, fine	16.7
Stone, calcareous, ordinary	13.68
Baked clay, brickwork	4.83
Brick-dust, sifted	1.33

Hood, in his "Warming and Ventilating of Buildings," p. 249, gives results of M. Depretz, which, placing the conducting power of marble at 1 give .483 as the value for firebrick.

THERMODYNAMICS.

Thermodynamics, the science of heat considered as a form energy, is useful in advanced studies of the theory of steam, gas, and engines, refrigerating machines, compressed air, etc. The method of tre ment adopted by the standard writers is severely mathematical, involv constant application of the calculus. The student will find the subje thorougly treated in the recent works by Rontgen (Dubois's translatio Wood, and Peabody.

First Law of Thermodynamics.—Heat and mechanical ener are mutually convertible in the ratio of about 778 foot-pounds for the Brit thermal unit. (Wood.) Heat is the living force or *vis viva* due to cert molecular motions of the molecules of bodies, and this living force may stated or measured in units of heat or in foot-pounds, a unit of heat British measures being equivalent to 772 [778] foot-pounds. (Trowbrid Trans. A. S. M. E., vii. 727.)

Second Law of Thermodynamics.—The second law has by d ferent writers been stated in a variety of ways, and apparently with ide so diverse as not to cover a common principle. (Wood, Therm., p. 389.)

It is impossible for a self-acting machine, unaided by any external agenc to convert heat from one body to another at a higher temperature. (Cla sius.)

If all the heat absorbed be at one temperature, and that rejected be one lower temperature, then will the heat which is transmuted into work to the entire heat absorbed in the same ratio as the difference between absolute temperature of the source and refrigerator is to the absolute tem perature of the source. In other words, the second law is an expression the efficiency of the perfect elementary engine. (Wood.)

The living force, or *vis viva*, of a body (called heat) is always proportion to the absolute temperature of the body. (Trowbridge.)

The expression $\frac{Q_1 - Q_2}{Q_1} = \frac{T_1 - T_2}{T_1}$ may be called the symbolical or al *gebraic enunciation* of the second law,—the law which limits the efficien *of heat* ... which does not depend on the nature of the work *medium* ... (Trowbridge.) Q_1 and T_1 = quantity and ...

of the heat received, Q_2 and T_2 = quantity and absolute tem- the heat rejected.

ssion $\frac{T_1 - T_2}{T_1}$ represents the efficiency of a perfect heat engine ves all its heat at the absolute temperature T_1, and rejects heat erature T_2, converting into work the difference between the eived and rejected.

—What is the efficiency of a perfect heat engine which receives ° F. (the temperature of steam of 200 lbs. gauge pressure) and at 100° F. (temperature of a condenser, pressure 1 lb. above

$$\frac{388 + 459.2 - (100 + 459.2)}{388 + 459.2} = 34\%, \text{ nearly.}$$

al engine this efficiency can never be attained, for the difference e quantity of heat received into the cylinder and that rejected denser is not all converted into work, much of it being lost by eakage, etc. In the steam engine the phenomenon of cylinder on also tends to reduce the efficiency.

PHYSICAL PROPERTIES OF GASES.

al matter on this subject will be found under Heat, Air, Gas, and

nass of gas is enclosed in a vessel it exerts a pressure against the s pressure is uniform on every square inch of the surface of the , at any point in the fluid mass the pressure is the same in every

vessels containining gases the increase of pressure due to weight glected, since all gases are very light; but where liquids are con- increase in pressure due to their weight must always be taken nt.

sion of Gases, Marriotte's Law.—The volume of a gas in the same ratio as the pressure upon it is increased.

is by experiment found to be very nearly true for all gases, and s Boyle's or Mariotte's law.

ressure at a volume v, and p_1 = pressure at a volume v_1, $p_1v_1 =$ $\frac{v}{v_1}p$; pv = a constant.

stant, C, varies with the temperature, everything else remaining

pressed by a pressure of seventy-five atmospheres has a volume ess than that computed from Boyle's law, but this is the greatest that is found below 160 atmospheres pressure.

f Charles.—The volume of a perfect gas at a constant pressure ional to its absolute temperature. If v_0 be the volume of a gas and v_1 the volume at any other temperature, t_1, then

$$v_1 = v_0\left(\frac{t_1 + 459.2}{491.2}\right); \quad v_1 = \left(1 + \frac{t_1 - 32°}{491.2}\right)v_0,$$

$$\text{or} \quad v_1 = [1 + 0.002036(t_1 - 32°)]v_0.$$

ressure also change from p_0 to p_1,

$$v_1 = v_0\frac{p_0}{p_1}\left(\frac{t_1 + 459.2}{491.2}\right).$$

ensities of Gases and Vapors are simply proportional to lc weights.

dro's Law.—Equal volumes of all gases, under the same con- temperature and pressure, contain the same number of mole-

the weight of a gas in pounds per cubic foot at 32° F., multiply olecular weight of the gas by .00559. Thus 1 cu. ft. marsh-gas, CH_4,

$$= \frac{12 + 4}{2} \times .00559 = .0447 \text{ lb.}$$

When a certain volume of hydrogen combines with one half its vo oxygen, there is produced an amount of water vapor which will occ same volume as that which was occupied by the hydrogen gas whe same temperature and pressure.

Saturation-point of Vapors.—A vapor that is not near th tion-point behaves like a gas under changes of temperature and p but if it is sufficiently compressed or cooled, it reaches a point whe gins to condense: it then no longer obeys the same laws as a gas pressure cannot be increased by diminishing the size of the vessel co it, but remains constant, except when the temperature is change only gas that can prevent a liquid evaporating seems to be its own v

Dalton's Law of Gaseous Pressures.—Every portion of of gas inclosed in a vessel contributes to the pressure against the the vessel the same amount that it would have exerted by itself other gas been present.

Mixtures of Vapors and Gases.—The pressure exerted the interior of a vessel by a given quantity of a perfect gas enclos is the sum of the pressures which any number of parts into which su tity might be divided would exert separately, if each were enclo vessel of the same bulk alone, at the same temperature. Although is not exactly true for any actual gas, it is very nearly true for many if 0.080728 lb. of air at 32° F., being enclosed in a vessel of one cu capacity, exerts a pressure of one atmosphere or 14.7 pounds, on eac inch of the interior of the vessel, then will each additional 0.080728 l which is enclosed, at 32°, in the same vessel, produce very nearly tional atmosphere of pressure. The same law is applicable to mix gases of different kinds. For example, 0,12344 lb. of carbonic-acid 32°, being enclosed in a vessel of one cubic foot in capacity, exerts a p of one atmosphere; consequently, if 0.080728 lb. of air and 0.123 carbonic acid, mixed, be enclosed at the temperature of 32°, in a v one cubic foot of capacity, the mixture will exert a pressure of tw pheres. As a second example: Let 0.080728 lb. of air, at 212°, be enc a vessel of one cubic foot; it will exert a pressure of

$$\frac{212 + 459.2}{32 + 459.2} = 1.366 \text{ atmospheres.}$$

Let 0.03797 lb. of steam, at 212°, be enclosed in a vessel of one cubic will exert a pressure of one atmosphere. Consequently, if 0.080728 l and 0.03797 lb. of steam be mixed and enclosed together, at 212°, in a v one cubic foot, the mixture will exert a pressure of 2.366 atmosphere a common but erroneous practice, in elementary books on physics scribe this law as constituting a difference between mixed and homo gases; whereas it is obvious that for mixed and homogeneous gases of pressure is exactly the same, viz., that the pressure of the wh gaseous mass is the sum of the pressures of all its parts This is on laws of mixture of gases and vapors.

A second law is that the presence of a foreign gaseous substance tact with the surface of a solid or liquid does not affect the density vapor of that solid or liquid unless there is a tendency to chemic bination between the two substances, in which case the density vapor is slightly increased. (Rankine, S. E., p. 239.)

Flow of Gases.—By the principle of the conservation of energy be shown that the velocity with which a gas under pressure will esc a vacuum is inversely proportional to the square root of its density; oxygen, which is sixteen times as heavy as hydrogen, would, under the same circumstances, escape through an opening only one fourth as the latter gas.

Absorption of Gases by Liquids.—Many gases are rea sorbed by water. Other liquids also possess this power in a greater degree. Water will for example, absorb its own volume of carbo gas, 430 times its volume of ammonia, 2½ times its volume of chlori only about 1/20 of its volume of oxygen.

The weight of gas that is absorbed by a given volume of liquid is tional to the pressure. But as the volume of a mass of gas is less ure is greater, the volume which a given amount of liquid can tain temperature will be constant, whatever the pressure. mple, can absorb its own volume of carbonic-acid gas at atm it will also dissolve its own volume if the pressure is in that case the gas will be twice as dense, and consequent ht of gas is dissolved.

AIR.

s of Air.—Air is a mechanical mixture of the gases oxygen
0.7 parts O and 79.3 parts N by volume, 23 parts O and 77 parts

f pure air at 32° F. and a barometric pressure of 29.92 inches
14.6963 lbs. per sq. in., or 2116.3 lbs. per sq. ft., is .080728 lb. per
lume of 1 lb. = 12.387 cu. ft. At any other temperature and
essure its weight in lbs. per cubic foot is $W = \frac{1.3253 \times B}{459.2 + T}$,
ght of the barometer, T = temperature Fahr., and 1.3253 =
f 459.2 c. ft. of air at 0° F. and one inch barometric pressure.
491.2 of its volume at 32° F. for every increase of 1° F., and
es inversely as the pressure.

, Density, and Pressure of Air at Various Temperatures. (D. K. Clark.)

ume at Atmos. Pressure.		Density, lbs. per Cubic Foot at Atmos. Pressure.	Pressure at Constant Volume.	
Feet lb.	Compara-tive Vol.		Lbs. per Sq. In.	Compara-tive Pres.
583	.881	.086331	12.96	.881
387	.943	.080728	13.86	.943
586	.958	.079439	14.08	.958
340	.977	.077884	14.36	.977
141	1.000	.076097	14.70	1.000
342	1.015	.074950	14.92	1.015
593	1.034	.073565	15.21	1.034
345	1.054	.072230	15.49	1.054
096	1.073	.070942	15.77	1.073
344	1.092	.069721	16.05	1.092
592	1.111	.068500	16.33	1.111
346	1.130	.067361	16.61	1.130
100	1.149	.066221	16.89	1.149
351	1.168	.065155	17.19	1.168
603	1.187	.064088	17.50	1.187
854	1.206	.063089	17.76	1.206
106	1.226	.062090	18.02	1.226
606	1.264	.060210	18.58	1.264
860	1.283	.059313	18.86	1.283
910	1.287	.059135	18.92	1.287

nanometer consists of a long vertical glass tube, closed at
open at the lower end, containing air, provided with a scale,
along with a thermometer, in a transparent liquid, such as
ntained in a strong cylinder of glass, which communicates
in which the pressure is to be ascertained. The scale shows
cupied by the air in the tube.
t volume, at the temperature of 32° Fahrenheit, and mean
atmosphere, p_0; let v_1 be the volume of the air at the tem-
l under the absolute pressure to be measured p_1; then

$$p_1 = \frac{(t + 459.2°)p_0 v_0}{491.2°\ v_1}.$$

of the Atmosphere at Different A... ...es.

vel the pressure of the air is 14.7 pounds p... ... at
bove *the sea-level it is 14.02* pounds; at ... ...
1 mile, 12.02; at 1¼ mile, 11.42; at 1½ ...

miles, 9.80 pounds per square inch. For a rough approximation w assume that the pressure decreases ½ pound per square inch for eve feet of ascent.

It is calculated that at a height of about 3½ miles above the sea-le weight of a cubic foot of air is only one half what it is at the surface earth, at seven miles only one fourth, at fourteen miles only one six at twenty-one miles only one sixty-fourth, and at a height of over five miles it becomes so attenuated as to have no appreciable weight.

The pressure of the atmosphere increases with the depth of shafts to about one inch rise in the barometer for each 900 feet increase in this may be taken as a rough-and-ready rule for ascertaining the d shafts.

Pressure of the Atmosphere per Square Inch an Square Foot at Various Readings of the Barome

RULE.—Barometer in inches × .4908 = pressure per square inch; p per square inch × 144 = pressure per square foot.

Barometer.	Pressure per Sq. In.	Pressure per Sq. Ft.	Barometer.	Pressure per Sq. In.
in.	lbs.	lbs.*	in.	lbs.
28.00	13.74	1978	29.75	14.60
28.25	13.86	1995	30.00	14.72
28.50	13.98	2013	30.25	14.84
28.75	14.11	2031	30.50	14.96
29.00	14.23	2049	30.75	15.09
29.25	14.35	2066	31.00	15.21
29.50	14.47	2083		

* Decimals omitted.

For lower pressures see table of the Properties of Steam.

Barometric Readings corresponding with Diffe Altitudes, in French and English Measures.

Altitude.	Reading of Barometer.	Altitude.	Reading of Barometer.	Altitude.	Reading of Barometer.	Altitude.
meters.	mm.	feet.	inches.	meters.	mm.	feet.
0	762	0.	30.	1147	660	3763.2
21	760	68.9	29.92	1269	650	4163.3
127	750	416.7	29.52	1393	640	4568.3
234	740	767.7	29.13	1519	630	4983.1
342	730	1122.1	28.74	1647	620	5403.2
453	720	1486.2	28.35	1777	610	5830.2
564	710	1850.4	27.95	1909	600	6243.
678	700	2224.5	27.55	2043	590	6702.9
793	690	2599.7	27.16	2180	580	7152.4
909	680	2962.1	26.77	2318	570	7605.1
1027	670	3369.5	26.38	2460	560	8071.

Levelling by the Barometer and by Boiling (Trautwine.)—Many circumstances combine to render the result kind of levelling unreliable where great accuracy is required. It is to read off from an aneroid (the kind of barometer usually empl engineering purposes) to within from two to five or six feet, depen its size. The moisture or dryness of the air affects the results; als the vicinity of mountains, and the daily atmospheric tides, whic incessant and irregular fluctuations in the barometer. A baromet ing quiet[illegible] will often vary 1/4 of an inch within a few ho [illegible]ence of elevation of nearly 100 feet. No form [illegible]t shall embrace these sources of error.

To Find the Difference in Altitude of Two Places.—Take from the table the altitudes opposite to the two boiling temperatures, or to the two barometer readings. Subtract the one opposite the lower reading from that opposite the upper reading. The remainder will be the required height, as a rough approximation. To correct this, add together the two thermometer readings, and divide the sum by 2, for their mean. From the table of corrections for temperature, take out the number under this mean. Multiply the approximate height just found by this number.

At 70° F. pure water will boil at 1° less of temperature for an average of about 550 feet of elevation above sea-level, up to a height of 1/2 a mile. At a height of 1 mile, 1° of boiling temperature will correspond to about 560 feet of elevation. In the table the mean of the temperatures at the two stations is assumed to be 32°F., at which no correction for temperature is necessary in using the table.

Boiling-point in deg. Fah.	Barom. in.	Altitude above Sea-level, feet.	Boiling-point in deg. Fah.	Barom. in.	Altitude above Sea-level, feet.	Boiling-point in deg. Fah.	Barom. in.	Altitude above Sea-level, feet.
4°	16.79	15,221	196	21.71	8,481	208	27.73	2,063
5	17.16	14,649	197	22.17	7,932	208.5	28.00	1,809
6	17.54	14,075	198	22.64	7,381	209	28.29	1,539
7	17.98	13,498	199	23.11	6,843	209.5°	28.56	1,290
8	18.32	12,934	200	23.59	6,304	210	28.85	1,025
[illegible]	18.72	12,367	201	24.08	5,764	210.5	29.15	754
[illegible]	19.13	11,799	202	24.58	5,225	211	29.42	512
[illegible]	19.54	11,243	203	25.08	4,697	211.5	29.71	255
[illegible]	19.96	10,685	204	25.59	4,169	212	30.00	S. L. = 0
[illegible]	20.39	10,127	205	26.11	3,642	212.5	30.30	−261
[illegible]	20.82	9,579	206	26.64	3,115	213	30.59	−511
[illegible]	21.26	9,031	207	27.18	2,589			

CORRECTIONS FOR TEMPERATURE.

Mean temp. F. in shade.	0	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
Multiply by	.933	.954	.975	.996	1.016	1.036	1.058	1.079	1.100	1.121	1.142

Moisture in the Atmosphere.—Atmospheric air always contains a small quantity of carbonic-acid gas and a varying quantity of aqueous vapor. Pure mountain air contains about 3 to 4 parts of carbonic acid in 10,000. A properly ventilated room should contain not more than six parts in 10,000.

The degree of saturation or relative humidity of the air is determined by the use of the dry and wet bulb thermometer. The degree of saturation for a number of different readings of the thermometer is given in the following table:

INDICATIONS OF THE HYGROMETER (DRY AND WET BULB), FROM MR. GLAISHER'S OBSERVATIONS AT GREENWICH.

Temperature of the Air, Fahrenheit.	Difference of Temperature or Degrees of Cold in the Wet-bulb Thermometer.																							
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
	Degrees of Humidity, Saturation being 100.																							
32°	87	75																						
42°	92	85	78	72	66	60	54	49	44	40	36	33	30	27										
52°	93	86	80	74	69	64	59	54	50	46	42	39	36	33	30	27	25							
62°	94	88	82	77	72	67	62	58	54	50	47	44	41	38	35	32	30	28	26	24				
72°	94	89	84	79	74	69	65	61	57	54	51	48	45	42	39	36	34	32	30	28	26	24	23	22
82°	95	90	85	80	76	72	68	64	60	57	54	51	48	45	42	40	38	35	33	31	29	27	26	25
92°	95	90	85	81	77	73	70	66	62	59	56	53	50	47	45	43	41	38	36	34	32	30	28	26

Weights of Air, Vapor of Water, and Saturated Mi of Air and Vapor at Different Temperatures, un the Ordinary Atmospheric Pressure of 29.92 inches of Mercury.

Temperature, Fahrenheit.	Weight of a Cubic Ft. of Dry Air at Different Temperatures, lbs.	Elastic Force of Vapor, Inches of Mercury.	Mixtures of Air Saturated with V			
			Elastic Force of the Air in Mixture of Air and Vapor, Inches of Mercury.	Weight of Cubic Foot of the Mixture of Air and Vapor.		
				Weight of the Air, lbs.	Weight of the Vapor, pounds.	Total W'ght of Mixture, pounds.
0°	.0864	.044	29.877	.0863	.000079	.086379
12	.0842	.074	29.849	.0840	.000130	.084130
22	.0824	.118	29.803	.0821	.000202	.082302
32	.0807	.181	29.740	.0802	.000304	.080504
42	.0791	.267	29.654	.0784	.000440	.078840
52	.0776	.388	29.533	.0766	.000627	.077227
62	.0761	.556	29.365	.0747	.000881	.075581
72	.0747	.785	29.136	.0727	.001221	.073921
82	.0733	1.092	28.829	.0706	.001667	.072267
92	.0720	1.501	28.420	.0684	.002250	.070717
102	.0707	2.036	27.885	.0659	.002997	.068897
112	.0694	2.731	27.190	.0631	.003946	.067046
122	.0682	3.621	26.300	.0599	.005142	.065042
132	.0671	4.752	25.169	.0564	.006639	.063039
142	.0660	6.165	23.756	.0524	.008473	.060873
152	.0649	7.930	21.991	.0477	.010716	.058416
162	.0638	10.099	19.822	.0423	.013415	.055715
172	.0628	12.758	17.163	.0360	.016682	.052682
182	.0618	15.960	13.961	.0288	.020536	.049336
192	.0609	19.828	10.093	.0205	.025142	.045642
202	.0600	24.450	5.471	.0109	.030545	.041445
212	.0591	29.921	0.000	.0000	.036820	.036820

The weight in lbs. of the vapor mixed with 100 lbs. of pure ai given temperature and pressure is given by the formula

$$\frac{62.3 \times E}{29.92 - E} \times \frac{29.92}{p},$$

where E = elastic force of the vapor at the given temperature, in mercury; p = absolute pressure in inches of mercury, = 29.92 for atmospheric pressure.

Specific Heat of Air at Constant Volume and at Co Pressure.—Volume of 1 lb. of air at 32° F. and pressure of 14.7 lb in. = 12.387 cu. ft. = a column 1 sq. ft. area × 12.387 ft. high. Raising ature 1° F. expands it $\frac{1}{491.2}$, or to 12.4122 ft. high—a rise of .0252 fo

Work done = 2116 lbs. per sq. ft. × .02522 = 53.37 foot-pounds, or 53 = .0686 heat units.

The specific heat of air at constant pressure, according to Re 0.2375; but this includes the work of expansion, or .0686 heat unit the specific heat at constant volume = 0.2375 − .0686 = 0.1689.

Ratio of specific heat at constant pressure to specific heat at volume = .2375 ÷ .1689 = 1.406. (See Specific Heat, p. 458.)

Flow of Air through Orifices.—The theoretical velocity per second of flow of any fluid, liquid, or gas through an orifice $\sqrt{2gh} = 8.02\sqrt{h}$, in which h = the "head" or height of the flui quired to produce the pressure of the fluid at the level of the

$\frac{v^2}{2g}$. … f flow in cubic feet per second is equal to the ity by the area of the orifice, in square feet, multiplied by a f flow," which takes into account the contraction of the vein eam, the friction of the orifice, etc.

ving through an orifice or short tube, from a reservoir of the nto a reservoir of the pressure p_2, Weisbach gives the follow- r the coefficient of flow, obtained from his experiments.

FLOW OF AIR THROUGH AN ORIFICE.

Coefficient c in formula $v = c\sqrt{2gh}$.

	Ratio of pressures $p_1 \div p_2$	1.05	1.09	1.43	1.65	1.89	2.15
e.	Coefficient.................	.555	.589	.692	.724	.754	.788
	Ratio of pressures	1.05	1.09	1.36	1.67	2.01	....
res	Coefficient.................	.558	.573	.634	.678	.723	....

FLOW OF AIR THROUGH A SHORT TUBE.

.,	Ratio of pressures $p_1 \div p_2$	1.05	1.10	1.30	....	....	....
n.	Coefficient	.730	.771	.830	....	....	....
m.,	Ratio of pressures........	1.41	1.69	....	....	....	....
m.	Coefficient.................	.813	.822	....	....	....	....
-.	Ratio of pressures........	1.24	1.38	1.59	1.85	2.14	....
m. ed.	Coefficient.................	.979	.986	.965	.971	.978	....

EQUATIONS FOR FLOW OF AIR FROM A RESERVOIR THROUGH AN ORIFICE. (Peabody's Thermodynamics, p. 135.)

$$\text{For } p_1 > 2p_a, \quad G = 0.530\, F\,\frac{p_1}{\sqrt{T_1}};$$

$$p_1 < 2p_a, \quad G = 1.060\, F\sqrt{\frac{p_a(p_1 - p_a)}{T_1}};$$

air through the orifice in lbs. per sec., F = area of orifice in sq. olute pressure in reservoir in lbs. per sq. in., p_a = pressure of T_1 = absolute temperature, Fahr., of air in reservoir.

es, Tables, and Data, p. 891) gives, for the velocity of flow of air rifice due to small differences of pressure,

$$V = C\sqrt{\frac{2gh}{12} \times 773.2 \times \left(1 + \frac{t - 32}{493}\right) \times \frac{29.92}{p}},$$

plified,

$$V = 352\, C\sqrt{\left(1 + .00203(t - 32)\right)\frac{h}{p}};$$

velocity in feet per second; $2g = 64.4$; h = height of the column inches, measuring the difference of pressure; t = the tempera- and p = barometric pressure in inches of mercury. 773.2 is the r at 32° under a pressure of 29.92 inches of mercury when that of ight of water is taken as 1.

the formula becomes $V = 363C\sqrt{\frac{h}{p}}$, and if $p = 29.92$ inches $V =$

cient of efflux C, according to Weisbach, is:

l mouthpiece, of form of the contracted vein, ssures of from .23 to 1.1 atmospheres......	$C = .97$ to .99
fices in thin plates................	$C = .56$ to .79
rical mouthpieces..............	$C = .81$ to .84
ounded at the inner end.....	$C = .92$ to .93
verging mouthpieces	$C = .90$ to .99

f Air in Pipes.—Hawksley (Proc. Inst. C. E., xxxiii, 55)

his formula for flow of water in pipes $v = 48\sqrt{\frac{HD}{L}}$ may also

d for flow *of air. In this case* H = height in feet of a *to produce the pressure causing* the flow, or the lo for a given flow; v = velocity in feet per second, D = diameter in feet, length in feet.

If the head is expressed in inches of water, h, the air being take 62° F., its weight per cubic foot at atmospheric pressure = .0761 lb. $H = \frac{62.36}{.0761 \times 12} = 68.3h$. If d = diameter in inches, $D = \frac{d}{12}$, and the for becomes $v = 114.5\sqrt{\frac{hd}{L}}$, in which h = inches of water column, d = d eter in inches and L = length in feet; $h = \frac{Lv^2}{13110d}$; $d = \frac{Lv^2}{13110h}$.

The quantity in cubic feet per second is

$$Q = .7854\frac{d^2}{144}v = .6245\sqrt{\frac{hd^5}{L}};\quad d = \sqrt[5]{\frac{Q^2L}{.39h}};\quad h = \frac{Q^2L}{.39d^5}.$$

The horse-power required to drive air through a pipe is the volum cubic feet per second multiplied by the pressure in pounds per squar and divided by 550. Pressure in pounds per square foot = P = inc water column × 5.196, whence horse-power =

$$HP. = \frac{QP}{550} = \frac{Qh}{105.9} = \frac{Q^3L}{41.3d^5}.$$

If the head or pressure causing the flow is expressed in pounds per s inch = p, then $h = 27.71p$, and the above formulæ become

$$v = 602.7\sqrt{\frac{pd}{L}};\quad p = \frac{Lv^2}{363,300d};\quad d = \frac{Lv^2}{363,300p};$$

$$Q = 3.287\sqrt{\frac{pd^5}{L}};\quad p = \frac{Q^2L}{10.806d^5};\quad d = \sqrt[5]{\frac{Q^2L}{10.806p}};$$

$$HP. = \frac{Q144p}{550} = .2618Qp = .02421\frac{Q^3L}{d^5}.$$

Volume of Air Transmitted in Cubic Feet per Minu Pipes of Various Diameters.

Formula $Q = \frac{.7854}{144}d^2v \times 60$.

Veloc'y of Flow Feet p. sec.	Actual Diameter of Pipe in Inches.										
	1	2	3	4	5	6	8	10	12	16	20
1	.327	1.31	2.95	5.24	8.18	11.78	20.94	32.73	47.12	83.77	130.9
2	.655	2.62	5.89	10.47	16.36	23.56	41.89	65.45	94.25	167.5	261.8
3	.982	3.93	8.84	15.7	24.5	35.3	62.8	98.2	141.4	251.3	392.7
4	1.31	5.24	11.78	20.9	32.7	47.1	83.8	131	188	335	523
5	1.64	6.54	14.7	26.2	41	59	104	163	235	419	654
6	1.96	7.85	17.7	31.4	49.1	70.7	125	196	283	502	785
7	2.29	9.16	20.6	36.6	57.2	82.4	146	229	330	586	916
8	2.62	10.5	23.5	41.9	65.4	94	167	262	377	670	1047
9	2.95	11.78	26.5	47	73	106	188	294	424	754	1178
10	3.27	13.1	29.4	52	82	118	209	327	471	838	1309
12	3.93	15.7	35.3	63	98	141	251	393	565	1005	1571
15	4.91	19.6	44.2	78	122	177	314	491	707	1256	1963
18	5.89	23.5	53	94	147	212	377	589	848	1508	2356
20	6.54	26.2	59	105	164	235	419	654	942	1675	2618
24	7.85	31.4	71	125	196	283	502	785	1131	2010	3141
[illegible]	8.18	32.7	73	131	204	294	523	818	1178	2094	3272
[illegible]	[illegible]	[illegible]	[illegible]	146	229	330	586	916	1319	2346	3665
[illegible]	[illegible]	[illegible]	[illegible]	157	245	353	628	982	1414	2513	3927

y's formula and its derivatives the numerical coefficients are s scarcely possible, however, that they can be accurate except ed range of conditions. In the case of water it is found that of friction, on which the loss of head depends, varies with the imeter of the pipe, and with the velocity, as well as with the he interior surface. In the case of air and other gases we ion, the decrease in density and consequent increase in volume y due to the progressive loss of head from one end of the pipe

that according to the experiments of D'Aubuisson and those of ommission on the resistance of air through long conduits or inution of pressure is very nearly directly as the length, and of the velocity and inversely as the diameter. The resistance y the density.

tements are correct, then the formulæ $h = \frac{Lv^2}{cd}$ and $h = \frac{Q^2L}{c'd^5}$ vatives are correct in form, and they may be used when the fficients c and c' are obtained by experiment.

e forms of the above formulæ as correct, and let C be a vari- t, depending upon the length, diameter, and condition of sur- ipe, and possibly also upon the velocity, the temperature and be determined by future experiments, then for h = head in r, d = diameter in inches, L = length in feet, v = velocity in d, and Q = quantity in cubic feet per second:

$$C\sqrt{\frac{hd}{L}}; \qquad d = \frac{Lv^2}{C^2h}; \qquad h = \frac{Lv^2}{C^2d};$$

$$54C\sqrt{\frac{hd^5}{L}}; \qquad d = \sqrt[5]{\frac{33683Q^2L}{C^2h}}; \qquad h = \frac{33683Q^2L}{C^2d^5}.$$

ce or loss of pressure p in pounds per square inch,

$$.71p \qquad \sqrt{h} = 5.264\sqrt{p};$$

$$264C\sqrt{\frac{pd}{L}}; \qquad d = \frac{Lv^2}{27.71C^2p}; \qquad p = \frac{Lv^2}{27.71C^2d};$$

$$02871C\sqrt{\frac{pd^5}{L}}; \qquad d = \sqrt[5]{\frac{1213Q^2L}{C^2p}}; \qquad p = \frac{1213Q^2L}{C^2d^5}.$$

ormulæ for flow of air, see Mine Ventilation.)

ressure in Ounces per Square Inch.—B. F. Sturte- y uses the following formulæ:

$$= \frac{Lv^2}{25000d}; \quad v = \frac{\sqrt{25000dp_1}}{L}; \quad d = \frac{Lv^2}{25000p_1};$$

loss of pressure in ounces per square inch, v = velocity of air cond, and L = length of pipe in feet. If p is taken in pounds h, these formulæ reduce to

$$= .0000025\frac{Lv^2}{d}; \quad v = \frac{.00158\sqrt{dp}}{L}; \quad d = \frac{.0000025Lv^2}{p}.$$

uced from the common formula (Weisbach's), $p = f\frac{l}{d}\frac{v^2}{2g}$, in 01608.

g table is condensed from one given in the catalogue of B. F. mpany.

ssure in pipes 100 feet long, in ounces per square inc' gth, the loss is proportional to the length.

Velocity of Air, feet per min.	Diameter of Pipe in Inches.									
	1	2	3	4	5	6	7	8	9	10
	Loss of Pressure in Ounces.									
600	.400	.200	.133	.100	.080	.067	.057	.050	.044	.040
1200	1.600	.800	.533	.400	.320	.267	.229	.200	.178	.160
1800	3.600	1.800	1.200	.900	.720	.600	.514	.450	.400	.360
2400	6.400	3.200	2.133	1.600	1.280	1.067	.914	.800	.711	.640
3000	10.	5.	3.333	2.5	2.	1.667	1.429	1.250	1.111	1.000
3600	14.4	7.2	4.8	3.6	2.88	2.4	2.057	1.8	1.6	1.44
4200	......	9.8	6.553	4.9	3.92	3.267	2.8	2.45	2.178	1.96
4800	.....	12.8	8.533	6.4	5.12	4.267	3.657	3.2	2.844	2.56
6000	....	20.	13.333	10.0	8.0	6.667	5.714	5.0	4.444	4.0
	Diameter of Pipe in Inches.									
	14	16	18	20	22	24	28	32	36	40
	Loss of Pressure in Ounces.									
600	.029	.026	.022	.020	.018	.017	.014	.012	.011	.010
1200	.114	.100	.089	.080	.073	.067	.057	.050	.044	.040
1800	.257	.225	.200	.180	.164	.156	.129	.112	.100	.090
2400	.457	.400	.356	.320	.291	.267	.239	.200	.178	.160
3600	1.029	.900	.800	.720	.655	.600	.514	.450	.400	.360
4200	1.400	1.225	1.089	.980	.891	.817	.700	.612	.544	.490
4800	1.829	1.600	1.422	1.280	1.164	1.067	.914	.800	.711	.640
6000	2.857	2.500	2.222	2.000	1.818	1.667	1.429	1.250	1.111	1.000

Effect of Bends in Pipes. (Norwalk Iron Works Co.

Radius of elbow, in diameter of pipe = 5 3 2 1½ 1¼ 1
Equivalent lgths. of straight pipe, diams 7.85 8.24 9.03 10.36 12.72 17.51

Compressed-air Transmission. (Frank Richards, *Am.* March 8, 1894.)—The volume of free air transmitted may be assum directly as the number of atmospheres to which the air is com Thus, if the air transmitted be at 75 pounds gauge-pressure, or six pheres, the volume of free air will be six times the amount give table (page 486). It is generally considered that for economical trans the velocity in main pipes should not exceed 20 feet per second. smaller distributing pipes the velocity should be decidedly less than

The loss of power in the transmission of compressed air in gener a serious one, or at all to be compared with the losses of power in th tion of compression and in the re-expansion or final application of

The formulas for loss by friction are all unsatisfactory. The sta of observed facts in this line are in a more or less chaotic state, a evidently unreliable.

A statement of the friction of air flowing through a pipe involves all the following factors: Unit of time, volume of air, pressure of ai eter of pipe, length of pipe, and the difference of pressure at the the pipe or the head required to maintain the flow. Neither of these can be allowed its independent and absolute value, but is subject to cations in deference to its associates. The flow of air being assume uniform at the entrance to the pipe, the volume and flow are not after that. The air is constantly losing some of its pressure and its is constantly increasing. The velocity of flow is therefore also se accelerated continually. This also modifies the use of the length pipe as a constant factor.

Then, besides the fluctuating values of these factors, there is the c of the pipe itself. The actual diameter of the pipe, especially smaller different from the nominal diameter. The pipe strai be crooked and have numerous elbows. Mr. R y as equivalent to a length of pipe.

ead or Additional Pressure in pounds per sq. in. equired to deliver Air at 75 Pounds Gauge-pressure hrough Pipes of Various Sizes and Lengths. (Frank ichards.)

1″ Pipe.

Free air per min.	50	100	300	500	1,000
	Loss of pressure, lbs. p. sq. in.				
[illegible]	.245	.49	1.47	2.45	4.9
[illegible]	.981	1.962	5.886	9 81	......
[illegible]	3.925	7.85	......	......	......
[illegible]	8.829	17.66	......	......	......

1¼″ Pipe.

Free air per min.	50	100	300	500	1,000
25	.056	.112	.336	.561	1.12
50	.224	.449	1.35	2.24	4.49
[illegible]	.897	1.79	5.38	8.97	......
[illegible]	2.02	3.94	12.11	......	......
[illegible]	3.59	7.18	......	......	......

1½″ Pipe.

Free air per min.	50	100	300	500	1,000
25	.017	.034	.103	.171	.34
50	.068	.137	.411	.685	1.37
[illegible]	.274	.548	1.64	2.74	5.48
[illegible]	.616	1.23	3.69	6.16	12.33
[illegible]	1.09	2.19	6.57	10.96	21.9

2″ Pipe.

Free air per min.	50	100	300	500	1,000
50	.019	.038	.114	.19	.38
[illegible]	.076	.152	.457	.761	1.52
[illegible]	.171	.343	1.03	1.71	3.44
[illegible]	.304	.609	1.83	3 04	6.09
[illegible]	.476	.952	2.86	4.76	9.53
[illegible]	.685	1.37	4.11	6.85	13.72

2½″ Pipe.

Free air per min.	200	300	500	1,000	2,000
[illegible]	.087	.13	.217	.434	.87
[illegible]	.347	.521	.868	1.74	3.47
[illegible]	.781	1.17	1.95	3.91	7.81
[illegible]	1.39	2.08	3.47	6.94	13.89
[illegible]	2.17	3.25	5.42	10.85	21.7

3″ Pipe.

Free air per min.	200	300	500	1,000	2,000
[illegible]	.0333	.05	.0833	.166	.33
[illegible]	.133	.2	.333	.666	1.33
[illegible]	.3	.45	.75	1 5	3
[illegible]	.533	.8	1.33	2.66	5 33
[illegible]	.833	1.25	2.08	4.16	8.33

3½″ Pipe.

Free air per min.	200	300	500	1,000	2,000
[illegible]	.0832	.125	.208	.416	.83
[illegible]	.332	.499	.832	1.66	3.32
[illegible]	.748	1.12	1.87	3.75	7.49
[illegible]	1.328	1.99	3.33	6.66	13.3
[illegible]	2.08	3.12	5.2	10.4	20.8

4″ Pipe.

Cubic ft. free air per min.	200	300	400	1,000	2,000
	Loss of pressure, lbs. p. sq. in.				
500	.16	.24	.4	.8	1.6
750	.36	.54	.9	1.8	3.6
1,000	.64	.96	1.6	3.2	6.4
1,250	1.	1.5	2.5	5.	10.
1,500	1.44	2.16	3.6	7.2	14.4

5″ Pipe.

Cubic ft. free air per min.	500	1,000	2,000	4,000	5,000
500	.11	.22	.44	.88	1.1
1,000	.44	.881	1.76	3.52	4.4
1,500	.99	1.98	3.96	7.92	9.9
2,000	1.76	3.52	7.04	14.08	......
2,500	2.75	5.5	11.	......	......

6″ Pipe.

Cubic ft. free air per min.	1,000	2,000	4,000	5,000	10,000
1,000	.354	.708	1.42	1.77	3.54
1,500	.799	1.599	3.2	3.99	7.99
2,000	1.417	2.83	5.67	7.09	14.17
2,500	2.22	4.44	8.89	11.1	.....
3,000	3.18	6.37	12.7	15.9	.....

8″ Pipe.

Cubic ft. free air per min.	2,000	4,000	8,000	10,000	15,000
2,000	.598	1.19	2.39	2.99	4.48
2,500	.935	1.87	3.74	4.68	7.02
3,000	1.25	2.49	4.99	6.24	9.36
4,000	2.39	4.79	9.58	11.97	.....
5,000	3.74	7.48	14.97	18.71	.....

10″ Pipe.

Cubic ft. free air per min.	2,000	4,000	8,000	10,000	15,000
2,500	.286	.57	1.14	1.43	2.15
5,000	1.14	2.29	4.57	5.71	8.56
7,500	2.57	5.15	10.29	12.86	......
10,000	4.57	9.14	......	......	......

12″ Pipe.

Cubic ft. free air per min.	2,000	4,000	8,000	10,000	20,000
2,500	.11	.22	.44	.55	1.101
5,000	.44	.88	1.76	2.2	4.4
7,500	.99	1.98	3.96	4.95	9.91
10,000	1.76	3.52	7.05	[illegible]	17.6

lthough Mr. Richards does not give any formula with [illegible] pection of it shows that for any given diameter the [illegible]

The impossibility of measuring the true quantity of air by an anemom[eter]
held stationary in one position is shown by the following figures, giv[en by]
Wm. Daniel (Proc. Inst. M. E., 1875), of the velocities of air found at diff[erent]
points in the cross-sections of two different airways in a mine.

DIFFERENCES OF ANEMOMETER READINGS IN AIRWAYS.

8 ft. square.

1712	1795	1859	1329
1622	1685	1782	1091
1477	1344	1524	1049
1262	1356	1293	1333

Average 1469.

5 × 8 ft.

1170	1309	1288
948	1104	1177
1134	1049	1106

Average 1132

Equation of Pipes.—It is frequently desired to know what nu[mber]
of pipes of a given size are equal in carrying capacity to one pipe of a l[arger]
size. At the same velocity of flow the volume delivered by two pip[es of]
different sizes is proportional to the squares of their diameters; thu[s one]
4-inch pipe will deliver the same volume as four 2-inch pipes. With the [same]
head, however, the velocity is less in the smaller pipe, and the volum[e de-]
livered varies about as the square root of the fifth power (i.e., as th[e 2.5]
power). The following table has been calculated on this basis. The fi[gure]
opposite the intersection of any two sizes is the number of the smaller
pipes required to equal one of the larger. Thus, one 4-inch pipe is eq[ual to]
5.7 2-inch pipes.

Diam. in.	1	2	3	4	5	6	7	8	9	10	12	14	16	18	20
2	5.7	1													
3	15.6	2.8	1												
4	31	5.7	2.1	1											
5	55.9	9.9	3.6	1.7	1										
6	88.2	15.6	5.7	2.8	1.6	1									
7	130	22.9	8.3	4.1	2.3	1.5	1								
8	181	32	11.7	5.7	3.2	2.1	1.4	1							
9	243	43.	15.6	7.6	4.3	2.8	1.9	1.3	1						
10	316	55.9	20.3	9.9	5.7	3.6	2.4	1.7	1.3	1					
11	401	70.9	25.7	12.5	7.2	4.6	3.1	2.2	1.7	1.3					
12	499	88.2	32	15.6	8.9	5.7	3.8	2.8	2.1	1.6	1				
13	609	108	39.1	19	10.9	7.1	4.7	3.4	2.5	1.9	1.2				
14	733	130	47	22.9	13.1	8.3	5.7	4.1	3.0	2.3	1.5	1			
15	871	154	55.9	27.2	15.6	9.9	6.7	4.8	3.6	2.8	1.7	1.2			
16	...	181	65.7	32	18.3	11.7	7.9	5.7	4.2	3.2	2.1	1.4	1		
17	...	211	76.4	37.2	21.3	13.5	9.2	6.6	4.9	3.8	2.4	1.6	1.2		
18	...	243	88.2	43	24.6	15.6	10.6	7.6	5.7	4.3	2.8	1.9	1.3	1	
19	...	278	101	49.1	28.1	17.8	12.1	8.7	6.5	5	3.2	2.1	1.5	1.1	
20	...	316	115	55.9	32	20.3	13.8	9.9	7.4	5.7	3.6	2.4	1.7	1.3	1
22	...	401	146	70.9	40.6	25.7	17.5	12.5	9.3	7.2	4.6	3.1	2.2	1.7	[illegible]
24	...	499	181	88.2	50.5	32	21.8	15.6	11.6	8.9	5.7	3.8	2.8	2.1	[illegible]
26	...	609	221	108	61.7	39.1	26.6	19	14.2	10.9	7.1	4.7	3.4	2.5	[illegible]
28	...	733	266	130	74.2	47	32	22.9	17.1	13.1	8.3	5.7	4.1	3	[illegible]
30	...	871	316	154	88.2	55.9	38	27.2	20.3	15.6	9.9	6.7	4.8	3.6	[illegible]
36	...	...	499	243	130	88.2	60	43	32	24.6	15.6	10.6	7.6	5.7	[illegible]
42	...	...	733	357	205	130	88.2	63.2	47	36.2	19	15.6	11.2	8.2	[illegible]
48	...	[illegible]	[illegible]	[illegible]	[illegible]	181	123	88.2	62.7	50.5	32	21.8	15.6	11.6	[illegible]
54	...	[illegible]	[illegible]	[illegible]	[illegible]	243	165	118	88.2	67.8	43	29.2	20.9	15.6	[illegible]
60	...	[illegible]	[illegible]	[illegible]	[illegible]	316	215	154	115	88.2	55.9	38	27.2	20.3	[illegible]

Pressure in Compressed Air Pipe-main, at St. Gothard Tunnel.

(E. Stockalper.)

atmospheric pressure and 32° F.	Volume per second of compressed air at mean density.	Mean density of compressed air. (Water = 1.)	Weight of air flowing per second.	Mean velocity in feet per second.	Observed Pressures. Pressure at beginning of pipe.	Pressure at end of pipe.	Loss of Pressure.		Value of c' in formula $p = \frac{Q'L}{c'^2d^5}$
t.	cu. ft.	den.	lbs.	feet.	at.	at.	lbs. per sq. in.	%	
50	6.534	.00650	2.669	19.32	5.60	5.24	5.292	6.4	610
	7.063	.00603	2 669	37.14	5.24	5.00	3.528	4.6	515
02	5.509	.00514	1.776	16.30	4.35	4.13	3.234	5.1	519
	5.863	.00482	1.776	.....	4.13	....	.....	...	...
54	5.262	.00449	1.483	15.58	3.84	3.65	2.793	5.0	466
	5.580	.00423	1.483	29.34	3.65	3.54	1.617	3.0	422

f the pipe 7.87 in diameter was 15,092 ft., and of the smaller The mean temperature of the air in the large pipe was 70° F. ll pipe 80° F.

WIND.

the Wind.—Smeaton in 1759 published a table of the essure of wind, as follows:

Y AND FORCE OF WIND, IN POUNDS PER SQUARE INCH.

sq. ft. pounds.	Common Appellation of the Force of Wind.	Miles per Hour.	Feet per second.	Force per sq. ft. pounds.	Common Appellation of the Force of Wind.
.005	Hardly perceptible.	18	26.4	1.55	Very brisk.
		20	29.34	1.968	
.020	Just perceptible.	25	36.67	3.075	
.044		30	44.01	4.429	High wind.
.079	Gentle pleasant wind.	35	51.34	6.027	
.123		40	58.68	7.873	Very high storm.
.177		45	66.01	9.963	
.241		50	73.35	12.30	
.315	Pleasant brisk gale.	55	80.7	14.9	
.400		60	88.02	17.71	Great Storm.
.492		66	95.4	20.85	
.708		70	102.5	24.1	
.964		75	110.	27.7	Hurricane.
.107		80	117.36	31.49	
.25		100	146.67	49.2	Immense hurricane.

s per square foot in the above table correspond to the 005V^2, in which V is the velocity in miles per hour. *Eng'g* 1893, says that the formula was never well established, and efly on Smeaton's name and for lack of a better. It was put for surfaces for use in windmill practice. The trend of ce is that it is approximately correct only for such surfaces arge solid bodies it often gives greatly too large results. y others are thus compared with Smeaton's f

n formula.........................

ed by Prof. Martin.........................

" *Whipple and Dines*.................

At 60 miles per hour these formulas give for the pressure pe
18, 14.4 and 10.44 lbs., respectively, the pressure varying by a
the square of the velocity. Lieut. Crosby's experiments (E
1890), claiming to prove that $P = fV$ instead of $P = fV^2$, ar
A. R. Wolff (The Windmill as a Prime Mover, p. 9) gives as t
pressure per sq. ft. of surface, $P = \frac{dQv}{g}$, in which d = density o
per cu. ft. $= \frac{.018743(p + P)}{t}$; p being the barometric pressu
foot at any level, and temperature of 32° F., t any absolute
Q = volume of air carried along per square foot in one secon
of the wind in feet per sec., $g = 32.16$. Since $Q = v$ cu. ft. pe
Multiplying this by a coefficient 0.93 found by experiment, a
the above value of d, he obtains $P = \frac{0.017431 \times p}{\frac{t \times 32.16}{v^2} - .018743}$, a
= 2116.5 lbs. per sq ft. or average atmospheric pressure at
$P = \frac{36.8929}{\frac{t \times 32.16}{v^2} - 0.18743}$, an expression in which the pressure is
with the temperature; and he gives a table showing the rel
velocity and pressure for temperatures from 0° to 100° F.,
from 1 to 80 miles per hour. For a temperature of 45° F. the p
with those in Smeaton's table, for 0° F. they are about 10 pe
and for 100° 10 per cent less. Prof. H. Allen Hazen, *Eng'g*
1890, says that experiments with whirling arms, by exposing p
wind, and on locomotives with velocities running up to 40 m
have invariably shown the resistance to vary with V^2. I
$P = .005SV^2$, in which P = pressure in pounds, S = surface
V = velocity in miles per hour, the doubtful question is t
the accuracy of the first two factors in the second member o
The first factor has been variously determined from .003 to .0
determined as low as .0014.—Ed. *Eng'g News*].

The second factor has been found in some experiments w
whirling arms and low velocities to vary with the perimete
but this entirely disappears with longer arms or straight li
the only question now to be determined is the value of the co
haps some of the best experiments for determining this valu
France in 1886 by carrying flat boards on trains. The resulti
this case was, for 44.5 miles per hour, $p = .00535SV^2$.

Mr. Crosby's whirling experiments were made with an ar
It is certain that most serious effects from centrifugal actio
up by using such a short arm, and nothing satisfactory can b
arms less than 20 or 30 ft. long at velocities above 5 miles per

Prof. Kernot, of Melbourne (*Engineering Record*, Feb. 20, 18
experiments at the Forth Bridge showed that the average p
faces as large as railway carriages, houses, or bridges never
thirds of that upon small surfaces of one or two square feet
been used at observatories, and also that an inertia effect, whic
overlooked, may cause some forms of anemometer to give
enormously exceeding the correct indication. Experiments
Crosby showed that the pressure varied directly as the veloci
the early investigators, from the time of Smeaton onwards, m
the square of the velocity. Experiments made by Prof. Ke
varying from 2 to 15 miles per hour agreed with the earlier a
tended to negative Crosby's results. The pressure upon one
or of a block proportioned like an ordinary carriage, was fou
that upon a thin plate of the same area. The same result wa
a square tower. A square pyramid, whose height was three
experienced .8 of the pressure upon a thin plate equal to one c
" an angle was turned to the wind the pressure was increase
bridge consisting of two plate-girders connected by a deck
ed to experience .9 of the pressure on a thin plate equ
e distance between the girders was equal to
d by one fifth when the distance betwee

th. A lattice-work in which the area of the openings was 55% rea experienced a pressure of 80% of that upon a plate of the he pressure upon cylinders and cones was proved to be equal on the diametral planes, and that upon an octagonal prism to than upon the circumscribing cylinder. A sphere was sub- ire of .36 of that upon a thin circular plate of equal diameter. al cup gave the same result as the sphere; when its concavity the wind the pressure was 1.15 of that on a flat plate of equal en a plane surface parallel to the direction of the wind was into contact with a cylinder or sphere, the pressure on the as augmented by about 20%, owing to the lateral escape of the ted. Thus it is possible for the security of a tower or chimney by the erection of a building nearly touching it on one side.

of Wind Registered in Storms.—Mr. Frizell has published records of Greenwich Observatory from 1849 to 1869, at the highest pressure of wind he finds recorded is 41 lbs. there are numerous instances in which it was between 30 and t. Prof. Henry says that on Mount Washington, N. H., a ve- iles per hour has been observed, and at New York City 60 and that the highest winds observed in 1870 were of 72 and 63 respectively.

ody, U. S. A., says, in substance, that the New England coast orms which produce a pressure of 50 lbs. per sq. ft. *Engi-* Aug. 20, 1880.

WINDMILLS.

d Efficiency of Windmills.—Rankine, S. E., p. 215, ving: Let Q = volume of air which acts on the sail, or part ubic feet per second, v = velocity of the wind in feet per ctional area of the cylinder, or annular cylinder of wind, the sail, or part of the sail, sweeps in one revolution. c = a found by experience; then $Q = cvs$. Rankine, from experi- ren by Smeaton, and taking c to include an allowance for or a wheel with four sails, proportioned in the best manner, A = weather angle of the sail at any distance from the axis, the portion of the sail considered makes with its plane of is angle gradually diminishes from the inner end of the sail the velocity of the same portion of the sail, and E = the effi- ficiency is the ratio of the useful work performed to whole stream of wind acting on the surface s of the wheel, which

D being the weight of a cubic foot of air. Rankine's formula

$$\frac{Ru}{\frac{Dsv^3}{2g}} = c\left\{\frac{u}{v}\sin 2A - \frac{u^2}{v^2}(1 - \cos 2A + f) - f\right\},$$

75 and f is a coefficient of friction found from Smeaton's Rankine gives the following from Smeaton's data:

eather-angle	= 7°	13°	19°
tio of speed of greatest effi- ciency, for a given weather- angle, to that of the wind	= 2.63	1.86	1.41
ficiency	= 0.24	0.29	0.31

s the following as the best values for the angle of weather at ces from the axis:

sixths of total radius	1	2	3	4	5	6
gle	18°	19°	18°	16°	12½°	7°

125) shows that Smeaton did not term these the best angles s they "answer as well as any," possibly any that were in ex me. *Wolff says that they* "cannot in the nature of thing *sirable angles." Mathematical* considerations, he says, con- *hat the angle of impulse* depends on the relative velocity of *sail and the wind*, the angle growing larger as th- *Smeaton's angles* do not fulfil this condition.

3. The great cold which results when air expands against a r
forbids expansive working, which is equivalent to saying, forbids
zation of a high degree of efficiency in the use of compressed air.

4. Friction of the air in the pipes, leakage, dead spaces, the resi
fered by the valves, insufficiency of valve-area, inferior workman
slovenly attendance, are all more or less serious causes of loss of

The first cause of loss of work, namely, the heat developed by
sion, is entirely unavoidable. The whole of the mechanical ener
the compressor-piston spends upon the air is converted into heat.
is dissipated by conduction and radiation, and its mechanical equ
work lost. The compressed air, having again reached thermal e
with the surrounding atmosphere, expands and does work in vi
intrinsic energy.

The intrinsic energy of a fluid is the energy which it is capabl
ing against a piston in changing from a given state as to temper
volume, to a total privation of heat and indefinite expansion.

Volumes, Mean Pressures per Stroke, Temperatur in the Operation of Air-compression from 1 Atm and 60° Fahr. (F. Richards, *Am. Mach.*, March 30, 1893.)

Gauge-pressure.	Atmospheres.	Volume with Air at Constant Temp.	Volume with Air not cooled.	Mean Pressure per Stroke; Air Constant Temp.	Mean Pressure per Stroke; Air not cooled.	Temp. of Air; not cooled.	Gauge-pressure.	Atmospheres.	Volume with Air at Constant Temp.	Volume with Air not cooled.	Mean Pressure per Stroke; Air Constant Temp.
1	2	3	4	5	6	7	1	2	3	4	5
0	1	1	1	0	0	60°	80	6.442	.1552	.266	27.38
1	1.068	.9363	.95	.96	.975	71	85	6.782	.1474	.2566	28.16
2	1.136	.8803	.91	1.87	1.91	80.4	90	7.122	.1404	.248	28.89
3	1.204	.8305	.876	2.72	2.8	88.9	95	7.462	.134	.24	29.57
4	1.272	.7861	.84	3.53	3.67	98	100	7.802	.1281	.2324	30.21
5	1.34	.7462	.81	4.3	4.5	106	105	8.142	.1228	.2254	30.81
10	1.68	.5952	.69	7.62	8.27	145	110	8.483	.1178	.2189	31.39
15	2.02	.495	.606	10.33	11.51	178	115	8.823	.1133	.2129	31.98
20	2.36	.4237	.543	12.62	14.4	207	120	9.163	.1091	.2073	32.54
25	2.7	.3703	.494	14.59	17.01	234	125	9.503	.1052	.2020	33.07
30	3.04	.3289	.4538	16.34	19.4	252	130	9.843	.1015	.1969	33.57
35	3.381	.2957	.42	17.92	21.6	281	135	10.183	.0981	.1922	34.05
40	3.721	.2687	.393	19.32	23.66	302	140	10.523	.095	.1878	34.57
45	4.061	.2462	.37	20.57	25.59	321	145	10.864	.0921	.1837	35.09
50	4.401	.2272	.35	21.69	27.39	339	150	11.204	.0892	.1796	35.48
55	4.741	.2109	.331	22.76	29.11	357	160	11.88	.0841	.1722	36.29
60	5.081	.1968	.3144	23.78	30.75	375	170	12.56	.0796	.1657	37.2
65	5.422	.1844	.301	24.75	32.32	389	180	13.24	.0755	.1595	37.96
70	5.762	.1735	.288	25.67	33.83	405	190	13.93	.0718	.154	38.68
75	6.102	.1639	.276	26.55	35.27	420	200	14.61	.0685	.149	39.42

Column 3 gives the volume of air after compression to the give
and after it is cooled to its initial temperature. After compressi
its heat very rapidly, and this column may be taken to represent
of air after compression available for the purpose for which
been compressed.

Column 4 gives the volume of air more nearly as the compre
deal with it. In any compressor the air will lose some of its h
compression. The slower the compressor runs the cooler the
smaller the volume.

Column 5 gives the mean effective resistance to be overcome
cylind- the stroke of compression, supposing the ai
ial temperature. Of course it will not so r
al to be kept in view in economical air-co

6 gives the mean effective resistance to be overcome by the pis-osing that there is no cooling of the air. The actual mean effec-sure will be somewhat less than as given in this column; but for g the actual power required for operating air-compressor cylinders s in this column may be taken and a certain percentage added—cent—and the result will represent very closely the power required mpressor.

an pressures given being for compression from one atmosphere hey will not be correct for computations in compound compression other initial pressure.

Due to Excess of Pressure caused by Heating in mpression-cylinder.—If the air during compression were constant temperature, the compression-curve of an indicator-dia-en from the cylinder would be an isothermal curve, and would fol-law of Boyle and Marriotte, $pv =$ a constant, or $p_1v_1 = p_0v_0$, or , p_0 and v_0 being the pressure and volume at the beginning of ion, and p_1v_1 the pressure and volume at the end, or at any inter-point. But as the air is heated during compression the pressure faster than the volume decreases, causing the work required for pressure to be increased. If none of the heat were abstracted ion or by injection of water, the curve of the diagram would be an curve, with the equation $p_1 = p_0\left(\frac{v_0}{v_1}\right)^{1.405.}$ Cooling the air dur-ression, or compressing it in two cylinders, called compounding, ng the air as it passes from one cylinder to the other, reduces the of this equation, and reduces the quantity of work necessary to iven compression. F. T. Gause (*Am. Mach.*, Oct. 20, 1892), describ-erations of the Popp air-compressors in Paris, says: The greatest alized in compressing in a single cylinder was 33 per cent of that ally possible. In cards taken from the 2000 H.P. compound com-t Quai De La Gare, Paris, the saving realized is 85 per cent of the al amount. Of this amount only 8 per cent is due to cooling dur-ression, so that the increase of economy in the compound com-mainly due to cooling the air between the two stages of compres-compression-curve with exponent 1.25 is the best result that was for compression in a single cylinder and cooling with a very fine The curve with exponent 1.15 is that which must be realized in a inder to equal the present economy of the compound compressor e La Gare.

power required to ress and deliver one foot of Free Air per to a given pressure with no of the air during the com-n; also the horse-power re-supposing the air to be main-at constant temperature the compresion.

Air not cooled.	Air constant temperature.
.0196	.0188
.0361	.0333
.0628	.0551
.0846	.0713
.1032	.0843
.1195	.0946
.1342	.1036
.1476	.1120
.1599	.1195
.1710	.1261
.1815	.1318

Horse-power required to compress and deliver one cubic foot of Compressed Air per minute at a given pressure with no cooling of the air during the compression; also the horse-power required, supposing the air to be maintained at constant temperature during the compression.

Gauge-pressure.	Air not cooled.	Air constant temperature.
5	.0263	.0251
10	.0606	.0559
20	.1483	.1300
30	.2573	.2168
40	.3842	.3138
50	.5261	.4166
60	.6818	.5266
70	.8508	.6456
80	1.0302	.7700
90	1.2177	.8979
100	1.4171	1.0291

se-power given above is the theoretical power, no allowance being friction of the compressor or other losses, which may amount to t or more.

Table for Adiabatic Compression or Expansio

(Proc. Inst. M.E., Jan. 1881, p. 123.)

Absolute Pressure.		Absolute Temperature.		Vol
Ratio of Greater to Less. (Expansion.)	Ratio of Less to Greater. (Compression.)	Ratio of Greater to Less. (Expansion.)	Ratio of Less to Greater. (Compression.)	Ratio of Greater to Less. (Compres- sion.)
1.2	.833	1.054	.948	1.138
1.4	.714	1.102	.907	1.270
1.6	.625	1.146	.873	1.396
1.8	.556	1.186	.843	1.518
2.0	.500	1.222	.818	1.636
2.2	.454	1.257	.796	1.750
2.4	.417	1.289	.776	1.862
2.6	.385	1.319	.758	1.971
2.8	.357	1.348	.742	2.077
3.0	.333	1.375	.727	2.182
3.2	.312	1.401	.714	2.284
3.4	.294	1.426	.701	2.384
3.6	.278	1.450	.690	2.483
3.8	.263	1.473	.679	2.580
4.0	.250	1.495	.669	2.676
4.2	.238	1.516	.660	2.770
4.4	.227	1.537	.651	2.863
4.6	.217	1.557	.642	2.955
4.8	.208	1.576	.635	3.046
5.0	.200	1.595	.627	3.135
6.0	.167	1.681	.595	3.569
7.0	.143	1.758	.569	3.981
8.0	.125	1.828	.547	4.377
9.0	.111	1.891	.529	4.759
10.0	.100	1.950	.513	5.129

Mean Effective Pressures for the Compression of the Stroke when compressing and deliv from one Atmosphere to given Gauge-pressur gle Cylinder. (F. Richards, *Am. Mach.*, Dec. 14, 1893.)

Gauge-pressure.	Adiabatic Compression	Isothermal Compression	Gauge-pressure.	Adiabatic Compression.
1	.44	.43	45	13.95
2	.96	.95	50	15.05
3	1.41	1.4	55	15.98
4	1.86	1.84	60	16.89
5	2.26	2.22	65	17.88
10	4.26	4.14	70	18.74
15	5.99	5.77	75	19.54
20	7.58	7.2	80	20.5
25	9.05	8.49	85	21.22
30	10.39	9.66	90	22.
35	11.59	10.72	95	22.77
40	12.8	11.7	100	23.43

The mean effective pressure for compression only is alway the mean effective pressure for the whole work

…n and Terminal Pressures of Compressed Air used …xpansively for Gauge-pressures from 60 to 100 lbs.

(Frank Richards, *Am. Mach.*, April 13, 1893.)

…ial …res- …re.	60.		70.		80.		90.		100.	
Cut-off.	Mean Air-pressure.	Terminal Air-pressure.	Mean Air-pressure.	Terminal Air-pressure.	Mean Air-pressure.	Terminal Air-pressure.	Mean Air-pressure.	Terminal Air-pressure.	Mean Air-pressure.	Terminal Air-pressure.
[illegible]	23.6	*10.65*	28.74	*12.07*	33.89	*13.49*	39 04	*14.91*	44.19	1.83
[illegible]	28.9	*13.77*	34.75	.6	40 61	2.44	46.46	4.27	53.32	6.11
[illegible]	32.13	.96	38.41	3.09	44.69	5 22	50.98	7.35	57.26	9.48
[illegible]	33.66	2.33	40.15	4.38	46.64	6.66	53.13	8.95	59.62	11.23
[illegible]	35.85	3.85	42 63	6.26	49.41	7.88	56.2	11.39	62.98	13.89
[illegible]	37.93	5 64	44.99	8.39	52.05	11.14	59.11	13.88	66.16	16.64
[illegible]	41.75	10.71	49 31	12.61	56.9	15.86	64 45	19.11	72.02	22.36
[illegible]	45.14	13.26	53.16	17	61.18	20.81	69.19	24.56	77.21	28.33
[illegible]	50.75	21.53	59.51	26.4	68.28	31.27	77.05	36.14	85.82	41.01
[illegible]	51.92	23.69	60.84	28.85	69.76	34.01	78.69	39.16	87.61	44.32
[illegible]	53.67	27.94	62 83	33.03	71.99	38.68	81.14	44 33	90.32	49.97
[illegible]	54.93	30.39	64 25	36 44	73 57	42.49	82 9	48.54	92.22	54.59
[illegible]	56.52	35.01	66.05	41.68	75.59	48.35	85.12	55.02	94.66	61.69
[illegible]	57.79	39.78	67.5	47.08	77.2	54.38	86.91	61.69	96.61	68.99
[illegible]	59.15	47.14	69.03	55.43	78 92	63.81	88.81	72.	98.7	80.28
[illegible]	59.46	49.65	69.38	58.27	79 31	66.89	89.24	75.52	99.17	87.82

…e pressures in the table are all gauge-pressures except those in italics, …h are absolute pressures (above a vacuum).

Straight-line Air-compressors, Ingersoll-Sergeant Rock-drill Co.

…eter …am- …der, …es.	Diameter of Air-cylinder, inches.	Length of Stroke, inches.	No. of Revolutions per minute.	Piston Speed in feet per minute.	Cubic Feet Free Air per minute (Theoretical).	Horse-power of Boiler required.
[illegible]	4¼	10	175	291	28	6
[illegible]	5¼	10	175	291	42	8
[illegible]	6¼	12	160	320	66	10
[illegible]	7¼	12	160	320	91	12
[illegible]	8¼	12	160	320	117	15
[illegible]	9¼	12	160	320	148	20
[illegible]	10¼	14	155	361	207	30
[illegible]	12¼	14	155	361	295	40
[illegible]	14¼	18	120	360	398	55
[illegible]	16¼	18	120	360	518	70
[illegible]	18¼	24	94	376	683	100
[illegible]	20¼	24	94	376	840	130
[illegible]	22¼	30	75	375	1011	155
[illegible]	24¼	30	75	375	1202	200

The same sizes are made to be driven by belt or gearing.

…mpressors at High Altitudes.—Cubic feet of compressed air …ered by air-compressors at high altitudes, expressed as a percentage of …ir delivered at the sea-level.

…ude above Sea-level, feet.	0	1000	2000	3000	4000	5000	6000	7000	8000	9000	10000
…elivered, per cent.	100	97	94	91	89	86	84	81	78	[illegible]	[illegible]

Standard Air-compressors driven by Steam.

(Norwalk Iron Works Co.)

In the following list the large air-cylinder gives the capacity of the [illegible] chine. For actual capacity, allowance of 10 per cent may be made [illegible] contingencies. The small piston only encounters the pressure of the [illegible] compression.

Diameter of Air-cylinder.	Length of Stroke.	Diameter of Compressing Cylinder.	Diameter of Steam-cylinder.	Revolutions or Double Strokes per minute.	Theoretical Capacity, cubic feet per minute, Free Air.	Steam-pipe.	Exhaust-pipe.	Air-pipe.	Water-pipe.
8	10	5	8	200	116	2	2½	2	½
10	12	6¾	10	190	207	2½	3	2½	¾
14	16	9½	14	150	427	3	4	4	1
20	24	13½	20	110	960	5	6	5	1¼
26	30	17½	24	90	1659	6	8	6	1¼
32	36	21½	30	80	2686	7	10	8	1½

Double-compound Compressors.

(Norwalk Iron Works Co.)

Diameter Air-cylinder.	Length of Stroke.	Diameter of— Compressing cylinder.	High-pressure Steam-cylinder.	Low-pressure Steam-cylinder.	Steam-pipe.	Revolutions per minute.
10	12	5	7½	12	2	190
12	12	5	7½	12	2	190
14	16	9½	10	16	2½	150
16	16	9½	10	16	2½	150
20	20	13½	14	22	3	120
20	24	13½	14	22	3	110
22	24	13½	14	22	3	110
26	30	17½	18	28	4½	90
28	30	17½	18	28	4½	90
32	36	21½	22	35	6	80

Mountain or High-altitude Compressors.

(Norwalk Iron Works Co.)

Diameter Air-cylinder.	Length of Stroke.	Diameter of Compressing Cylinder.	Diameter of Steam-cylinder.	Revolutions per minute.	At Sea-level. Capacity, cubic feet.	At Sea-level. Horse-power.	At 2000 feet. Capacity.	At 2000 feet. Horse-power.	At 6000 feet. Capacity.	At 6000 feet. Horse-power.	At 10,000 feet. Capacity.
12	12	7	10	190	298	35	280	34	244	32	214
16	16	9½	14	150	558	70	524	68	462	64	405
20	20	13½	18	120	872	110	819	107	722	100	[illegible]
22	24	13½	20	110	1160	145	1090	140	960	132	841
26	30	17½	24	90	1659	215	1560	207	1373	195	1200

The delivery and power of the compressors decrease as the height [illegible] As the capacity decreases in a greater ratio than the po[illegible] [illegible]ary to compress, it follows that operations at a high altitude are [illegible] [illegible]ve than at sea-level. At 10,000 feet this extra expense amounts [illegible] per cent.

Rand Drill Co.'s Air-compressors.

Dimensions of Air-cylinders in inches.	Revolutions per minute.	Theoretical Volume of Air delivered in cubic feet per minute, at Sea-level.						
		Free.	Compressed to a Gauge-pressure of—					
			10 lbs.	20 lbs.	40 lbs.	60 lbs.	80 lbs.	100 lbs.
10 × 16 S*..	100	145.44	86.56	61.61	39.08	28.62	22.57	18.64
10 × 16 D*..	100	290.88	173.12	123.23	78.17	57.24	45.15	37.28
14 × 22 S...	85	333.20	198.31	141.10	89.51	65.54	51.93	42.67
14 × 22 D...	85	666.40	396.61	282.20	179.01	131.07	103.86	85.34
16½ × 30 S.	75	556.83	331.39	235.89	149.64	109.57	86.43	71.36
16½ × 30 D.	75	1113.66	662.79	471.79	299.28	219.15	172.86	142.72
18 × 30 S...	75	662.68	394.39	280.73	178.08	130.40	102.86	84.92
18 × 30 D...	75	1325.36	788.78	561.46	356.17	260.81	205.72	169.84
20 × 48 S...	50	872.66	519.36	369.69	234.51	171.72	135.46	111.84
20 × 48 D...	50	1745.32	1038.72	739.38	469.03	343.45	270.92	223.68
28 × 48 S...	40	1368.34	814.36	579.67	367.72	269.27	212.40	175.36
28 × 48 D...	40	2736.68	1628.71	1159.34	735.45	538.54	424.80	350.73
32 × 48 S...	40	1787.22	1063.65	757.12	480.29	351.70	277.42	229.05
32 × 48 D...	40	3574.44	2127.30	1514.24	960.58	703.40	554.85	458.10
32 × 60 S...	35	1954.77	1163.37	828.10	525.32	384.67	303.43	250.52
32 × 60 D...	35	3909.55	2326.73	1656.20	1050.63	769.34	606.86	501.05
36 × 60 S..	30	2120.61	1262.07	898.35	572.07	417.72	329.16	272.82
36 × 60 D...	30	4241.22	2524.14	1796.70	1144.14	835.44	658.32	545.64
8 × 12.......	120	83.78	49.86	35.49	22.51	16.49	13.00	10.74
10 × 14.......	110	139.95	83.27	59.29	37.62	27.50	21.72	17.94
12 × 16.......	100	209.44	124.65	88.73	56.28	41.22	32.51	26.66
14 × 22.......	95	372.40	221.64	157.70	100.04	73.25	58.04	47.69
16 × 24.......	90	502.66	299.15	212.94	135.08	98.92	78.03	64.42
17½ × 24 ...	90	601.29	357.85	254.95	161.60	118.33	93.33	77.06
20 × 30......	80	872.67	519.36	369.69	234.52	171.73	135.46	111.84

* S, Single; D, Duplex.

tical Results with Compressed Air.—*Compressed-air at the Chapin Mines, Iron Mountain, Mich.*—These mines are three om the falls which supply the power. There are four turbines at one of 1000 horse-power and three of 900 horse-power each. The is 60 pounds at 60° Fahr. Each turbine runs a pair of compressors. e to the mines is 24 inches in diameter. The power is applied at the o Corliss engines, running pumps, hoists, etc., and direct to rock-

made in 1888 gave 1430.27 horse-power at the compressors, and 390.17 ower as the sum of the horse-power of the engines at the mines. re, only 27% of the power generated was recovered at the mines. ludes the loss due to leakage and the loss of energy in heat, but not tion in the engines or compressors. (F. A. Pocock, Trans. A. I. M. E.,

Saunders (*Jour. F. I.* 1892) says: "There is not a properly designed ssed-air installation in operation to-day that loses over 5% by trans- alone. The question is altogether one of the size of pipe; and if e is large enough, the friction loss is a small item. The largest com- l-air power plant in America is that at the Chapin Mines in Michigan, power is generated at Quinnesec Falls, and transmitted three miles. is not an economical plant, but the loss of pressure as shown by the is only 2 lbs., and this is the loss which may be laid strictly to trans- n.

e loss of power in common practice, where compressed air is used to machinery in mines and tunnels, is about 70%. I refer to cases where m American air-compressors are used, and where the ... trans- far enough to lose its heat of compression and is exh...

many industries becomes possible, while in cases where it is necess have a constant supply of cold air economy ceases to be a matter of first importance.

The following table shows the results of tests of a small rotary engine for driving sewing-machines, and indicating about a tenth of a horse-p

TRIALS OF A SMALL ROTARY RIEDINGER ENGINE.

Numbers of trials	I.	II.
Initial air-pressure, lbs. per sq. in	86	[illegible]
Initial temperature, deg. Fahr	54°	[illegible]
Ft.-lbs. per sec., measured on the brake	51.63	[illegible]
Revolutions per minute	384	[illegible]
Consumption of air per 1 horse-power per hour	1377	[illegible]

The following table shows the results obtained with a one-half h power variable expansive Riedinger rotary engine. These trials rep the best practice that has been obtained up to the present time (189 volumes of air were in all cases taken at atmospheric pressure:

TRIALS OF A .5-HORSE-POWER RIEDINGER ROTARY ENGINE.

Numbers of trials	I.	II.	III.
Initial pressure of air, lbs. per sq. in	54	69.7	85
" temperature of air, deg. Fahr	338	356	388
Final " " " "	77	68	...
Revolutions per minute	335	350	310
Ft.-lbs. per second, measured on brake	271	477	376
Consumption of air per horse-power per hour	883	791	900

Trials made with an old single-cylinder 80-horse-power Farcot ste gine, indicating 72 horse-power, gave a consumption of air per brake power as low as 465 cu. ft. per hour. The temperature of admissio 320° F., and of exhaust 95° F.

Prof. Elliott gives the following as typical results of efficiency for v systems of compressors and air-motors:

Simple compressor and simple motor, efficiency ....
Compound compressor and simple motor, " ....
" " " compound motor, efficiency ....
Triple compressor and triple motor, " ....

The efficiency is the ratio of the indicated horse-power in the motor ders to the indicated horse-power in the steam-cylinders of the comp The pressure assumed is 6 atmospheres absolute, and the losses are to those found in Paris over a distance of 4 miles.

Summary of Efficiencies of Compressed-air Transmi at Paris, between the Central Station at St. Fargea a 10-horse-power Motor Working with Pressure duced to 4½ Atmospheres.

(The figures below correspond to mean results of two experiments co two heated.)

1 indicated horse-power at central station gives 0.845 indicated horse in compressors, and corresponds to the compression of 348 cubic feet per hour from atmospheric pressure to 6 atmospheres absolute. (The of this air is about 25 pounds.)

0.845 indicated horse-power in compressors delivers as much air as 0.52 indicated horse-power in adiabatic expansion after it has fallen perature to the normal temperature of the mains.

The fall of pressure in mains between central station and Paris (say metres) reduces the possibility of work from 0.52 to 0.51 indicated power.

The further fall of pressure through the reducing valve to 4½ atmos (absolute) reduces the possibility of work from 0.51 to 0.50.

Incomplete expansion, wire-drawing, and other such causes redu actual indicated horse-power of the motor from 0.50 to 0.39.

By heating the air before it enters the motor to about 320° F., the indicated horse-power at the motor is, however, increased to 0.54. Th of gain by [illegible] the air is, therefore, $\frac{0.54}{0.39} = 1.38$.

additional heat is supplied by the combustion of about 0.39
er indicated horse-power per hour, and if this be taken into
indicated efficiency of the whole process becomes 0.47

old air the work spent in driving the motor itself reduces
se-power from 0.39 to 0.26.
eated air the work spent in driving the motor itself reduces
se-power from 0.54 to 0.44.
the efficiencies is as follows:
ain engines 0.845.
mpressors 0.52 ÷ 0.845 = 0.61.
ansmission through mains 0.51 ÷ 0.52 = 0.98.
ducing valve 0.50 ÷ 0.51 = 0.98.
fficiency of the mains and reducing valve between 5 and
is thus 0.98 × 0.98 = 0.96. If the reduction had been to 4,
eres, the corresponding efficiencies would have been 0.93,
ectively.
ncy of motor 0.39 ÷ 0.50 = 0.78.
ency of whole process with cold air 0.39. Apparent indi-
f whole process with heated air 0.54.
fficiency of whole process with heated air 0.47.
iency of motor, cold, 0.67.
iency of motor, hot, 0.81.
mpressed air in Paris is used for driving motors, but the
se is of the most varied kind. A list of motors driven from
on shows 225 installations, nearly all motors working at
ver to 50 horse-power, and the great majority of them more
vay from the station. The new station at Quai de la Gare
an the one at St. Fargeau. Experiments on the Riedler
at Paris, made in December, 1891, to determine the ratio
ated work done by the air-pistons and the indicated work
nders, showed a ratio of 0.8997. The compressors are driven
pansion Corliss engines of 2000 horse-power each.

ated by Compressed Air.—The *Iron Age*, March 2,
e shops of the Wuerpel Switch and Signal Co., East St. Louis,
s of which are operated by compressed air, each of the
ng its own air engine, and the smaller tools being belted
iven by an air engine. Power is supplied by a compound
d at 55 horse-power. The air engines are of the Kriebel
2 to 8 horse-power.

Postal Transmission.—A paper by A. Falkenau,
Philadelphia, April 1894, entitled the "First United States
System," gives a description of the system used in London
at recently introduced in Philadelphia between the main
substation. In London the tubes are 2¼ and 3 inch lead
t-iron pipes for protection. The carriers used in 2¼-inch
inches diameter, the remaining space being taken up by
rs are despatched singly. First, vacuum alone was used;
d compressed air. The tubes used in the Continental cities
rought iron, the Paris tubes being 2½ inches diameter.
ers are despatched in trains of six to ten, propelled by a
delphia the size of tube adopted is 6⅛ inches, the tubes
n bored to size. The lengths of the outgoing and return
et each. The pressure at the main station is 7 lbs., at the
, and at the end of the return pipe atmospheric pressure.
has two air-cylinders 18 × 24 in. Each carrier holds about
100 to 150 are taken as an average. Eight carriers may be
minute, giving a delivery of 48,000 to 72,000 letters per hour.
d in transmission is about 57 seconds.

rski Compressed-air Tramway at Berne,
(*Eng'g News*, April 20, 1893.)—The Mekarski system has
in Berne, Switzerland, on a line about two miles long, with
3.7% and 5.2%. A special feature of the Mekarski system is
e air, to maintain it at a constant temperature, by passing
heated water at 330° F. The air thus becomes saturated
ch subsequently partly condenses, its latent heat being
expanding air. The pressure in the car reservoirs

constructed like an ordinary steam tramway loco would be useless to make the vanes of the fan of a greater width inlet opening can freely supply. On the proportion of the length of the vane and the diameter of the inlet opening rest the three portant points, viz., *quantity* and *density* of air, and expenditure

In the 14-inch blade the tip has a velocity 2.6 times greater heel; and, by the laws of centrifugal force, the air will have a d times greater at the tip of the blade than that at the heel. The enter on the heel with a density higher than that of the atmosphe its passage along the vane it becomes compressed in proport centrifugal force. The greater the length of the vane, the great the difference of the centrifugal force between the heel and the blade; consequently the greater the density of the air.

Reasoning from these experiments, Mr. Buckle recommends fo erence the following proportions for the construction of the fan:

1. Let the width of the vanes be one fourth of the diameter; diameter of the inlet openings in the sides of the fan-chest be on diameter of the fan; 3. Let the length of the vanes be one fou diameter of the fan.

In adopting this mode of construction, the area of the inlet o the sides of the fan-chest will be the same as the circumference of the blade, multiplied by its width; or the same area as described by the heel of the blade.

Best Proportions of Fans. (Buckle.)

PRESSURE FROM 3 OUNCES TO 6 OUNCES PER SQUARE INCH; OR 5. TO 10.4 INCHES OF WATER.

Diameter of Fan.		Vanes.				Diameter of Inlet Openings.		Diameter of Fan.		Vanes.			
		Width.		Length.						Width.		Length	
ft.	ins.	ft.	ins.	ft.	ins.	ft.	ins.	ft.	ins.	ft.	ins.	ft.	ins.
3	0	0	9	0	9	1	6	4	6	1	1½	1	1½
3	6	0	10¼	0	10½	1	9	5	0	1	3	1	3
4	0	1	0	1	0	2	0	6	0	1	6	1	6

PRESSURE FROM 6 OUNCES TO 9 OUNCES PER SQUARE INCH, AND OR 10.4 INCHES TO 15.6 INCHES OF WATER.

3	0	0	7	1	0	1	0	4	6	0	10½	1	4½
3	6	0	8½	1	1½	1	3	5	0	1	0	1	6
4	0	0	9½	1	3½	1	6	6	0	1	2	1	10

The dimensions of the above tables are not laid down as prescr but as approximations obtained from the best results in practice

Experiments were also made with reference to the admission the transit or outlet pipe. By a slide the width of the opening int was varied from 12 to 4 inches. The object of this was to pro opening to the quantity of air required, and thereby to lessen necessary to drive the fan. It was found that the less this openi provided we produce sufficient blast, the less noise will procee fan; and by making the tops of this opening level with the tips o the column of air has little or no reaction on the vanes.

The number of blades may be 4 or 6. The case is made of th an arithmetical spiral, widening the space between the case and ing blades, circumferentially, from the origin to the opening for

The following rules deduced from experiments are given in treatise on Casting and Founding:

The fan-case should be an arithmetical spiral to the extent of of the blade at least.

The diameter of the tips of the blades should be about double th *of the hole in the* centre; the width to be about two thirds of th *the tips of* ... The velocity of the tips of the blades shoul than the velocity due to the air at the pressure required, say one more velocity.

me cases, two fans mounted on one shaft would be more useful than de one, as in such an arrangement twice the area of inlet opening is ed as compared with a single wide fan. Such an arrangement may pted where occasionally half the full quantity of air is required, as them may be put out of gear, thus saving power.

ssure due to Velocity of the Fan-blades.—"By increas- number of revolutions of the fan the head or pressure is increased, being that the total head produced is equal (in centrifugal fans) to the height due to the velocity of the extremities of the blades, or approximately in practice" (W. P. Trowbridge, Trans. A. S. M. E., .) This law is analogous to that of the pressure of a jet striking a surface. T. Hawksley, Proc. Inst. M. E., 1882, vol. lxix., says: "The re of a fluid striking a plane surface perpendicularly and then escap- right angles to its original path is that due to twice the height h due locity."

discussion of this question, showing that it is an error to take the re as equal to a column of air of the height $h = v^2 \div 2g$, see Wolff on mills, p. 17.)

kle says: "From the experiments it further appears that the velocity tips of the fan is equal to nine tenths of the velocity a body would re in falling the height of a homogeneous column of air equivalent to ensity." D. K. Clark (R. T. & D., p. 924), paraphrasing Buckle, appar says: "It further appears that the pressure generated at the circum ce is one ninth greater than that which is due to the actual circumfer- l velocity of the fan." The two statements, however, are not in ony, for if $v = 0.9\sqrt{2gH}$, $H = \frac{v^2}{0.81 \times 2g} = 1.234\frac{v^2}{2g}$ and not $1\frac{1}{9}\frac{v^2}{2g}$.

we take the pressure as that equal to a head or column of air of twice eight due the velocity, as is correctly stated by Trowbridge the para- cal statements of Buckle and Clark—which would indicate that the al pressure is greater than the theoretical—are explained, and the ula becomes $H = .617\frac{v^2}{g}$ and $v = 1.273\sqrt{gH} = 0.9\sqrt{2gH}$, in which H head of a column producing the pressure, which is equal to twice the retical head due the velocity of a falling body $\left(\text{or } h = \frac{v^2}{2g}\right)$, multiplied he coefficient .617. The difference between 1 and this coefficient ex- es the loss of pressure due to friction, to the fact that the inner por- s of the blade have a smaller velocity than the outer edge, and probably ther causes. The coefficient 1.273 means that the tip of the blade must iven a velocity 1.273 times that theoretically required to produce the l H.

convert the head H expressed in feet to pressure in lbs. per sq. in. tiply it by the weight of a cubic foot of air at the pressure and tempera- of the air expelled from the fan (about .08 lb. usually) and divide by Multiply this by 16 to obtain pressure in ounces per sq. in. or by 2.035 btain inches of mercury, or by 27.71 to obtain pressure in inches of r column. Taking .08 as the weight of a cubic foot of air,

p lbs. per sq. in. $= .00001066v^2$; $v = 310\sqrt{p}$ nearly;
p_1 ounces per sq. in. $= .0001706v^2$; $v = 80\sqrt{p_1}$ "
p_2 inches of mercury $= .00002169v^2$; $v = 220\sqrt{p_2}$ "
p_3 inches of water $= .0002954v^2$; $v = 60\sqrt{p_3}$ "

hich v = velocity of tips of blades in feet per second.

sting the above formula by the experiment of Buckle with the vane ches long, quoted above, we have $p = .00001066v^2 = 9.56$ oz. The ex- ment gave 9.1 oz.

sting it by the experiment of H. I. Snell, given below, in which the mferential speed was about 150 ft. per second, we obtain 3.85 ounces, the experiment gave from 2.38 to 3.50 ounces, according to the amount ening for discharge. The numerical coefficients of the above formulæ all based on Buckle's *statement that the velocity of the tips of the fan ual to nine tenths of the velocity a body would* acquire in falling the

Experiments were undertaken for the purpose of showing the results obtained by running the same fan at different speeds with the discharge-opening the same throughout the series.

The discharge-pipe was a conical tube 8½ inches inside diameter at end, having an area of 56.74, which is 7% larger than 53 sq. inches; then 53 square inches, equal to .368 square feet, is called the area of discharge, that is the practical area by which the volume of air is computed.

Experiments on a Fan with Constant Discharge-opening and Varying Speed.—The first four columns are given by Snell, the others are calculated by the author.

Revs. per min.	Pressure in ounces, p	Vol. of Air in cu. ft. per minute, V.	Horse-power.	Velocity of Tips of Blades, ft. per sec.	Velocity due Pressure from Formula $v = 80\sqrt{p}$.	Coefficient of Formula $v = x\sqrt{p}$ from Experiment.	Velocity of Air per minute in Efflux Pipe, $V \div .368$.	Theoretical Horse-power.
600	.50	1336	.25	60.2	56.6	85.1	3,630	.18
800	.88	1787	.70	80 3	75.0	85.6	4,856	.42
1000	1.38	2245	1.35	100.4	94.	85.4	6,100	.84
1200	2.00	2712	2.20	120.4	113.	85.1	7,370	1.47
1400	2.75	3177	3.45	140.5	133.	84.8	8,633	2.28
1600	3.80	3670	5.10	100.6	156.	82.4	9,973	3.8
1800	4.80	4172	8.00	180.6	175.	82.4	11,337	5.4
2000	5.95	4674	11.40	200.7	195.	85.6	12,701	7.3

Mr. Snell has not found any practical difference between the efficiency of blowers with curved blades and those with straight radial ones

From these experiments, says Mr. Snell, it appears that we may expect to receive back 65% to 75% of the power expended, and no more.

The great amount of power often used to run a fan is not due to the fan itself, but to the method of selecting, erecting, and piping it.

(For opinions on the relative merits of fans and positive rotary blowers see discussion of Mr. Snell's paper, Trans. A. S. M. E., ix. 66, etc.)

Comparative Efficiency of Fans and Positive Blowers (H. M. Howe, Trans. A. I. M. E., x. 482.)—Experiments with fans and positive (Baker) blowers working at moderately low pressures, under 20 ounces, show that they work more efficiently at a given pressure when delivering large volumes (*i e.*, when working nearly up to their maximum capacity) than when delivering comparatively small volumes. Therefore, when great variations in the quantity and pressure of blast required are liable to arise, the highest efficiency would be obtained by having a number of blowers, and driving them up to their full capacity, and regulating the amount of blast by altering the number of blowers at work, instead of having one or two very large blowers and regulating the amount of blast by the speed of the blowers.

There appears to be little difference between the efficiency of fans and Baker blowers when each works under favorable conditions as regards quantity of work, and when each is in good order.

For a given speed of fan, any diminution in the size of the blast-orifice decreases the consumption of power and at the same time raises the pressure of the blast; but it increases the consumption of power per unit of volume for a given pressure of blast. When the orifice has been reduced to the normal size for any given fan, further diminishing it causes but slight elevation of the blast pressure; and, when the orifice becomes comparatively small, further diminishing it causes no sensible elevation of the blast pressure, which remains practically constant, even when the orifice is entirely closed.

Many of the failures of fans have been due to too low speed, to too small *pulleys*, to improper fastening of belts, or to the belts being too nearly vertical; *in brief*, to bad mechanical arrangement, rather than to inherent defects in the principles of the machine.

eral fans are used, it is probably essential to high efficiency to pro-eparate blast pipe for each (at least if the fans are of different size d), while any number of positive blowers may deliver into the same thout lowering their efficiency.

Capacity of Fans and Blowers.

ollowing tables show the guaranteed air-supply and air-removal of forms of blowers and exhaust fans. The figures given are often d in practice, especially when the blowers and fans are driven at speeds than stated. The ratings, particularly of the blowers, are hose generally given in catalogues, but it was the desire to present nservative and assured practice. (A. R. Wolff on Ventilation.)

TY OF AIR SUPPLIED TO BUILDINGS BY BLOWERS OF VARIOUS SIZES.

Ordinary Number of Revs. per min.	Horse-power to Drive Blower.	Capacity cu. ft. per min. against a Pressure of 1 ounce per sq. in	Diameter of Wheel in feet.	Ordinary Number of Revs. per min.	Horse-power to Drive Blower.	Capacity cu. ft. per min. against a Pressure of 1 ounce per sq. in.
350	6.	10,635	9	175	29	56,800
325	9.4	17,000	10	160	35.5	70,340
275	13.5	29,618	12	130	49 5	102,000
230	18.4	42,700	14	110	66	139,000
200	24	46,000	15	100	77	160,000

resistance exceeds the pressure of one ounce per square inch. of ble, the capacity of the blower will be correspondingly decreased, r increased, and allowance for this must be made when the distrib-cts are small, of excessive length, and contain many contractions ds.

Y OF AIR MOVED BY AN APPROVED FORM OF EXHAUST FAN, THE N DISCHARGING DIRECTLY FROM ROOM INTO THE ATMOSPHERE.

Ordinary Number of Revs. per min.	Horse-power to Drive Fan.	Capacity in cu. ft. per min.	Diameter of Wheel in feet.	Ordinary Number of Revs. per min.	Horse-power to Drive Fan.	Capacity in cu. ft. per min.
600	0.50	5,000	4.0	475	3.50	28,000
550	0.75	8,000	5.0	350	4.50	35,000
500	1.00	12,000	6.0	300	7.00	50,000
500	2.50	20,000	7.0	250	9.00	80,000

pacity of exhaust fans here stated, and the horse-power to drive e for free exhaust from room into atmosphere. The capacity de-nd the horse-power increases materially as the resistance, resulting gths, smallness and bends of ducts, enters as a factor. The differ-ressures in the two tables is the main cause of variation in the re-records. The fan referred to in the second table could not be used igh a resistance as one ounce per square inch, the rated resistance owers.

CENTRIFUGAL FANS.

Pressures, Velocities, Volume of Air, Horse-P Required, etc. (B. F. Sturtevant Co.)

Pressure per sq. in. in ounces, from ¼ to 20 ounces; which includes the strongest blast found on any cupola in this country.	Velocity in feet per minute of Air (at 50°F.) escaping into open air through any shaped hole from any pipe or reservoir in which the Air is compressed.	Cubic feet of Air per minute (at 50° F.), which may be discharged through a proper shaped mouth-piece, the diameter of which must be 1.362 inches, the area being 1.07 square inches.	Actual H. P. contained in the blast discharged through the mouth-piece described in column 3.	*Cubic feet of Air per minute that may be discharged with one H. P., no allowance being made for friction in the blast-machine (whatever power that friction amounts to must be added). It makes no difference how the Air is discharged, provided the pressure is steady, the same as given to the first column.	† Number of mouth-pieces de-
1	2	3	4	5	
¼	2584.80	17.944	0.001224	14662.76	
½	3657.60	25.400	0.003463	7333.70	
¾	4482.00	31.124	0.005659	4889.11	
1	5175.00	35.93	0.0098	3666.62	
2	7338.24	50.96	0.0278	1833.00	
3	9006.42	62.54	0.0512	1222.30	
4	10421.58	72.37	0.0789	916.27	
5	11676.00	81.08	0.1106	733.39	
6	12817.08	89.01	0.1456	611.10	
7	13872.72	96.34	0.1839	523.81	
8	14861.16	103.20	0.2251	458.43	
9	15795.06	109.69	0.2692	407.42	
10	16683.51	115.86	0.3160	366.69	
11	17533.50	121.76	0.3652	333.40	
12	18350.34	127.43	0.4170	305.56	
13	19138.26	132.90	0.4712	282.05	
14	19900.68	138.20	0.5277	261.91	
15	20640.48	143.34	0.5864	244.44	
16	21360.00	148.33	0.6473	229.17	
17	22060.80	153.26	0.7103	215.77	
18	22745.40	157.96	0.7754	203.71	
19	23415.00	162.60	0.8426	192.98	
20	24070.80	167.16	0.9118	183.33	

* Always give the wind a good wide opening into the furnace … see by this table how much more wind can be discharged with one … low pressure than at high.

This table shows the great advantage of large tuyeres, large pip… blower, and slow speed when the nature of the work will admit.

† Number of forges driven with 1.2 H. P. with Sturtevant blower.

Caution in Regard to Use of Fan and Blower T… Many engineers report that manufacturers' tables overrate the … of their fans and underestimate the horse-power required to driv… In some cases the complaints may be due to restricted air outl… and crooked pipes, slipping of belts, too small engines, etc.

Fans, and Steam-coils combined for the r System of Heating. (Buffalo Forge Co.)

gine, revs.	Capacity of Fan per Minute at 1 oz. Pressure.	Weight of Fan and Engine.	Floor-space of Fan and Engine. Inches.	Required H.P. to drive Fan.	Usual Size Heater in ft. of Pipe.	Required H.P. of Boiler.
0	8,740	1,200	49 × 38	3.1	1,000	12
5	11,000	1,525	51 × 45	4	1,200	15
0	15,280	1,700	52 × 50	4.5	1,600	20
0	19,900	2,200	52 × 56	6	2,000	25
0	25,900	2,450	59 × 74	7.2	2,500	30
0	32,500	2,700	62 × 84	9.1	3,000	35
0	39,300	3,200	69 × 94	11	3,500	42
5	49,161	3,900	79 × 104	13.5	4,000	48
0	57,720	4,500	83 × 111	15	4,500	54
0	81,120	5,300	87 × 133	20	5,000	62
5	101,250	6,000	92 × 148	22	6,000	72

evant Steel Pressure-blower, applied to Cupola Furnaces.

No. of sq. in. of Blast.	Cub. feet of Air per minute.	Speed.	Pressure in ounces of Blast.	Horse-power Required.	Power Saved by Reducing the Speed and Pressure of Blast.					
					Speed.	Oz. Press.	H. P.	Speed.	Oz. Press.	H. P.
*4	324	4135	5	0.5						
5.7	507	3756	6	1	3445	5	0.8	3100	4	0.6
8	768	3250	7	1.8	3000	6	1.5	2750	5	1.1
10.7	1102	3100	8	3	2900	7	2.5	2700	6	2.
14.2	1646	2900	10	5.5	2560	8	4.	2390	7	3.3
18.7	2375	2820	12	9.7	2550	10	7.4	2260	8	5.3
24.3	3353	2600	14	16.	2380	12	12.7	2150	10	9.4
32	4416	2270	14	22.	2100	12	16.7	1900	10	12.7
43	6364	2100	16	35.	1960	14	28.4	1800	12	22.5
60	8880	1815	16	48.	1700	14	39.6	1566	12	31.7

ch of blast is sufficient for one forge-fire, or 90 square pola furnaces.
n is regulated so as to give the pressure of blast stated in e inch.
are inches of blast " refers to the area of a proper shaped harging blast into the open air.
apacity per hour in pounds of iron is made up from an n a few of the best cupolas found, and is reliable in cases s are well constructed and driven with the greatest force the table.
he steel pressure-blower as applied to forge-fires, and for her patterns of blowers and exhausters, see catalogue of Co.
concerning Cupolas, see Foundry Practice.)

Blast-pipes for Pressure-blowers for Cupola naces and Forges. (B. F. Sturtevant Co.)

table has been constructed on this basis, namely: A e of ¼ oz. in the *process of transmission through a* *e as a standard, the increased friction due to le* *compensated for by an enlargement of the pipe* to keep the loss still at ½ oz. The quantities of air in the left-h[and column] of each division indicate the capacity of the given blower wh[en run] under pressures of 4, 8, 12, and 16 ozs. Thus a No. 6 Blower wi[ll deliver ...] cubic ft. of air, at 8 oz. pressure, through 50 ft. of 12¼-in. pipe [with a loss] of ½ oz. pressure. If it is desired to force the air 300 ft. without [increasing the] loss by friction, the pipe must be enlarged to 17¼ in. diameter.

Cubic Feet of Air transmitted per minute.	Lengths of Blast-pipe in Feet. 50	100	150	200	300
	Diameter in inches.				
Blower No. 1.					
360	5⅜	6¼	6¾	7¼	7⅞
515	6⅜	7⅛	7¾	8¼	8⅞
635	6¾	7¾	8⅛	9	9⅝
740	7¼	8¼	9	9½	10¼
Blower No. 2.					
504	6¼	7⅛	7¾	8¼	8⅞
721	7¼	8¼	9	9½	10¼
889	7⅞	9	9¾	10⅜	11
1036	8⅜	9½	10⅜	11	11¾
Blower No. 3.					
720	7¼	8¼	9	9½	10¼
1030	8¾	9½	10⅜	11	11¾
1270	9⅛	10¾	11¼	11⅞	12¾
1480	9⅝	11	12	12⅝	13½
Blower No. 4.					
1008	8¼	9⅜	10¼	10⅞	11⅝
1442	9½	10⅞	11⅞	12½	13⅜
1778	10⅜	11⅞	12⅞	13⅝	14⅝
2072	11	12⅝	13¾	14½	15½
Blower No. 5.					
1440	9½	10⅞	11⅞	12½	13⅜
2060	11	12⅝	13¾	14½	15½
2540	11⅞	13⅝	14⅞	15⅝	16⅞
2960	12¾	14½	15⅞	16⅝	18

Cubic Feet of Air transmitted per minute.	Lengths of Blast-pi[pe] 50	100	150
	Diameter in i[nches]		
Blower No. 6			
1872	10⅝	12⅛	13¼
2678	12¼	14	15⅛
3302	13¼	15½	16½
3848	14⅛	16⅛	17½
Blower No. 7			
2592	12	13¾	15
3708	13⅞	15⅞	17¼
4572	15⅛	17⅜	18⅞
5328	16	18½	20
Blower No. 8			
3312	13¼	15½	16½
4738	15¼	17⅝	19½
5842	16⅝	19⅛	20¾
6808	17⅝	20¼	22⅛
Blower No. 9			
4320	14¾	17	18⅝
6180	17	19½	21¼
7620	18¾	21⅛	23¼
8880	19½	22½	24½
Blower No. 1[0]			
5760	16½	19	20¾
8240	18⅞	21¾	23¾
10160	20⅝	23¾	25⅞
11840	22⅛	25¼	27½

gal Ventilators for Mines.—Of different appliances for nines various forms of centrifugal machines having proved their ve now almost completely replaced all others. Most if not all ines in use in this country are of this class, being either open-ns, or closed, with chimney and spiral casing, of a more or less ibal type. The theory of such machines has been demonstrated el Murgue in "Theories and Practices of Centrifugal Ventilating ranslated by A. L. Stevenson, and is discussed in a paper by R. is, Trans. A. I. M. E. xx. 637. From this paper the following for-ken:

rea in sq. ft. of an orifice in a thin plate, of such area that its resistance to the passage of a given quantity of air equals the resistance of the mine;
rifice in a thin plate of such area that its resistance to the passage of a given quantity of air equals that of the machine;
uantity of air passing in cubic feet per minute;
elocity of air passing through a in feet per second;
elocity of air passing through o in feet per second;
ead in feet air-column to produce velocity V;
ead in feet air-column to produce velocity V_0.

$$Q = 0.65aV; \quad V = \sqrt{2gh}; \quad Q = 0.65a\sqrt{2gh};$$

$$a = \frac{Q}{0.65\sqrt{2gh}} = \text{equivalent orifice of mine;}$$

g to water-gauge in inches and quantity in thousands of feet per

$$\frac{.403Q}{\sqrt{W.G.}}; \quad Q = 0.65oV_0; \quad V_0 = \sqrt{2gh_0}; \quad Q = 0.65o\sqrt{2gh_0};$$

$$o = \sqrt{\frac{Q^2}{0.65^2h_0 2g}} = \text{equivalent orifice of machine.}$$

retical depression which can be produced by any centrifugal ven-ouble that due to its tangential speed. The formula

$$H = \frac{T^2}{2g} - \frac{V^2}{2g},$$

T is the tangential speed, V the velocity of exit of the air from the ween the blades, and H the depression measured in feet of air-s an expression for the theoretical depression which can be pro-an uncovered ventilator; this reaches a maximum when the air blades without speed, that is, $V = 0$, and $H = T^2 \div 2g$.
he theoretical depression which can be produced by any uncovered is equal to the height due to its tangential speed, and one half-h can be produced by a covered ventilator with expanding

as the condition of the mine remains constant:
ume produced by any ventilator varies directly as the speed of

ression produced by any ventilator varies as the square of the otation.
same tangential speed with decreased resistance the quantity of ses and the depression diminishes.
owing table shows a few results, selected from Mr. Norris's paper, e range of efficiency which may be expected under different cir-es. Details of these and other fans, with diagrams of the results in the paper.

Experiments on Mine-ventilating Fans.

Fan.	Revolutions per Minute, Fan.	Periphery Speed, Feet per Min.	Cubic Feet Air per Minute.	Cubic Feet Air per Revolution	Cubical Contents of Fan-blades.	Cub. Feet Air per 100 Feet Periphery Motion.	Water-gauge, Inches.	Horse-power in Air.	Indicated Horse-power of Engine.	Efficiency Engine and Fan.
A	84	5517	236,684	2818	3040	4290	1.80	67.13	88.40	75.9
	100	6282	336,862	3369	3040	5393	2.50	132.70	155.43	85.4
	111	6973	347,396	3130	3040	5002	3.20	175.17	209.64	83.6
	123	7727	394,100	3204	3040	5100	3.60	223.56	295.21	75.7
B	100	6282	188,888	1889	1520	3007	1.40	41.67	97.99	42.5
	130	8167	274,876	2114	1520	3366	2.00	86.63	194.95	44.6
C	59	3702	59,587	1010	1520	1610	1.20	11.27	16.76	67.83
	83	5208	82,969	1000	1520	1593	2.15	27.86	48.54	57.38
D	40	3140	49,611	1240	3096	1580	0.87	6.80	13.82	49.2
	70	5495	137,760	1825	3096	2507	2.55	55.35	67.44	82.07
E	50	2749	147,232	2944	1522	5356	0.50	11.60	28.55	40.63
	69	3793	205,761	2982	1522	5451	1.00	32.42	45.98	70.50
	96	5278	299,600	3121	1522	5676	2.15	101.50	120.64	84.10
F	200	7540	133,198	666	746	1767	3.35	70.30	102.79	68.40
	200	7540	180,809	904	746	2398	3.05	86.89	129.07	67.80
	200	7540	209,150	1046	746	2774	2.80	92.50	150.08	61.70
G	10	785	28,896	2890	3022	3680	0.10	0.45	1.30	35.
	20	1570	57,120	2856	3022	3637	0.20	1.80	3.70	49.
	25	1962	66,640	2665	3022	3399	0.29	2.90	6.10	48.
	30	2355	73,080	2436	3022	3103	0.40	4.60	9.70	47.
	35	2747	94,080	2688	3022	3425	0.50	7.40	15.00	48.
	40	3140	112,000	2800	3022	3567	0.70	12.30	24.90	49.
	50	3925	132,700	2654	3022	3381	0.90	18.80	38.80	48.
	60	4710	173,600	2893	3022	3686	1.35	36.90	66.40	55.
	70	5495	203,280	2904	3022	3718	1.80	57.70	107.10	54.
	80	6280	222,320	2779	3022	3540	2.25	78.80	152.60	52.

Type of Fan.	Diam.	Width.	No. Inlets.	Diam. In[illegible]	
A. Guibal, double..................	20 ft.	6 ft.	4	8 ft.	10
B. Same, only left hand running.	20	6	4	8	10
C. Guibal........................	20	6	2	8	10
D. Guibal........................	25	8	1	11	6
E. Guibal, double................	17½	4	4	8	
F. Capell........................	12	10	2	7	
G. Guibal........................	25	8	1	12	

An examination of the detailed results of each test in Mr. Norris's shows a mass of contradictions from which it is exceedingly difficult to any satisfactory conclusions. The following, he states, appear to be or less warranted by some of the figures:

1. *Influence of the Condition of the Airways on the Fan.*—Mines varying equivalent orifices give air per 100 feet periphery-motion of within limits as follows, the quantity depending on the resistance o mine:

Equivalent Orifice.	Cu. Ft. Air per 100 ft. Periphery-speed.	Average.	Equivalent Orifice.	Cu. Ft. Air per 100 ft. Periphery-speed.
Under 20 sq. ft.	1100 to 1700	1300	60 to 70	3300 to 5100
20 to 30	1300 to 1800	1600	70 to 80	4000 to 4700
30 to 40	1500 to 2500	2100	80 to 90	3000 to 5600
40 to 50	2300 to 3500	2700	90 to 100	
50 to 60	2700 to 4800	3500	100 to 114	5200 to 6200

The influence of the mine on the efficiency of the fan does not seem *very clear.* Eight fans, with equivalent orifices over 50 square feet

r 70% ; four, with smaller equivalent mine-orifices, give about
es ; while, on the contrary, six fans, with equivalent orifices of
feet, give lower efficiencies, as do ten fans, all drawing from
all equivalent orifices.
m that, on the whole, large airways tend to assist somewhat
ge efficiency.
of the Diameter of the Fan.—This seems to be practically *nil*,
tage of large fans being in their greater width and the lower
of the engines.
of the Width of a Fan.—This appears to be small as regards
of the machine ; but the wider fans are, as a rule, exhausting

of Shape of Blades.—This appears, within reasonable limits,
lly *nil*. Thus, six fans with tips of blades curved forward,
h flat blades, and one with blades curved back to a tangent
mference, all give very high efficiencies—over 70%.
of the Shape of the Spiral Casing.—This appears to be con-
e shapes of spiral casing in use fall into two classes, the first
arge spiral, beginning at or near the point of cut-off, and the
ar casing reaching around three quarters of the circumference
h a short spiral reaching to the *evasée* chimney.
the first form of casing appear to give in almost every case
es.
ave a spiral belonging to the first class, but very much con-
only medium efficiencies. It seems probable that the proper
l casing would be one of such form that the air between each
could constantly and freely discharge into the space between
sing, the whole being swept along to the *evasée* chimney. This
a spiral beginning near the point of cut-off, enlarging by
easing increments to allow for the slowing of the air caused by
ainst the casing, and reaching the chimney with an area such
uld make its exit with its then existing speed—somewhat less
hery-speed of the fan.
of the Shutter.—This certainly appears to be an advantage, as
area can be regulated to suit the varying quantity of air given
d in this way re-entries can be prevented. It is not uncommon
less fans into the chimneys of which bits of paper may be
ch are drawn *into* the fan, make the circuit, and are again
This peculiarity has not been noticed with fans provided with

of the Speed at which a Fan is Run.—It is noticeable that
fans giving high efficiency were running at a rather high
ocity. The best speed seems to be between 5000 and 6000 feet

pear to reach a maximum efficiency at somewhere about the
nd to decrease rapidly in efficiency when this maximum point

n of Mr. Norris's paper, Mr. A. H. Storrs says: From the "cu-
volution" and "cubical contents of fan-blades," as given in the
l that the enclosed fans empty themselves from one half to
olution, while the open fans are emptied from one and three-
arly three times. This for fans of both types, on mines cover-
range of equivalent orifices. One open fan, on a very large
nptied nearly four times, while a closed fan, on a still larger
shows one and one-half times. For the open fans the "cubic
motion" is greater, in proportion to the fan width and equiv-
than for the enclosed type. Notwithstanding this apparently
e of the open fans, they show very low efficiencies.
ng the very large capacity of centrifugal fans to pass air, if
of the mine are made favorable, a 16-ft. diam. fan, 4 ft. 6 in.
volutions, passed 360,000 cu. ft. per min., and another, of same
slightly wider and with larger intake circles, passed 500,000 cu.
gauge in both instances being about ½ in.
says: The efficiency reported in some cases by Mr. Norris is
have ever been able to determine by experiment. My own ex-
corded in the Pennsylvania Mine Inspectors' Reports from 1875
t show more than *60% to 65%*.

DISK FANS.

Experiments made with a Blackman Disk Fan, diam., by Geo. A. Suter, to determine the volumes of air delivered u various conditions, and the power required; with calculations of effici and ratio of increase of power to increase of velocity, by G. H. Babc (Trans. A. S. M. E., vii. 547):

Rev. per min.	Cu. ft. of Air delivered per min., V.	Horse-power, HP.	Water-gauge, in., h.	Ratio of Increase of Speed.	Ratio of Increase of Delivery.	Ratio of Increase of Power.	Exponent x, $HP \propto V^x$.	Exponent y, $h \propto V^y$.	Efficiency
350	25,797	0.65	....	....	....	....	......	......	[illegible]
440	32,575	2.29	....	1.257	1.262	3.523	5.4	......	[illegible]
534	41,929	4.42	....	1.186	1.287	1.843	2.4	......	[illegible]
612	47,756	7.41	....	1.146	1.139	1.677	3.97	......	[illegible]
	For	series	....	1.749	1.851	11.140	4.	......	....
340	20,372	0.76	....	....	....	....	......	......	[illegible]
453	26,660	1.99	....	1.332	1.308	2.618	3.55	......	[illegible]
536	31,649	3.86	....	1.183	1.187	1.940	3.86	......	[illegible]
627	36,543	6.47	....	1.167	1.155	1.676	3.59	......	[illegible]
	For	series	....	1.761	1.794	8.513	3.63	......	....
340	9,983	1.12	0.28	....	....	....	......	......	[illegible]
430	13,017	3.17	0.47	1.265	1.304	2.837	3.93	1.95	[illegible]
534	17,018	6.07	0.75	1.242	1.307	1.915	2.25	1.74	[illegible]
570	18,649	8.46	0.87	1.068	1.096	1.394	3.63	1.60	[illegible]
	For	series	....	1.676	1.704	7.554	3.24	1.81	....
330	8,399	1.31	0.26	....	....	....	......	......	[illegible]
437	10,071	3.27	0.45	1.324	1.199	3.142	6.31	3.06	[illegible]
516	11,157	6.00	0.75	1.181	1.108	1.457	3.66	4.96	[illegible]
	For	series	....	1.563	1.329	4.580	5.35	3.72	....

Nature of the Experiments.—First Series: Drawing air through 30 f 48-in. diam. pipe on inlet side of the fan.

Second Series: Forcing air through 30 ft. of 48-in. diam. pipe on outlet of the fan.

Third Series: Drawing air through 30 ft. of 48-in. pipe on inlet side of fan—the pipe being obstructed by a diaphragm of cheese-cloth.

Fourth Series: Forcing air through 30 ft. of 48-in. pipe on outlet side o —the pipe being obstructed by a diaphragm of cheese-cloth.

Mr. Babcock says concerning these experiments: The first four ex ments are evidently the subject of some error, because the efficiency is as to prove on an average that the fan was a source of power sufficie overcome all losses and help drive the engine besides. The second seri less questionable, but still the efficiency in the first two experiments is la than might be expected. In the third and fourth series the resistance of cheese-cloth in the pipe reduces the efficiency largely, as would be expe In this case the value has been calculated from the height equivalent to water-pressure, rather than the actual velocity of the air.

This record of experiments made with the disk fan shows that this kin fan is not adapted for use where there is any material resistance to the of the air. In the centrifugal fan the power used is nearly proportion the amount of air moved under a given head, while in this fan the powe quired for the same number of revolutions of the fan increases very m rially with the resistance, notwithstanding the quantity of air moved is at *same time* considerably reduced. In fact, from the inspection of the t and fourth series of tests, it would appear that the power required is nearly the same for a given pressure, whether more or less air be m It would seem that the main advantage, if any, of the disk fan over th trifugal fan for slight resistances consists in the fact that the delive full area of the disk, while with centrifugal fans intended to move quantity of air the opening is much smaller.

be seen by columns 8 and 9 of the table that the power used in- much more rapidly than the cube of the velocity, as in centrifugal he different experiments do not agree with each other, but a general may be assumed as about the cube root of the eleventh power.

Feet of Air removed by Exhaust Disk-wheel per minute. (Buffalo Forge Co.)

r of	Diameter of Wheel.							
	24 Inch.	30 Inch.	36 Inch.	42 Inch.	48 Inch.	54 Inch.	60 Inch.	72 Inch.
	Amount of Air in cubic feet per minute.							
..	....	....	....	....	4,245	6,059	8,387	14,936
..	....	....	....	....	6,405	9,154	12,822	22,926
..	....	....	3,594	5,607	8,686	12,410	17,457	31,267
..	1,307	2,696	4,541	7,079	11,098	15,822	22,292	39,956
..	1,684	3,338	5,550	8,621	13,641	19,408	27,327	48,996
..	2,014	4,042	6,621	10,233	16,315	23,147	32,565	58,386
..	2,375	4,808	7,755	11,915	19,119	27,048	37,997	67,985
..	2,770	5,636	8,950	13,967	22,055	31,112	43,632	76,900
..	3,197	6,516	10,210	15,489	25,127	35,338	49,467	....
.	3,656	7,446	11,430	17,381	28,325	39,727	55,152	....
..	4,148	8,426	12,816	19,345	31,518	44,277	60,401	....
.	4,671	9,456	14,265	21,375	34,310	48,992	....	....
..	5,221	10,536	15,776	23,420	36,940	53,858	....	....

iency of Disk Fans.—Prof. A. B. W. Kennedy (*Industries*, Jan. made a series of tests on two disk fans, 2 and 3 ft. diameter, known Verity Silent Air-propeller. The principal results and conclusions densed below.

h case the efficiency of the fan, that is, the quantity of air delivered ctive horse-power, increases very rapidly as the speed diminishes, lower speeds are much more economical than higher ones. On the and, as the quantity of air delivered per revolution is very nearly t, the actual useful work done by the fan increases almost directly speed. Comparing the large and small fans with about the same ery, the former (running at a much lower speed, of course) is much e economical. Comparing the two fans running at the same speed, r, the smaller fan is very much the more economical. The delivery er revolution of fan is very nearly directly proportional to the area an's diameter.

ir delivered per minute by the 3-ft. fan is nearly $12.5R$ cubic feet the number of revolutions made by the fan per minute). For the the quantity is $5.7R$ cubic feet. For either of these or any other fans of which the area is A square feet, the delivery will be about cubic feet. Of course any change in the pitch of the blades might change these figures.

et H.P. taken up is not far from proportional to the square of the of revolutions above 100 per minute. Thus for the 3-ft. fan the net $\frac{(R-100)^2}{200{,}000}$, while for the 2-ft. fan the net H.P. is $\frac{(R-100)^2}{1{,}000{,}000}$.

enominators of these two fractions are very nearly proportional in- to the square of the fan areas or the fourth power of the fan diam- The net H.P. required to drive a fan of diameter D feet or area A feet, at a speed of R revolutions per minute, will therefore be ap- ately $\frac{D^4(R-100)^2}{17{,}000{,}000}$ *or* $\frac{A^2(R-100)^2}{10{,}400{,}000}$.

ft. fan was noiseless at all speeds. The 3-ft. fan was also noiseless *r 450 revolutions per minute.*

	Propeller, 2 ft. diam.			Propeller, 3 ft. diam.		
Speed of fan, revolutions per minute.	750	676	577	576	459	[illegible]
Net H.P. to drive fan and belt........	0.42	0.32	0.227	1.02	0.575	[illegible]
Cubic feet of air per minute.	4,183	3,830	3,410	7,400	5,800	[illegible]
Mean velocity of air in 3-ft. flue, feet per minute........................	593	543	482	1,046	830	[illegible]
Mean velocity of air in flue, same diameter as fan....................	1,330	1,220	1,085	......	......	......
Cu.ft.of air per min.per effective H.P.	9,980	11,970	15,000	7,250	10,070	[illegible]
Motion given to air per rev. of fan, ft.	1.77	1.81	1.88	1.82	1.79	[illegible]
Cubic feet of air per rev. of fan.......	5.58	5.66	5.90	12.8	12.6	[illegible]

POSITIVE ROTARY BLOWERS. (P. H. & F. M. Roots.)

Size number	¼	½	1	2	3	4	5	6
Cubic feet per revolution....	¾	1½	3	5	8	13	23	42
Revolutions per minute, Smith fires....	300 to 350	250 to 300	225 to 275	200 to 250	175 to 225	150 to 200	125 to 175	100 to 150
Furnishes blast for Smith fires....................	2 to 4	6 to 8	10 to 14	16 to 20	24 to 30	32 to 43	47 to 67	70 to 100
Revolutions per minute for cupola, melting iron.....	...	...	275 to 375	275 to 325	200 to 300	185 to 275	170 to 250	150 to 200
Size of cupola, inches, inside lining................	...	...	18 to 24	24 to 30	30 to 36	36 to 42	42 to 50	50 to 60
Will melt iron per hour, tons	...	...	1½ to 2	2½ to 3	3 to 4⅝	4⅞ to 7	8 to 12	12½ to 16⅝
Horse-power required........	1	2	3½	5½	8	11½	17¾	27

The amount of iron melted is based on 30,000 cubic feet of air per ton of iron. The horse-power is for maximum speed and a pressure of ¾ pound, ordinary cupola pressure. (See also Foundry Practice.)

BLOWING-ENGINES.

Blast-furnace Blowing-engines of the Variable Puppet-valve Cut-off Type. (Philada. Engineering Works.)

Diameter of Steam-cylinder.	Diameter of Blowing-cylinder.	Stroke.	Shop Weights, approximate.	Revolutions, ordinary speed.	Displacement of Piston per minute at ordinary speed.	Maximum Blast-pressure for Regular Work.
in.	in.	in.	pounds.		cubic feet.	lbs. per sq. in.
28	66	36	80,000	60	8,550	10
28	66	48	90,000	50	9,500	10
32	72	48	106,000	50	11,308	12
36	72	48	130,000	50	11,308	15
36	84	48	140,000	50	15,392	11
36	84	60	165,000	40	15,392	11
42	84	48	165,000	50	15,392	15
42	84	60	190,000	40	15,392	15
42	90	48	170,000	50	17,700	18
42	90	60	195,000	40	17,700	[illegible]
48	96	48	220,000	50	20,000	[illegible]
48	96	60	280,000	40	20,000	[illegible]

The blowing-engines of the country are usually very wasteful of [illegible] *by reason of wire-drawing valve-gear, and especially of slow piston* [illegible] *The latter is perhaps the greatest and the least recognized of* [illegible] *engine defects.* Almost any expense to increase the economy of [illegible] **engines is warranted.** (A. L. Holley, Trans. A. I. M. E., vol. iv. p. 8[illegible]) lculations of power, capacity, etc., of blowing-engines are the same for air-compressors. They are built without any provision for he air during compression. About 400 feet per minute is the usual peed for recent forms of engines, but with positive air-valves, which en introduced to some extent, this speed may be increased. The y of the engine, that is, the ratio of the I.H.P. of the air-cylinder to the steam-cylinder, is usually taken at 90 per cent, the losses by leakage, etc., being taken at 10 per cent.

STEAM-JET BLOWER AND EXHAUSTER.

wer and exhauster is made by L. Schutte & Co., Philadelphia, on ciple of the steam-jet ejector. The following is a table of capacities:

Quantity of Air per hour in cubic feet.	Diameter of Pipes in inches.		Size No.	Quantity of Air per hour in cubic feet.	Diameter of Pipes in inches.	
	Steam.	Air.			Steam.	Air.
1,000	1/2	1	5	30,000	2 1/2	5
2,000	3/4	1 1/2	6	36,000	2 1/2	6
4,000	1	2	7	42,000	3	6
6,000	1 1/4	2 1/2	8	48,000	3	7
12,000	1 1/2	3	9	54,000	3 1/2	7
18,000	2	3 1/2	10	60,000	3 1/2	8
24,000	2	4				

lmissible vacuum and counter pressure, for which the apparatus is ted, is up to a rarefaction of 20 inches of mercury, and a counter- up to one sixth of the steam-pressure.

ble of capacities is based on a steam-pressure of about 60 lbs., and r-pressure of about 8 lbs. With an increase of steam-pressure or of counter-pressure the capacity will largely increase.

er steam-jet blower is used for boiler-firing, ventilation, and similar s where a low counter-pressure or rarefaction meets the require-

lumes as given in the following table of capacities are under the ion of a steam-pressure of 45 lbs. and a counter-pressure of, say, of water:

Cubic eet of Air elivered r hour.	Diameter of Steam-pipe in inches.	Diameter in inches of—		Size No.	Cubic feet of Air delivered per hour	Diam. of Steam-pipe in inches.	Diameter in inches of—	
		Inlet	Disch.				Inlet.	Disch.
6,000	3/8	4	3	4	250,000	1	17	14
12,000	1/2	5	4	6	500,000	1 1/4	24	20
30,000	1/2	8	6	8	1,000,000	1 1/2	32	27
60,000	3/4	11	8	10	2,000,000	2	42	36
25,000	1	14	10					

Steam-jet as a Means for Ventilation.—Between 1810 the steam-jet was employed to a considerable extent for ventilat- lish collieries, and in 1852 a committee of the House of Commons that it was the most powerful and at the same time the cheapest for the ventilation of mines; but experiments made shortly after- roved that this opinion was erroneous, and that furnace ventilation than half as expensive, and in consequence the jet was soon aban- a permanent method of ventilation.

account of these experiments see Colliery Engineer, Feb. 1890. *however, is sometimes advantageously* used as a substitute, for *in the case of a fan standing* for repairs, or after an explosion, *urnace may not be* kept going, or in the case of the fan having *red useless.*

HEATING AND VENTILATION.

Ventilation. (A. R. Wolff, *Stevens Indicator*, April, 1890.)—The nlar impression that the impure air falls to the bottom of a crowded r is erroneous. There is a constant mingling of the fresh air admitted the impure air due to the law of diffusion of gases, to difference of tem ature, etc. The process of ventilation is one of dilution of the impur by the fresh, and a room is properly ventilated in the opinion of the hy ists when the dilution is such that the carbonic acid in the air does not ceed from 6 to 8 parts by volume in 10,000. Pure country air contains a 4 parts CO_2 in 10,000, and badly-ventilated quarters as high as 80 parts.

An ordinary man exhales 0.6 of a cubic foot of CO_2 per hour. New gas gives out 0.75 of a cubic foot of CO_2 for each cubic foot of gas b An ordinary lamp gives out 1 cu. ft. of CO_2 per hour. An ordinary ca gives out 0.3 cu. ft. per hour. One ordinary gaslight equals in vitia effect about 5½ men, an ordinary lamp 1⅔ men, and an ordinary cand man.

To determine the quantity of air to be supplied to the inmates of an lighted room, to dilute the air to a desired standard of purity, we can es lish equations as follows:

Let v = cubic feet of fresh air to be supplied per hour;
r = cubic feet of CO_2 in each 10,000 cu. ft. of the entering air:
R = cubic feet of CO_2 which each 10,000 cu. ft. of the air in the r may contain for proper health conditions;
n = number of persons in the room;
.6 = cubic feet of CO_2 exhaled by one man per hour.

Then $\frac{v \times r}{10,000} + .6n$ equals cubic feet of CO_2 communicated to the room ing one hour.

This value divided by v and multiplied by 10,000 gives the proportio CO_2 in 10,000 parts of the air in the room, and this should equal R, the dard of purity desired. Therefore

$$R = \frac{10,000\left[\frac{v \times r}{10,000} + .6n\right]}{v}, \quad \text{or } v = \frac{6000n}{R - r}. \quad . \quad . \quad . \quad .$$

If we place r at 4 and R at 6, $v = \frac{6000}{6-4}n = 3000n$, ...

or the quantity of air to be supplied per person is 3000 cubic feet per ho

If the original air in the room is of the purity of external air, and the c contents of the room is equal to 100 cu. ft. per inmate, only 3000 − 100 = cu. ft. of fresh air from without will have to be supplied the first ho keep the air within the standard purity of 6 parts of CO_2 in 10,000. I cubic contents of the room equals 200 cu. ft. per inmate, only 3000 − 200 = cu. ft. will have to be supplied the first hour to keep the air within standard purity, and so on.

Again, if we only desire to maintain a standard of purity of 8 part carbonic acid in 10,000, equation (1) gives as the required air-supply per

$v = \frac{6000}{8-4}n = 1500n$, or 1500 cu. ft. of fresh air per inmate per hour.

Cubic feet of air containing 4 parts of carbonic acid in 10,000 necessary person per hour to keep the air in room at the composition of

6	7	8	9	10	15	20	parts of carbonic ac 10,000.
3000	*2000*	*1500*	*1200*	*1000*	*545*	*375*	cubic feet.

If the original air in the room is of purity of external atmosphere (4 of carbonic acid in 10,000), the amount of air to be supplied the fir for given cubic spaces per inmate, to have given standards of p exceeded at the end of the hour is obtained from the following tab

ubic Feet of Space in Room per ndividual.	Proportion of Carbonic Acid in 10,000 Parts of the Air, not to be Exceeded at End of Hour.						
	6	7	8	9	10	15	20
	Cubic Feet of Air, of Composition 4 Parts of Carbonic Acid in 10,000, to be Supplied the First Hour.						
100	2900	1900	1400	1100	900	445	275
200	2800	1800	1300	1000	800	345	175
300	2700	1700	1200	900	700	245	75
400	2600	1600	1100	800	600	145	None
500	2500	1500	1000	700	500	45	......
600	2400	1400	900	600	400	None	....
700	2300	1300	800	500	300	....	....
800	2200	1200	700	400	200	....	....
900	2100	1100	600	300	100	....	....
1000	2000	1000	500	200	None	....	....
1500	1500	500	None	None	....	....	....
2000	1000	None	....	....	....	....	....
2500	500	....	....	....	....	....	....

t is exceptional that systematic ventilation supplies the 3000 cubic feet r inmate per hour, which adequate health considerations demand. Large ditoriums in which the cubic space per individual is great, and in which e atmosphere is thoroughly fresh before the rooms are occupied, and the cupancy is of two or three hours' duration, the systematic air-supply may reduced, and 2000 to 2500 cubic feet per inmate per hour is a satisfactory owance.

Hospitals where, on account of unhealthy excretions of various kinds, the -dilution must be largest, an air-supply of from 4000 to 6000 cubic feet per nate per hour should be provided, and this is actually secured in some spitals. A report dated March 15, 1882, by a commission appointed to amine the public schools of the District of Columbia, says:

"In each class-room not less than 15 square feet of floor-space should be otted to each pupil. In each class-room the window-space should not be s than one fourth the floor-space, and the distance of desk most remote m the window should not be more than one and a half times the height of e top of the window from the floor. The height of the class-room should ver exceed 14 feet. The provisions for ventilation should be such as to ovide for each person in a class-room not less than 30 cubic feet of fresh per minute (1800 per hour), which amount must be introduced and oroughly distributed without creating unpleasant draughts, or causing any o parts of the room to differ in temperature more than 2° Fahr., or the aximum temperature to exceed 70° Fahr."

When the air enters at or near the floor, it is desirable that the velocity of let should not exceed 2 feet per second, which means larger sizes of gister openings and flues than are usually obtainable, and much higher locities of inlet than two feet per second are the rule in practice. The locity of current into vent-flues can safely be as high as 6 or even 10 feet r second, without being disagreeably perceptible.

The entrance of fresh air into a room is co-incident with, or dependent on, e removal of an equal amount of air from the room. The ordinary means removal is the vertical vent-duct, rising to the top of the building. Some-nes reliance for the production of the current in this vent-duct is placed lely on the difference of temperature of the air in the room and that of e external atmosphere; sometimes a steam coil is placed within the flue ar its bottom to heat the *air within the duct*; sometimes steam pipes sers and *returns*) *run up the duct performing* the same functions; or steam *s within the flue, or exhaust fans, driven* by steam or electric power, act *ectly as exhausters; sometimes the heating* of the air in the flue is ac-*plished by gas-jets.*

draft of such a duct is caused by the difference of weight of the heated air in the duct, and a column of equal height and cross-sectic of weight of the external air.

Let d = density, or weight in pounds, of a cubic foot of the exter
Let d_1 = density, or weight in pounds, of a cubic foot of the h within the duct.
Let h = vertical height, in feet, of the vent-duct.
$h(d - d_1)$ = the pressure, in pounds per square foot, with which forced into and out of the vent-duct.

This pressure can be expressed in height of a column of the air o within the vent-duct, and evidently the height of such column pressure would be, $\frac{h(d - d_1)}{d_1}$. . . . .

Or, if t = absolute temperature of external air, and t_1 = absolute ature of the air in vent-duct in the form, then the pressure equals

$$\frac{h(t_1 - t)}{t}. \quad . \ . \ . \ . \ . \ . \ . \ . \ .$$

The theoretical velocity, in feet per second, with which the a travels through the vent-duct under this pressure is

$$v = \sqrt{\frac{2gh(t_1 - t)}{t}} = 8.02\sqrt{\frac{h\,(t_1 - t)}{t}}. \ . \ . \ .$$

The actual velocity will be considerably less than this, on accoun due to friction. This friction will vary with the form and cross- area of the vent-duct and its connections, and with the degree of ness of its interior surface. On this account, as well as to preven of air through crevices in the wall, tin lining of vent-flues is desira
The loss by friction may be estimated at approximately 50%, and s for the actual velocity of the air as it flows through the vent-duct:

$$v = \frac{1}{2}\sqrt{2gh\frac{(t_1 - t)}{t}}, \text{ or, approximately, } v = 4\sqrt{h\frac{(t_1 - t)}{t}}$$

If V = velocity of air in vent-duct, in feet per minute, and the ext be at 32° Fahr., since the absolute temperature on Fahrenheit sca thermometric temperature plus 459.4,

$$V = 240\sqrt{h\frac{(t_1 - t)}{491.4}}. \ . \ . \ . \ . \ . \ .$$

from which has been computed the following table:

Quantity of Air, in Cubic Feet, Discharged per through a Ventilating Duct, of which the Cro tional Area is One Square Foot (the External Te ture of Air being 32° Fahr.).

Height of Vent-duct in feet.	Excess of Temperature of Air in Vent-duct abov External Air.							
	5°	10°	15°	20°	25°	30°	50°	100°
10	77	108	133	153	171	188	242	34
15	94	133	162	188	210	230	297	41
20	108	153	188	217	242	265	342	48
25	121	171	210	242	271	297	383	541
30	133	188	230	265	297	325	419	59
35	143	203	248	286	320	351	453	640
40	153	217	265	306	342	375	484	65
45	162	230	282	325	363	396	514	[illegible]
50	171	242	297	342	383	419	541	[illegible]

Multiplying the figures in above table by 60 gives the cubic fee *charged* per hour per square foot of cross-section of vent-duc s-sectional area of vent-ducts we can find the total discharge; or sired air-removal, we can proportion the cross-sectional area of ets required.

icial Cooling of Air for Ventilation. (*Engineering* uly 7, 1892.)—A pound of coal used to make steam for a fairly effi- frigerating-machine can produce an actual cooling effect equal to oduced by the melting of 16 to 46 lbs. of ice, the amount varying conditions of working. Or, 855 heat-units per lb. of coal converted rk in the refrigerating plant (at the rate of 3 lbs. coal per horse- our) will abstract 2275 to 6545 heat-units of heat from the refriger- dy. If we allow 2000 cu. ft. of fresh air per hour per person as suffi- r fair ventilation, with the air at an initial temperature of 80° F., its per cubic foot will be .0736 lb.; hence the hourly supply per person gh 2000 × .0736 lb. = 147.2 lbs. To cool this 10°, the specific heat of g 0.238, will require the abstraction of 147.2 × 0.238 × 10 = 350 heat- r person per hour.

g the figures given for the refrigerating effect per pound of coal as ated, and the required abstraction of 350 heat-units per person per have a satisfactory cooling effect, the refrigeration obtained from a f coal will produce this cooling effect for 2275 ÷ 350 = 6½ hours with t efficient working, or 6545 ÷ 350 = 18.7 hours with the most efficient . With ice at $5 per ton, Mr. Wolff computes the cost of cooling with bout $5 per hour per thousand persons, and concludes that this is too ve for any general use. With mechanical refrigeration, however, if me 10 hours' cooling per person per pound of coal as a fair practical in regular work, we have an expense of only 15 cts. per thousand per hour, coal being estimated at $3 per short ton. This is for fuel nd the various items of oil, attendance, interest, and depreciation on t, etc., must be considered in making up the actual total cost of cal refrigeration.

-ventilation—Friction of Air in Underground Pas- —In ventilating a mine or other underground passage the resistance ercome is, according to most writers on the subject, proportional to nt of the frictional surface exposed; that is, to the product *lo* of the f the gangway by its perimeter, to the density of the air in circula- he square of its average speed, v, and lastly to a coefficient k, whose al value varies according to the nature of the sides of the gangway irregularities of its course.

rmula for the loss of head, neglecting the variation in density as tant, is $p = \frac{ksv^2}{a}$, in which p = loss of pressure in pounds per square square feet of rubbing-surface exposed to the air, v the velocity of feet per minute, a the area of the passage in square feet, and k the nt of friction. W. Fairley, in *Colliery Engineer*, Oct. and Nov. es the following formulæ for all the quantities involved, using the tation as the above, with these additions: h = horse-power of ven- l = length of air-channel; o = perimeter of air-channel; q = quan- ir circulating in cubic feet per minute; u = units of work, in foot- applied to circulate the air: w = water-gauge in inches. Then,

$$1.\ a = \frac{ksv^2}{p} = \frac{ksv^2q}{u} = \frac{ksv^3}{pv} = \frac{u}{pv} = \frac{q}{v}.$$

$$2.\ h = \frac{u}{33,000} = \frac{qp}{33,000} = \frac{5.2qw}{33,000}.$$

$$3.\ k = \frac{pa}{sv^2} = \frac{u}{sv^3} = \frac{p}{sv^2 \div a} = \frac{5.2w}{sv^2 \div a}.$$

$$4.\ l = \frac{s}{o} = \frac{pa}{kv^2o}.$$

$$5.\ o = \frac{s}{l} = \frac{pa}{kv^2l}.$$

$$6.\ p = \frac{ksv^2}{a} = \frac{u}{q} = 5.2w = \left(\sqrt[3]{\frac{u}{ks}}\right)^2 \frac{ks}{a} = \frac{ksv^3}{q} = \frac{u}{av}.$$

$$11.\ v = \frac{u}{pa} = \frac{q}{a} = \sqrt[3]{\frac{u}{ks}} = \sqrt[3]{\frac{qp}{ks}} = \sqrt{\frac{pa}{ks}}.$$

$$12.\ v^2 = \frac{pa}{ks} = \left(\sqrt[3]{\frac{u}{ks}}\right)^2.$$

$$13.\ v^3 = \frac{u}{ks} = \frac{qp}{ks} = \frac{vpa}{ks}.$$

$$14.\ w = \frac{p}{5.2} = \frac{ksv^2}{5.2a}.$$

To find the quantity of air with a given horse-power and
engine:

$$q = \frac{h \times 33{,}000 \times e}{p}.$$

The value of k, the coefficient of friction, as stated, varie
the nature of the sides of the gangway. Widely divergent va
given by different authorities (see *Colliery Engineer*, Nov.
generally accepted one until recently being probably that of
.0000000217, which is the pressure per square foot in decimals
each square foot of rubbing-surface and a velocity of one fo
Mr. Fairley, in his "Theory and Practice of Ventilating Coal-
value less than half of Atkinson's, or .00000001; and recent exp
Murgue show that even this value is high under most conditi
results are given in his paper on Experimental Investigation
Head of Air-currents in Underground Workings, Trans. A.
vol. xxiii. 63. His coefficients are given in the following table,
in twelve experiments:

		Coeffic Head French
Rock gangways.	Straight, normal section	.00092
	Straight, normal section	.00094
	Straight, large section	.00104
	Straight, normal section	.00122
Brick-lined	Straight, normal section	.00030
	Straight, normal section	.00036

i in square feet of equivalent orifice.

$$\frac{Q}{w} = \frac{Q}{2.7\sqrt{w}};^* \quad Q = \frac{A \times \sqrt{w}}{0.37}; \quad w = 0.1369 \times \left(\frac{Q}{A}\right)^2.$$

olumn or the Head of Air Due to Differences ature, etc. (Fairley.)
tive column in feet;
perature of upcast;
ght of one cubic foot of the flowing air;
perature of downcast;
oth of downcast.

$$\frac{T-t}{T \times 459} \text{ or } \frac{5.2 \times w}{f}; \quad p = f \times M; \quad w = \frac{f \times M}{5.2} = \frac{p}{5.2}.$$

ieter of a round airway to pass the same amount of air as a the length and power remaining the same:
meter of round airway, A = area of square airway; O = peri-
re airway. Then $D^2 = \sqrt[5]{\frac{A^3 \times 3.1416}{.7854^2 \times O}}$.

are employed to ventilate a mine, each of which when worked oduces a certain quantity, which may be indicated by A and B tity of air that will pass when the two fans are worked together
$+ B^2$. (For mine-ventilating fans, see page 521.)

Efficiency of Fans and Heated Chimneys for n.—W. P. Trowbridge, Trans. A. S. M. E. vii. 531, gives a theo- n of the relative amounts of heat expended to remove a given pure air by a fan and by a chimney. Assuming the total effi- n to be only 1/25, which is made up of an efficiency of 1/10 for 10 for the fan itself, and 8/10 for efficiency as regards friction, es an expenditure of heat to drive it of only 1/38 of the amount required to produce the same ventilation by a chimney 100 ft. chimney 500 ft. high the fan will be 7.6 times more efficient.
of moderate ventilation of rooms or buildings where the air

etc., is calculated as follows:

S = amount of transmitting surface in square feet;
t = temperature F. inside, t_0 = temperature outside;
K = a coefficient representing, for various materials compo
the loss by transmission per square foot of surface
mal units per hour, for each degree of difference o
on the two sides of the material;
Q = total heat transmission = $SK(t - t_0)$.

This quantity of heat is also the amount that must be co
room in order to make good the loss by transmission, but it
the additional heat to be conveyed on account of the change
poses of ventilation. The coefficients K given below are those
law by the German Government in the design of the heating
public buildings, and generally used in Germany for all bu
have been converted into American units by Mr. Wolff, and
they agree well with good American practice:

VALUE OF K FOR EACH SQUARE FOOT OF BRICK W

Thickness of brick wall.	4″	8″	12″	16″	20″	24″	28″	32″
K =	0.68	0.46	0.32	0.26	0.23	0.20	0.174	0.1

1 sq. ft., wooden-beam construction, planked over or ceiled,	.... as flooring	
	.... as ceiling,	
1 sq. ft., fireproof construction, floored over,	.... as flooring	
	.... as ceiling,	
1 sq. ft., single window	....	
1 sq. ft., single skylight	....	
1 sq. ft., double window	....	
1 sq. ft., double skylight	....	
1 sq. ft., door	....	

These coefficients are to be increased respectively as follows
exposure is a northerly one, and winds are to be counted on
factors; 10% when the building is heated during the daytime
location of the building is not an exposed one; 30% when
heated during the daytime only, and the location of the buildi
50% when the building is heated during the winter months
with long intervals (say days or weeks) of non-heating.

The value of the radiating-surface is about as follows: Ord
cast-iron radiating-surfaces, in American radiators (of Bu

heated to 69°, except a double skylight in ceiling, 14 × 24 ft., outside temperature of 0°. Store-room beyond east wall at 12 ft. in wall. Corridor beyond south wall heated to 59°. 12, in wall. Cellar below, temperature 36°.
g table shows the calculation of heat transmission:

of Transmitting Surface.	Thickness of Wall in inches.	Calculation of Area of Transmitting Surface.	Square feet of Surface.	$K(t-t_0)$.	Thermal Units.
wall................	36″	63 × 22 – 448	938	9	8,442
ndows (single)......		4 × 8 × 14	448	72	32,256
all (store-room).....	36″	42 × 22 – 72	852	4	3,408
....		6 × 12	72	19	1,368
all (corridor).......	24″	45 × 22 – 72	918	2	1,836
....		6 × 12	72	5	360
all (corridor).......	36″	17 × 22 – 72	302	1	302
....		6 × 12	72	5	360
....		32 × 42 – 336	1,008	10	10,080
skylight..............		14 × 24	336	43	14,448
....		62 × 42	2,604	4	10,416
					83,276
nentary allowance, north outside wall, 10%.........					844
" " north outside windows, 10%.....					3,226
					87,346
l location and intermittent day or night use, 30% ...					26,204
l thermal units					113,550

e that the lecture-room must be heated to 69 degrees Fahr. in hen unoccupied, so as to be at this temperature when first , there will be required, ventilation not being considered, and low-pressure steam-radiators being the heating media, about = 455 sq. ft. of radiating-surface. (This gives a ratio of about contents of room for each sq. ft. of heating-surface.)
e that there are 160 persons in the lecture-room, and we pro- feet of fresh air per person per hour, we will supply 160 × cubic feet of air per hour (i.e., $\frac{400,000}{48,000}$ = over eight changes of om per hour).
air from 0° Fahr. to 69° Fahr. will require 400,000 × 0.0189 × hermal units per hour (0.0189 being the product of a weight of y the specific heat of air). Accordingly there must be provided = 1304 sq. ft. of indirect surface, to heat the air required for zero weather. If the room were to be warmed entirely indi- by the air supplied to room (including the heat to be conveyed by transmission through walls, etc.), there would have to be he fresh-air supply 521,640 + 113,550 = 635,190 heat-units. This he provision of an amount of indirect heating-surface of the pe of 635,190 ÷ 400 = 1589 sq. ft., and the fresh air entering the have to be at a temperature of about 84° Fahr., viz., 69° = $\frac{}{9}$, or 69 + 15 = 84° Fahr.

alculations do not, however, take into account that 160 per- cture-room give out 160 × 400 = 64,000 thermal units per hour; 50 electric lights give out 50 × 1600 = 80,000 thermal units per 50 gaslights, 50 × 4800 = 240,000 thermal units per hour. The 0 people and the gas-lighting would diminish considerably the at required. Practically, it appears that the heat generated nce of 160 people, 64,000 heat-units, and by 50 electric lights, its, a total of 144,000 heat-units, more than covers the amount mitted through walls, etc. Moreover, that if the 50 gaslights 00 thermal units per hour, the air supplied for ventilation must rably below 69° Fahr., or the room will be heated to an gh *temperature. If 400,000* cubic feet of fresh air per hour are supplied, and 240,000 thermal units per hour generated by th be abstracted, it means that the air must, under these condi $\frac{240,000}{400,000 \times .0189}$ = about 32° less than 84°, or at about 52° Fahr. more, the additional vitiation due to gaslighting would necessit larger supply of fresh air than when the vitiation of the atmosp people alone is considered, one gaslight vitiating the air as mu men.

Various Rules for Computing Radiating-surf following rules are compiled from various sources. They are n nature of "rule-of-thumb" rules than those given by Mr. Wo above, but they may be useful for comparison.

Divide the cubic feet of space of the room to be heated, the of wall surface, and the square feet of the glass surface by given under these headings in the following table, and add the together; the result will be the square feet of radiating-surfac (F. Schumann.)

SPACE, WALL AND GLASS SURFACE WHICH ONE SQUARE FOOT OF SURFACE WILL HEAT.

Air Change.	Steam-pressure in pounds.	Space in cubic feet.	Exposure of Rooms.				
			All Sides.		Northwest.		Sou
			Wall Surface, sq. ft.	Glass Surface, sq. ft.	Wall Surface, sq. ft.	Glass Surface, sq. ft.	Wall Surface, sq. ft.
Once per hour.	1	190	13.8	7	15.87	8.05	16.56
	3	210	15.0	7.7	17.25	8.85	18.00
	5	225	16.5	8.5	18.97	9.77	19.80
Twice per hour.	1	75	11.1	5.7	12.76	6.55	13.22
	3	82	12.1	6.2	13.91	7.13	14.52
	5	90	13.0	6.7	14.52	7.60	15.60

EMISSION OF HEAT-UNITS PER SQUARE FOOT PER HOUR FROM CAST- OR RADIATORS. TEMP. OF AIR IN ROOM, 70° F. (F. Schumann.

Mean Temperature of Heated Pipe, Radiator, etc.	By Contact.		By Radiation.	By R and C
	Air quiet.	Air moving.		Air quiet.
Hot water.......... 140°	55.51	92.52	59.63	115.14
" " ....150°	65.45	109.18	69.69	135.11
" " ....160°	75.68	126.13	80.19	155.87
" " ....170°	86.18	143.30	91.12	177.30
" " ....180°	96.93	161.55	102.15	199.43
" " ....190°	107.90	179.83	114.45	222.35
" " ....200°	119.13	198.55	127.00	216.13
" " or steam..210°	130.49	217.48	139.96	270.49
Steam. ....220°	142.20	237.00	155.27	297.47
" ....230°	153.95	256.58	169.56	323.51
" ....240°	165.90	279.83	184.58	350.48
" ....250°	178.00	296.65	200.18	378.18
" ....260°	189.90	316.50	214.36	404.26
" ....270°	202.70	337.83	233.42	436.12
" ....280°	215.30	358.85	251.21	466.51
" ....290°	228.55	380.91	267.73	496.28
" ....300°	240.85	401.41	279.12	[illegible]

TING-SURFACE REQUIRED FOR DIFFERENT KINDS OF BUILDINGS. (From ice of the Dubuque Steam Supply Co., External Air 0° F. Chas. A. h.)

	Cubic ft. of Room heated by 1 sq. ft. of Surface. Direct System.	Indirect System.		Cubic ft. of Room heated by 1 sq. ft. of Surface. Direct System.	Indirect System.
ngs	50	40	Banks, offices, drug-stores	70	60
wholesale	125	100	Large hotels	125	100
retail	100	80	Churches	200	150

Nason Mfg. Co.'s catalogue gives the following: One square foot of will heat from 40 to 100 cu. ft. of space to 75° in − 10° latitudes. nge is intended to meet conditions of exposed or corner rooms of gs, and those less so, as intermediate ones of a block. As a general sq. ft. of surface will heat 70 cu. ft. of air in outer or front rooms and ft. in inner rooms. In large stores in cities with buildings on each o 100 is ample.

APPROXIMATE PROPORTIONS OF RADIATING-SURFACES.

quare foot radiating-surface will heat:

	In dwellings, schoolrooms, offices, etc.	In hall, stores, lofts, factories, etc.	In churches, large auditoriums, etc.
ct radiation...	60 to 80 ft.	75 to 100 ft.	150 to 200 ft.
rect radiation.	40 to 50 "	50 to 70 "	100 to 140 "

ed buildings exposed to prevailing north or west winds should have ous addition made to the heating-surface on their exposed sides. ollowing rule is given in the catalogue of the Babcock & Wilcox Co., also recommended by the Nason Mfg. Co.:

ting surface may be calculated by the rule: Add together the square glass in the windows, the number of cubic feet of air required to be d per minute, and one twentieth the surface of external wall and multiply this sum by the difference between the required temperature room and that of the external air at its lowest point, and divide the t by the difference in temperature between the steam in the pipes e required temperature of the room. The quotient is the required ng-surface in square feet.

rhead Steam-pipes. (A. R. Wolff, *Stevens Indicator*, 1887.)— he overhead system of steam-heating is employed, in which system radiating-pipes, usually 1¼ in. in diam., are placed in rows overhead, ded upon horizontal racks, the pipes running horizontally, and side , around the whole interior of the building, from 2 to 3 ft. from the and from 2 to 4 ft. from the ceiling, the amount of 1¼ in. pipe re- according to Mr. C. J. H. Woodbury, for heating mills (for which s system is deservedly much in vogue), is about 1 ft. in length for 0 cu. ft. of space. Of course a great range of difference exists, due pecial character of the operating machinery in the mill, both in re- o the amount of air circulated by the machinery, and also the aid to g the room by the friction of the journals.

rect Heating-surface.—J. H. Kinealy, in *Heating and Ven-*, May 15, 1894, gives the following formula, deduced from results of ents by C. B. Richards, W. J. Baldwin, J. H. Mills, and others, upon heaters of various kinds, supplied with varying amounts of air per r square foot of surface:

$$= \frac{35.04}{\frac{T_2 - T_1}{T_0 - T_1} - 0.369}; \quad T_2 = (T_0 - T_1)\left(0.369 + \frac{35.04}{N}\right) + T_1.$$

ubic feet of air, reduced to 70° F., supplied to the heater per square foot of heating-surface per hour;

emperature of the steam or water in the heater;

emperature of the air when it enters the heater;

emperature of the air when it leaves the heater.

formula is based upon an average of experiments made *direct heaters, the results obtained by the use of the* *me cases be slightly too small and in others slightly* although the error will in no case be great. No single formula ought to be expected to apply equally well to all dispositions of heating-surface in indirect heaters, as the efficiency of such heater can be varied between wide limits by the construction and arrangement of the surface.

In indirect heating, the efficiency of the radiating-surface will increase and the temperature of the air will diminish, when the quantity of the air caused to pass through the coil increases. Thus 1 sq. ft. radiating-surface with steam at 212°, has been found to heat 100 cu. ft. of air per hour from zero to 150°, or 300 cu. ft. from zero to 100° in the same time. The best results are attained by using indirect radiation to supply the necessary ventilation, and direct radiation for the balance of the heat. (*Steam.*)

In indirect steam-heating the least flue area should be 1 to 1¼ sq. in. to every square foot of heating-surface, provided there are no long horizontal reaches in the duct, with little rise. The register should have twice the area of the duct to allow for the fretwork. For hot water heating from 25 to 30% more heating-surface and flue area should be given than for low-pressure steam. (*Engineering Record*, May 26, 1894.)

Boiler Heating-surface Required. (A. R. Wolff, *Stevens Indicator*, 1887.)—When the direct system is used to heat buildings in which the street floor is a store, and the upper floors are devoted to sales and stock rooms and to light manufacturing, and in which the fronts are of stone or iron, and the sides and the rear of building of brick—a safe rule to follow is to supply 1 sq. ft. of boiler heating-surface for each 700 cu. ft., and 1 sq. ft. of radiating-surface for each 100 cu. ft. of contents of building.

For heating mills, shops, and factories, 1 sq. ft. of boiler heating-surface should be supplied for each 475 cu. ft. of contents of building; and the same allowance should also be made for heating exposed wooden dwellings. For heating foundries and wooden shops, 1 sq. ft. of boiler heating-surface should be provided for each 400 cu. ft. of contents; and for structures in which glass enters very largely in the construction—such as conservatories, exhibition buildings, and the like—1 sq. ft. of boiler heating-surface should be provided for each 275 cu. ft. of contents of building.

When the indirect system is employed, the radiator-surface and the boiler capacity to be provided will each have to be, on an average, about 25% more than where direct radiation is used. This percentage also marks approximately the increased fuel consumption in the indirect system.

Steam (Babcock & Wilcox Co.) has the following: 1 sq. ft. of boiler-surface will supply from 7 to 10 sq. ft. of radiating-surface, depending upon the size of boiler and the efficiency of its surface, as well as that of the radiating-surface. Small boilers for house use should be much larger proportionately than large plants. Each horse-power of boiler will supply from 240 to [illegible] ft. of 1-in. steam-pipe, or 80 to 120 sq. ft. of radiating surface. Cubic feet of space has little to do with amount of steam or surface required, but is a convenient factor for rough calculations. Under ordinary conditions one horse-power will heat, approximately, in—

Brick dwellings, in blocks, as in cities	15,000	to	20,000 cu. ft.
" stores " "	10,000	"	15,000 "
" dwellings, exposed all round	10,000	"	15,000 "
" mills, shops, factories, etc.	7,000	"	10,000 "
Wooden dwellings, exposed	7,000	"	10,000 "
Foundries and wooden shops	6,000	"	10,000 "
Exhibition buildings, largely glass, etc.	4,000	"	15,000 "

Proportion of Grate-surface to Radiator-surface.

(J. R. Willett, *Heating and Ventilation*, Feb. 1894.)

Radiator-surf., sq. ft.	100	200	400	600	800	1000	1200	1400	1600	1800
Grate-surface, sq. in.	120	208	362	501	630	754	872	986	1100	1210

Steam-consumption in Car-heating.

C., M. & St. Paul Railway Tests. (*Engineering*, June 27, 1890, p. 7[illegible]

Outside Temperature.	Inside Temperature.	Water of Condensation per Car per Hour.
[illegible]	70	70 lbs.
[illegible]	70	85
[illegible]	70	100

ernal Diameters of Steam Supply-mains, with Total Resistance equal to 2 inches of Water-column.*

eam, Pressure 10 lbs. per square inch above atm., Temperature 239° F.

Formula, $d = 0.5374\sqrt[5]{\frac{Q^2 l}{h}}$; where d = internal diameter in inches;

= 9.2 cubic feet of steam per minute per 100 sq. ft. of radiating-surface;
= length of mains in feet; h = 159.3 feet head of steam to produce flow.

surface.	Internal Diameters in inches for Lengths of Mains from 1 ft. to 600 ft.										
	1 ft.	10 ft.	20 ft.	40 ft.	60 ft.	80 ft.	100 ft.	200 ft.	300 ft.	400 ft.	600 ft.
.ft.	inch.	inch.	inch.	inch.	inch.	inch.	inch.	inch.	inch.	inch.	inch.
1	0.075	0.119	0.136	0.157	0.170	0.180	0.189	0.216	0.234	0.248	0.270
10	0.19	0.30	0.34	0.39	0.43	0.45	0.47	0.54	0.59	0.62	0.68
20	0.25	0.39	0.45	0.52	0.56	0 60	0.62	0.72	0.78	0.82	0.89
40	0.33	0.52	0.60	0.69	0.74	0.79	0.82	0.95	1.03	1.09	1.18
60	0.39	0.61	0.71	0.81	0.87	0.93	0.97	1.11	1.21	1.28	1.39
80	0.43	0.68	0.79	0.90	0.98	1.04	1.09	1.25	1.35	1.43	1.55
100	0.47	0.75	0.86	0.99	1.07	1.14	1.19	1.36	1.48	1.57	1.70
200	0 62	0.99	1.14	1.30	1.41	1.50	1.57	1.80	1.95	2.07	2.24
00	0.73	1.16	1.34	1.53	1.66	1.76	1.84	2 12	2.30	2.43	2.64
00	0.82	1.30	1.50	1.72	1.86	1.98	2.07	2.37	2.57	2.73	2.96
00	0.90	1.43	1.64	1.88	2.04	2.16	2.26	2.60	2.81	2.98	3.23
00	0.97	1.53	1.76	2.03	2.20	2.33	2.43	2.79	3.03	3.21	3.48
00	1.09	1.72	1.98	2.27	2 46	2.61	2.73	3.13	3.40	3.60	3.90
00	1.19	1.88	2.16	2.48	2.69	2.85	2.98	3.43	3.71	3.94	4.27
00	1.28	2.04	2.33	2.67	2.90	3.07	3.21	3.68	4.00	4.23	4.59
00	1.36	2.15	2.47	2.84	3.08	3.26	3.41	3.92	4.25	4.50	4.88
00	1.43	2.27	2.61	3.00	3.25	3.44	3.60	4.13	4.49	4.75	5.15
00	1.50	2.38	2.74	3.14	3.41	3.61	3.78	4.34	4.70	4.98	5.40
00	1.57	2.48	2.85	3.28	3.55	3.76	3.93	4.52	4.90	5.19	5.68
00	1.84	2.92	3.36	3.85	4.18	4.43	4.63	5.32	5.77	6.11	6.63
00	2.07	3.28	3.76	4.32	4.69	4.96	5.19	5.96	6.47	6.85	7.44

* From Robert Briggs's paper on American Practice of Warming Buildings Steam (Proc. Inst. C. E., 1882, vol. lxxi).

For other resistances and pressures above atmosphere multiply by the pective factors below:

ter col.	6 in.	12 in.	24 in.	Press. ab. atm.	0 lbs.	3 lbs.	30 lbs.	60 lbs.
ltiply by	0.8027	0.6988	0.6084	Multiply by	1.023	1.015	0.973	0.948

Registers and Cold-air Ducts for Indirect Steam Heating.

he *Locomotive* gives the following table of openings for registers and l-air ducts, which has been found to give satisfactory results. The cold-boxes should have 1¼ sq. in. area for each square foot of radiator suface, never less than ¾ the sectional area of the hot-air ducts. The hot air ts should have 2 sq. in. of sectional area to each square foot of radiator ace on the first floor, and from 1½ to 2 inches on the second floor.

ating Surface in Stacks.	Cold-air Supply, First Floor.		Size Register.	Cold-air Supply, 2d Floor.
		inches	inches	inches
square feet	45 square inches =	5 by 9	9 by 12	4 by 10
" "	60 " " =	6 by 10	10 by 14	4 by 14
" "	75 " " =	8 by 10	10 by 14	5 by 15
" "	90 " " =	9 by 10	12 by 15	6 by 15
" "	108 " " =	9 by 12	12 by 19	6 by 18
" "	120 " " =	10 by 12	12 by 22	8 by 15
" "	135 " " =	11 by 12	14 by 24	9 by 15
" "	150 " " =	12 by 12	16 by 20	12 by 1?

sizes in the table approximate to the rules given, and it will b*ey will allow an easy flow of air* and a full distribution throu*o be heated.*

Physical Properties of Steam and Condensed Wat[er] under Conditions of Ordinary Practice in Warming [by] Steam. (Briggs.)

A	Steam-pressure per square inch, above atm.	lbs.	0	3	10	30
	Steam-pressure per square inch, total	lbs.	14.7	17.7	24.7	44.7
B	Temperature of steam	Fahr.	212°	222°	239°	274°
C	Temperature of air	Fahr.	60°	60	60°	60°
D	Difference = B − C	Fahr.	152°	162°	179°	214°
E	Heat given out per minute per 100 sq. ft. of radiating-surface = D × 3	units	456	486	537	642
F	Latent heat of steam	Fahr.	965°	958°	946°	921°
G	Volume of 1 lb. weight of steam	cu. ft.	26.4	22.1	16.2	[illegible]
H	Weight of 1 cubic foot of steam	lb.	0.0380	0.0452	0.0618	[illegible]
J	Volume Q of steam per minute to give out E units = E × G ÷ F.	cu. ft.	12.48	11.21	9.20	6.44
K	Weight of 1 cubic foot of condensed water at temperature B.	lbs.	59.64	59.51	59.05	[illegible]
L	Volume of condensed water to return to boiler per minute = J × H ÷ K.	cu. ft.	0.0079	0.0085	0.0096	0.0120
M	Head of steam equivalent to 12 inches water-column = K ÷ H.	feet	1569	1317	955.5	586.7
	STEAM-SUPPLY MAINS.					
N	Head h of steam, equivalent to assumed 2 inches water-column for producing steam flow Q, = M ÷ 6,	feet	261.5	219.5	159.3	[illegible]
P	Internal diameter d of tube* for flow Q when l = 1 foot,	inch	0.484	0.481	0.474	0.461
R	Do. do. when l = 100 feet,	inch	1.217	1.207	1.190	[illegible]
S	Ratios of values of d.	ratio	1.023	1.015	**1.000**	[illegible]
	WATER-RETURN MAINS.					
T	Head h assumed at ½-inch water-column for producing full-bore water-flow Q.	foot	0.0417	0.0417	0.0417	[illegible]
U	Internal diameter d of tube* for flow Q when l = 1 foot,	inch	0.147	0.151	[illegible]	[illegible]
V	Do. do. when l = 100 feet,	inch	0.368	0.379	[illegible]	[illegible]
W	Ratios of values of d	ratio	0.926	0.952	**1.000**	[illegible]

* P, R, U, V are each determined from the formula $d = 0.5374 \sqrt[5]{\frac{Q^2 l}{h}}$

Size of Steam Pipes for Steam Heating. (See also Flow [of] Steam in Pipes.)—*Sizes of vertical main pipes. Direct radiation.* (Willett, *Heating and Ventilation*, Feb., 1894.)

Diameter of pipe, inches.	1	1¼	1½	2	2½	3	3½	4	5
Sq. ft. of radiator surface	40	70	110	220	360	560	810	1110	[illegible]

A horizontal branch pipe for a given extent of radiator surface shou[ld be] one size larger than a vertical pipe for the same surface.

The Nason Mfg. Co. gives the following:

Diameter of pipe, in	1¼	1½	2	2½	3	3½
Radiator surface sq. ft. (maximum)	125	200	500	1000	1500	[illegible]

When mains and surfaces are very much above the boiler the [illegible] *be as la[illegible]* given above; under very favorable circumstances ns a 4-inch pipe may supply from 2000 to 2500 sq. ft. of surface, a 6- e for 5000 sq. ft., and a 10-inch pipe for 15,000 to 20,000 sq. ft., if the of run from boiler is not too great. Less than 1½-inch pipe should sed horizontally in a main unless for a single radiator connection . by the Babcock & Wilcox Co., says: Where the condensed water ed to the boiler, or where low pressure of steam is used, the diame- nains leading from the boiler to the radiating-surface should be inches to one tenth the square root of the radiating-surface, mains , in square feet. Thus a 1-inch pipe will supply 100 square feet of itself included. Return-pipes should be at least ¾ inch in diame- never less than one half the diameter of the main—longer returns g larger pipe. A thorough drainage of steam-pipes will effectually all cracking and pounding noises therein.

Wolff's Practice.—Mr. Wolff gives the following figures showing his practice (1897) in proportioning mains and returns. They are based timated loss of pressure of 2% for a length of 100 ft. of pipe, not in- allowance for bends and valves (see p. 678). For longer runs divide nal units given in the table by $0.1\sqrt{\text{length in ft.}}$ Besides giving the units the table also indicates the amount of direct radiating surface e steam-pipes can supply, on the basis of an emission of 250 thermal hour for each square foot of direct radiating surface.

Size of Pipes for Steam Heating.

2 lbs. Pressure		5 lbs. Pressure		Diam. of Supply.	Diam. of Return.	2 lbs. Pressure		5 lbs. Pressure	
Thermal Units per Hr., Thous'ds	Heating-surface, Sq. Ft.	Thermal Units per Hr., Thous'ds	Heating-surface, Sq. Ft.	In.	In.	Thermal Units, per Hr., Thous'ds	Heating-surface, Sq. Ft.	Thermal Units, per Hr., Thous'ds	Heating-surface, Sq. Ft.
9	36	15	60	5	3½	930	3720	1550	6200
18	72	30	120	6	3½	1500	6000	2500	10000
30	120	50	200	7	4	2250	9000	3750	15000
70	280	120	480	8	4	3200	12800	5400	21600
132	528	220	880	9	4½	4450	17800	7500	30000
225	900	375	1500	10	5	5800	23200	9750	39000
330	1320	550	2200	12	6	9250	37000	15500	62000
480	1920	800	3200	14	7	13500	54000	23000	92000
690	2760	1150	4600	16	8	19000	76000	32500	130000

ting a Greenhouse by Steam.—Wm. J. Baldwin answers a in the *American Machinist* as below: With five pounds steam- , how many square feet or inches of heating-surface is necessary to square feet of glass on the roof, ends, and sides of a greenhouse to maintain a night heat of 55° to 65°, while the thermometer out- ges at from 15° to 20° below zero; also, what boiler-surface is neces- Which is the best for the purpose to use—2″ pipe or 1¼″ pipe?

Reliable authorities agree that 1.25 to 1.50 cubic feet of air in an space will be cooled per minute per sq. ft. of glass as many degrees nternal temperature of the house exceeds that of the air outside. + 65° and − 20° there will be a difference of 85°, or, say, one cubic air cooled 127.5° F. for each sq ft. of glass for the most extreme n mentioned. Multiply this by the number of square feet of d by 60, and we have the number of cubic feet of air cooled 1° per thin the building or house. Divide the number thus found by 48, and the units of heat required, approximately. Divide again by 953, ill give the number of pounds of steam that must be condensed from re and temperature of five pounds above atmosphere to water at e temperature in an hour to maintain the heat. Each square foot ce of pipe will condense from ¼ to nearly ½ lb. of steam per hour, g as the coils are exposed or well or poorly arranged, for which ge of ⅓ lb. may be taken. According to this, it will require 3 sq. ft. surface per lb. of steam to be condensed. Proportion the heating- of the boiler to have about one fifth the actual radiating-surface, if to keep steam over night, and proportion the grate to burn not an six pounds of coal per sq. ft. of grate per hour. With very slow tion, such as takes place in base-burning boilers, the grate mig oned for four to five pounds of coal per hour. It is cheaper t ¼″ pipe than of 2″, and there is nothing to be gained by s the coils are very long. The pipes in a greenhouse sh

under or in front of the benches, with every chance for a good ci
of air. "Header" coils are better than "return-bend" coils for this
Mr. Baldwin's rule may be given the following form: Let $H =$ h
transferred per hour, $T =$ temperature inside the greenhouse, $t =$
ture outside, $S =$ sq. ft. of glass surface; then $H = 1.5S(T - t)$
$= 1.875S(T - t)$. Mr. Wolff's coefficient K for single skylights w
$H = 1.118S(T - t)$.

Heating a Greenhouse by Hot Water.—W. M. Mack
Richardson & Boynton Co., in a lecture before the Master Plumbe
ciation, N. Y., 1889, says: I find that while greenhouses were
heated by 4-inch and 3-inch cast-iron pipe, on account of the large
water which they contained, and the supposition that they gave bet
faction and a more even temperature, florists of long experie
have tried 4-inch and 3-inch cast-iron pipe, and also 2-inch wro
pipe for a number of years in heating their greenhouses by h
and who have also tried steam-heat, tell me that they get better sat
greater economy, and are able to maintain a more even temperatu
inch wrought-iron pipe and hot water than by any other system t
used. They attribute this result principally to the fact that this
contains less water and on this account the heat can be raised and
quicker than by any other arrangement of pipes, and a more unifo
perature maintained than by steam or any other system.

HOT-WATER HEATING.

(Nason Mfg. Co.)

There are two distinct forms or modifications of hot-water appa
pending upon the temperature of the water.
In the first or open-tank system the water is never above 212°
ture, and rarely above 200°. This method always gives satisfacti
the surface is sufficiently liberal, but in making it so its cost is con
greater than that for a steam-heating apparatus.
In the second method, sometimes called (erroneously) high-pres
water heating, or the closed-system apparatus, the tank is closed
provided with a safety-valve set at 10 lbs. it is practically as safe as
tank system.

Law of Velocity of Flow.—The motive power of the ci
in a hot-water apparatus is the difference between the specific gr
the ascending and the descending pipes. This effective pressur
small, and is equal to about one grain for each foot in height for
gree difference between the pipes; thus, with a height of 12″ in "
and a difference between the temperatures of the up and down pi
the difference in their specific gravities is equal to 8.16 grains on ea
inch of the section of return-pipe, and the velocity of the circulati
portioned to these differences in temperature and height.

To Calculate Velocity of Flow.—Thus, with a height o
ing pipe equal to 10′ and a difference in temperatures of the flow a
pipes of 8°, the difference in their specific gravities will equal 81.6 g
$\div 7000 = .01166$ lbs., or $\times 2.31$ (feet of water in one pound) $= .0269$ ft
the law of falling bodies the velocity will be equal to $8\sqrt{.0269} = 1.3$
second, or $\times 60 = 78.7$ ft. per minute. In this calculation the effec
tion is entirely omitted. Considerable deduction must be made
account. Even in apparatus where length of pipe is not great,
pipes of larger areas and with few bends or angles, a large dedu
friction must be made from the theoretical velocity, while in l
complex apparatus with small head, the velocity is so much re
friction that sometimes as much as from 50% to 90% must be deduct
tain the true rate of circulation.
Main flow-pipes from the heater, from which branches may be ta
to be preferred to the practice of taking off nearly as many pipes
heater as there are radiators to supply.
It is not necessary that the main flow and return pipes should
capacity that of all their branches. The hottest water will seek the
level, while gravity will cause an even distribution of the heated wa
surface is properly proportioned.
It is ... to reduce the size of the vertical mains as they
... size for each floor.
4. ... h hot water, the pipes must be unconfined

n of the pipes consequent on having their temperatures in-

ion tank is required to keep the apparatus filled with water, expands 1/24 of its bulk on being heated from 40° to 212°, and ust have capacity to hold certainly this increased bulk. It is d that the supply cistern be placed on level with or above the of the apparatus, in order to receive the air which collects in nd radiators, and capable of holding at least 1/20 of the water apparatus.

ximate Proportions of Radiating-surfaces to Cubic Capacities of Space to be Heated.

Foot of Ra- rface will with—	In Dwellings, School-rooms, Offices, etc.	In Halls, Stores, Lofts, Factories, etc.	In Churches, Large Auditoriums, etc.
rature di- ater radi- ...	50 to 70 cu. ft.	65 to 90 cu. ft.	130 to 180 cu. ft.
rature di- ater radi- ...	30 to 50 " "	35 to 65 " "	70 to 130 " "
rature in- -water ra- ...	30 to 60 " "	35 to 75 " "	70 to 150 " "
rature in- -water ra- ...	20 to 40 " "	25 to 50 " "	50 to 100 " "

er of Main and Branch Pipes and square feet of coil will supply, in a low-pressure hot-water apparatus (212°) for direct radiation, when coils are at different altitudes for direct in the lower story for indirect radiation:

Direct Radiation. Height of Coil above Bottom of Boiler, in feet.

10	20	30	40	50	60	70	80	90	100
sq. ft.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	sq. ft.	sq. ft.
50	52	53	55	57	59	61	63	65	68
89	92	95	98	101	103	108	112	116	121
140	144	149	153	158	161	169	175	182	189
202	209	214	222	228	235	243	252	261	271
359	370	380	393	405	413	433	449	465	483
561	577	595	613	633	643	678	701	727	755
807	835	856	888	912	941	974	1009	1046	1086
1099	1132	1166	1202	1241	1283	1327	1374	1425	1480
1436	1478	1520	1571	1621	1654	1733	1795	1861	1933
1817	1871	1927	1988	2052	2120	2193	2272	2356	2445
2244	2309	2376	2454	2531	2574	2713	2805	2907	3019
3228	3341	3424	3552	3648	3763	3897	4036	4184	4344
4396	4528	4664	4808	4964	5132	5308	5496	5700	5920
5744	5912	6080	6284	6484	6616	6932	7180	7444	7735
7268	7484	7708	7952	8208	8482	8774	9088	9424	9780
8976	9236	9516	9816	10124	10296	10852	11220	11628	12076
10860	11180	11519	11879	12262	12666	13108	13576	14078	14620
12912	13364	13696	14208	14592	15052	15588	16144	16736	17376
15169	15615	16090	16591	17126	17697	18307	18961	19633	20420
17584	18109	18656	19232	19856	20528	21232	21984	[illegible]	[illegible]
20195	20789	21419	22089	22801	23561	24373	25244	[illegible]	[illegible]
22978	*23643*	*24320*	*25136*	*25936*	26464	27728	28720	[illegible]	[illegible]

The best forms of hot-water-heating boilers are proportioned ab[illegible] follows:

1 sq. ft. of grate-surface to about 40 sq. ft. of boiler-surface.
1 " " boiler- " " 5 " " radiating-surface.
1 " " grate- " " 200 " " " "

Rules for Hot-water Heating.—J. L. Saunders (Heatin[illegible] Ventilation, Dec. 15, 1894) gives the following: Allow 1 sq. ft. of radi[illegible] surface for every 3 ft. of glass surface, and 1 sq. ft. for every 20 sq[illegible] wall surface, also 1 sq. ft. for the following numbers of cubic feet of [illegible] in the several cases mentioned.

In dwelling-houses:	Libraries and dining-rooms, first floor	35 to 40
	Reception halls, first floor	40 to 50
	Stair halls, " "	40 to 50
	Chambers above, " "	50 to 65
	Libraries, sewing-rooms, nurseries, etc., above first floor	45 to 55
	Bath-rooms	30 to 40
Public-school rooms		60 to 80
Offices		50 to 65
Factories and stores		65 to 90
Assembly halls and churches		90 to 150

To find the necessary amount of indirect radiation required to heat a [illegible] Find the required amount of direct radiation according to the for[illegible] method and add 50%. This if wrought-iron pipe coil surface is used; [illegible] iron pin indirect-stack surface is used it is advisable to add from 70% t[illegible]

Sizes of hot-air flues, cold-air ducts, and registers for indirect [illegible] Hot-air flues, first floor: Make the net internal area of the flue eq[illegible] ¾ sq. in. to every square foot of radiating surface in the indirect stack[illegible] air flues, second floor: Make the net internal area of the flue equal to ⅝[illegible] to every square foot of radiating surface in the indirect stack.

Cold-air ducts, first floor: Make the net internal area of the duc[illegible] to ⅝ sq. in. to every square foot of radiating surface in the indirect[illegible] Cold air ducts, second floor: Make the net internal area of the duct[illegible] to ½ sq. in. to every square foot of radiating surface in the indirect s[illegible]

Hot-air registers should have their net area equal in full to the area [illegible] hot-air flues. Multiply the length by the width of the register in inch[illegible] of the product is the net area of register.

Arrangement of Mains for Hot-water Heating. [illegible] Mackay, Lecture before Master Plumbers' Assoc., N. Y., 1889.)—The[illegible] two different systems of mains in general use, either of which, if pr[illegible] placed, will give good satisfaction. One is the taking of a single lar[illegible] main from the heater to supply all the radiators on the several floors, [illegible] corresponding return main of the same size. The other is the takin[illegible] number of 2-inch wrought-iron mains from the heater, with the same [illegible] ber of return mains of the same size, branching off to the several ra[illegible] or coils with 1¼-inch or 1-inch pipe, according to the size of the radia[illegible] coil. A 2-inch main will supply three 1¼-inch or four 1-inch branche[illegible] these branches should be taken from the top of the horizontal main [illegible] nipple and elbow, except in special cases where it is found necessary to [illegible] the flow of water to the near radiator, for the purpose of assisting the [illegible] lation in the far radiator; in this case the branch is taken from the [illegible] the horizontal main. The flow and return mains are usually run side b[illegible] suspended from the basement ceiling, and should have a gradual ascen[illegible] the heater to the radiators of at least 1 inch in 10 feet. It is customar[illegible] an advantage where 2-inch mains are used, to reduce the size of the m[illegible] every point where a branch is taken off.

The single or large main system is best adapted for large building[illegible] there is a limit as to size of main which it is not wise to go beyond—[illegible] ally 6-inch, except in special cases.

The proper area of cold-air pipe necessary for 100 square feet of h[illegible] radiation in hot-water heating is 75 square inches, while the hot a[illegible] should have at least 100 square inches of area. There should be a dam[illegible] the cold-air pipe for the purpose of controlling the amount of air admi[illegible] the radiator, depending on the severity of the weather.

BLOWER SYSTEM OF HEATING AND VENTILATING.

provides for the use of a fan or blower which takes its supply rom the outside of the building to be heated, forces it over cated either centrally or divided up into a number of indepen- nd then into the several ducts or flues leading to the various novement of the warmed air is positive, and the delivery of various points of supply is certain and entirely independent conditions. For engines, fans, and steam-coils used with the , see page 519.

nts with Radiators of 60 sq. ft. of Surface. Dec., 1893.)—After having determined the volume and tem- he warm air passing through the flues and radiators from , a fan was applied to each flue, forcing in air, and new sets of were made. The results showed that more than two and one- much air was warmed with the fans in use, and the falling off ture of this greatly increased air-volume was only about 12.6%. ion of steam in the radiators with the forced-air circulation 66⅔% greater than with natural-air draught. One of the test figures obtained is as follows:

	Natural Draught in Flue.	Forced-air Circulation.
of air per minute	457.5	1227
on of steam per minute in ounces	11.7	19.6
sure in radiator, pounds	9	9
re of air after leaving radiator	142°	124°
" " before passing through radiator.	61°	61°
radiating surface in square feet	60	60
in both cases	12 × 18 inches.	

obably an error in the determination of the volume of air in appears from the following calculation. (W. K.) Assume am in condensing from 9 lbs. pressure and cooling to the tem- ich the water may have been discharged from the radiator eat-units, or 62.5 h. u. per ounce; that the air weighed .076 lb. and that its specific heat is .238. We have

	Natural Draught.	Forced Draught.
by steam, ounces × 62.5 =	731	1225 H.U.
by air, cu. ft. × .076 × diff. of tem. × .238 =	673	1399 "

of forced draught the air received 14% more heat than the , which is impossible. Taking the heat given up by the steam neasure of the work done by the radiator, the temperature t 237°, and the average temperature of the air in the case of it at 102° and in the other case at 93°, we have for the tem- ence in the two cases 135° and 144° respectively; dividing heat-units we find that each square foot of radiating surface heat-units per hour per degree of difference of temperature, natural draught, and 8.5 heat-units in the case of forced × 144° = 1224 heat-units per square foot of surface).

en's Homœopathic Hospital in Philadelphia, 2000 feet of heats 250,000 cubic feet of space, ventilating as well; this re foot of pipe surface for about 350 cubic feet of space, or are feet for 1000 cubic feet. The fan is located in a sepa- bout 100 feet from the hospital, and the air, after being heated s conveyed through an underground brick duct with a loss of degrees in cold weather. (H. I. Snell, Trans. A. S. M. E., ix. 106.

Building to 70° F. Inside when the Outside re is Zero.—It is customary in some contracts for heating at the apparatus will heat the interior of the building to 70° . As it may not be practicable to obtain zero weather for a test, it may be difficult to prove the performance of the E. Macgovern, in *Engineering Record*, Feb. 3, 1894, gives a ding to show that a test may be made in weather of a hi an zero, if the *heat of the interior* is raised above 70° *perature of the rooms* the lower is the efficiency of th *nce the efficiency* depends upon the difference betw temperature inside of the radiator and the temperature of the room concludes that a heating apparatus sufficient to heat a given building in zero weather with a given pressure of steam will be found to heat the same building, steam-pressure constant, to 110° at 60°, 95° at 50°, 82° and 74° at 32°, outside temperature. The accuracy of these figures, however, has not been tested by experiment.

The following solution of the question is proposed by the author. It gives results quite different from those of Mr. Macgovern, but, like them, lacks experimental confirmation.

Let S = sq. ft. of surface of the steam or hot-water radiator;
W = sq. ft. of surface of exposed walls, windows, etc.;
T_s = temp. of the steam or hot water, T_1 = temp. of inside of building or room, T_0 = temp. of outside of building or room;
a = heat-units transmitted per sq. ft. of surface of radiator per hour per degree of difference of temperature;
b = average heat-units transmitted per sq. ft. of walls per hour per degree of difference of temperature, including allowance for ventilation.

It is assumed that within the range of temperatures considered Newton's law of cooling holds good, viz., that it is proportional to the difference of temperature between the two sides of the radiating-surface.

Then $aS(T_s - T_1) = bW(T_1 - T_0)$. Let $\frac{bW}{aS} = C$; then

$$T_s - T_1 = C(T_1 - T_0); \quad T_1 = \frac{T_s + CT_0}{1 + C}; \quad C = \frac{T_s - T_1}{T_1 - T_0}.$$

If $T_1 = 70$, and $T_0 = 0$, $C = \frac{T_s - 70}{70}$.

Let T_s =	140°,	213.5°,	308°;
Then C =	1,	2.05,	3.4.

From these we derive the following:

Temperature of Steam or Hot Water, T_s.	Outside Temperatures, T_0.					
	− 20°	− 10°	0°	10°	20°	30°
	Inside Temperatures, T_1.					
140°	60	65	70	75	80	85
213.5	56.6	63.3	70	76.7	83.4	90.1
308	54.5	62.3	70	77.7	85.5	93.2

Heating by Electricity.—If the electric currents are generated by a dynamo driven by a steam-engine, electric heating will prove very expensive, since the steam-engine wastes in the exhaust-steam and by radiation about 90% of the heat-units supplied to it. In direct steam-heating, with a good boiler and properly covered supply-pipes, we can utilize about 60% of the total heat value of the fuel. One pound of coal, with a heating value of 13,000 heat-units, would supply to the radiators about 13,000 × .60 = 7800 heat-units. In electric heating, suppose we have a first-class condensing engine developing 1 H.P. for every 2 lbs. of coal burned per hour. This would be equivalent to 1,980,000 ft.-lbs. ÷ 778 = 2545 heat-units, or 1272 heat-units for 1 lb. of coal. The friction of the engine and of the dynamo, the loss by electric leakage, and by heat radiation from the conducting wires, might reduce the heat-units delivered as electric current to the electric radiator, and these converted into heat to 50% of this, or only 636 heat-units, or less than one twelfth of that delivered to the steam-radiator in direct steam-heating. Electric heating, therefore, will prove uneconomical unless the electric current is derived from water or wind power, which would otherwise be wasted. (See Electrical Engineering.)

WATER.

Expansion of Water.—The following table gives the relative vol-
es of water at different temperatures, compared with its volume at 4° C.
cording to Kopp, as corrected by Porter.

ent.	Fahr.	Volume.	Cent.	Fahr.	Volume.	Cent.	Fahr.	Volume.
4°	39.1°	1.00000	35°	95°	1.00586	70°	158°	1.02241
5	41	1.00001	40	104	1.00767	75	167	1.02548
10	50	1.00025	45	113	1.00967	80	176	1.02872
15	59	1.00083	50	122	1.01186	85	185	1.03213
20	68	1.00171	55	131	1.01423	90	194	1.03570
25	77	1.00286	60	140	1.01678	95	203	1.03943
30	86	1.00425	65	149	1.01951	100	212	1.04332

eight of 1 cu. ft. at 39.1° F. = 62.4245 lb. ÷ 1.04332 = 59.833, weight of 1 cu.
at 212° F.

Weight of Water at Different Temperatures.—The weight
ater at maximum density, 39.1°, is generally taken at the figure given
Rankine, 62.425 lbs. per cubic foot. Some authorities give as low as
9. The figure 62.5 commonly given is approximate. The highest
horitative figure is 62.425. At 62° F. the figures range from 62.291 to 62.360.
figure 62.355 is generally accepted as the most accurate.
t 32° F. figures given by different writers range from 62.379 to 62.418.
k gives the latter figure, and Hamilton Smith, Jr., (from Rosetti,) gives
16.

Weight of Water at Temperatures above 212° F.—Porter
hards' "Steam-engine Indicator," p. 52) says that nothing is known
ut the expansion of water above 212°. Applying formulæ derived from
eriments made at temperatures below 212°, however, the weight and
ume above 212° may be calculated, but in the absence of experimental
we are not certain that the formulæ hold good at higher temperatures.
urston, in his "Engine and Boiler Trials," gives a table from which we
the following (neglecting the third decimal place given by him):

deg. F.	Weight, lbs. per cubic foot.	Temperature, deg. F.	Weight, lbs. per cubic foot.	Temperature, deg. F.	Weight, lbs. per cubic foot.	Temperature, deg. F.	Weight, lbs. per cubic foot.	Temperature, deg. F.	Weight, lbs. per cubic foot.
2	59.71	280	57.90	350	55.52	420	52.86	490	50.03
0	59.64	290	57.59	360	55.16	430	52.47	500	49.61
0	59.37	300	57.26	370	54.79	440	52.07	510	49.20
0	59.10	310	56.93	380	54.41	450	51.66	520	48.78
0	58.81	320	56.58	390	54.03	460	51.26	530	48.36
4	58.52	330	56.24	400	53.64	470	50.85	540	47.94
0	58.21	340	55.88	410	53.26	480	50.44	550	47.52

ox on Heat gives the following:

perature F.......	212°	250°	300°	350°	400°	450°	500°
per cubic foot....	59.82	58.85	57.42	55.94	54.34	52.70	51.02

t 212° figures given by different writers (see Trans. A. S. M. E., x
ge from 59.56 to 59.845, averaging about 59.77.

Weight of Water per Cubic Foot, from 32° to 212° F.
units per pound, reckoned above 32° F.: The following table,
terpolating the table given by Clark as calculated from Rankine
with corrections for apparent errors, was published by the au
Trans. A. S. M. E., vi. 90. (For heat units above 212° see Steam

Temp., deg. F.	Weight, lbs. per cubic foot.	Heat-units.	Temperature, deg. F.	Weight, lbs. per cubic foot.	Heat-units.	Temperature, deg. F.	Weight, lbs. per cubic foot.	Heat-units.	Temperature, deg. F.
32	62.42	0.	78	62.25	46.03	123	61.68	91.16	168
33	62.42	1.	79	62.24	47.03	124	61.67	92.17	169
34	62.42	2.	80	62.23	48.04	125	61.65	93.17	170
35	62.42	3.	81	62.22	49.04	126	61.63	94.17	171
36	62.42	4.	82	62.21	50.04	127	61.61	95.18	172
37	62.42	5.	83	62.20	51.04	128	61.60	96.18	173
38	62.42	6.	84	62.19	52.04	129	61.58	97.19	174
39	62.42	7.	85	62.18	53.05	130	61.56	98.19	175
40	62.42	8.	86	62.17	54.05	131	61.54	99.20	176
41	62.42	9.	87	62.16	55.05	132	61.52	100.20	177
42	62.42	10.	88	62.15	56.05	133	61.51	101.21	178
43	62.42	11.	89	62.14	57.05	134	61.49	102.21	179
44	62.42	12.	90	62.13	58.06	135	61.47	103.22	180
45	62.42	13.	91	62.12	59.06	136	61.45	104.22	181
46	62.42	14.	92	62.11	60.06	137	61.43	105.23	182
47	62.42	15.	93	62.10	61.06	138	61.41	106.23	183
48	62.41	16.	94	62.09	62.06	139	61.39	107.24	184
49	62.41	17.	95	62.08	63.07	140	61.37	108.25	185
50	62.41	18.	96	62.07	64.07	141	61.36	109.25	186
51	62.41	19.	97	62.06	65.07	142	61.34	110.26	187
52	62.40	20.	98	62.05	66.07	143	61.32	111.26	188
53	62.40	21.01	99	62.03	67.08	144	61.30	112.27	189
54	62.40	22.01	100	62.02	68.08	145	61.28	113.28	190
55	62.39	23.01	101	62.01	69.08	146	61.26	114.28	191
56	62.39	24.01	102	62.00	70.09	147	61.24	115.29	192
57	62.39	25.01	103	61.99	71.09	148	61.22	116.29	193
58	62.38	26.01	104	61.97	72.09	149	61.20	117.30	194
59	62.38	27.01	105	61.96	73.10	150	61.18	118.31	195
60	62.37	28.01	106	61.95	74.10	151	61.16	119.31	196
61	62.37	29.01	107	61.93	75.10	152	61.14	120.32	197
62	62.36	30.01	108	61.92	76.10	153	61.12	121.33	198
63	62.36	31.01	109	61.91	77.11	154	61.10	122.33	199
64	62.35	32.01	110	61.89	78.11	155	61.08	123.34	200
65	62.34	33.01	111	61.88	79.11	156	61.06	124.35	201
66	62.34	34.02	112	61.86	80.12	157	61.04	125.35	202
67	62.33	35.02	113	61.85	81.12	158	61.02	126.36	203
68	62.33	36.02	114	61.83	82.13	159	61.00	127.37	204
69	62.32	37.02	115	61.82	83.13	160	60.98	128.37	205
70	62.31	38.02	116	61.80	84.13	161	60.96	129.38	206
71	62.31	39.02	117	61.78	85.14	162	60.94	130.39	207
72	62.30	40.02	118	61.77	86.14	163	60.92	131.40	208
73	62.29	41.02	119	61.75	87.15	164	60.90	132.41	209
74	62.28	42.03	120	61.74	88.15	165	60.87	133.41	210
75	62.28	43.03	121	61.72	89.15	166	60.85	134.42	211
76	62.27	44.03	122	61.70	90.16	167	60.83	135.43	212
77	62.26	45.03							

**Comparison of Heads of Water in Feet with Pr
Various Units.**

One foot of water at 39°.1 Fahr. = 62.425 lbs. on the squar
" " " = 0.4335 lbs. on the squa
" " " = 0.0295 atmosphere;
" " " = 0.8826 inch of mercury
" " = 773.3 { feet of air at 32° a atmospheric

on the square foot, at 39°.1 Fahr........ = 0.01602 foot of water;
on the square inch " .... = 2.307 feet of water;
nosphere of 29.922 inches of mercury.... = 33.9 " " "
h of mercury at 32°.1.... .... = 1.133 " " "
t of air at 32 deg., and one atmosphere.. = 0.001293 " " "
t of average sea-water........ .... = 1.026 foot of pure water;
t of water at 62° F....................... = 62.355 lbs. per sq. foot;
" " " 62° F............. ... ..·.. = 0.43302 lbs. per sq. inch;
h of water at 62° F................ .. .. = 0.036085 " " " "
und of water on the square inch at 62° F. = 2.3094 feet of water.

sure in Pounds per Square Inch for Different Heads of Water.

° F. 1 foot head = 0.433 lb. per square inch, .433 × 144 = 62.352 lbs. bic foot.

eet.	0	1	2	3	4	5	6	7	8	9
		0.433	0.866	1.299	1.732	2.165	2.598	3.031	3.464	3.897
	4.330	4.763	5.196	5.629	6.062	6.495	6.928	7.361	7.794	8.227
	8.660	9.093	9.526	9.959	10.392	10.825	11.258	11.691	12.124	12.557
	12.990	13.423	13.856	14.289	14.722	15.155	15.588	16.021	16.454	16.887
	17.320	17.753	18.186	18.619	19.052	19.485	19.918	20.351	20.784	21.217
	21.650	22.083	22.516	22.949	23.382	23.815	24.248	24.681	25.114	25.547
	25.980	26.413	26.846	27.279	27.712	28.145	28.578	29.011	29.444	29.877
	30.310	30.743	31.176	31.609	32.042	32.475	32.908	33.341	33.774	34.207
	34.640	35.073	35.506	35.939	36.372	36.805	37.238	37.671	38.104	38.537
	38.970	39.403	39.836	40.269	40.702	41.135	41.568	42.001	42.436	42.867

l in Feet of Water, Corresponding to Pressures in Pounds per Square Inch.

er square inch = 2.30947 feet head, 1 atmosphere = 14.7 lbs. per sq. 33.94 ft. head.

e.	0	1	2	3	4	5	6	7	8	9
		2.309	4.619	6.928	9.238	11.547	13.857	16.166	18.476	20.785
	23.0947	25.404	27.714	30.023	32.333	34.642	36.952	39.261	41.570	43.880
	46.1894	48.499	50.808	53.118	55.427	57.737	60.046	62.356	64.665	66.975
	69.2841	71.594	73.903	76.213	78.522	80.831	83.141	85.450	87.760	90.069
	92.3788	94.688	96.998	99.307	101.62	103.93	106.24	108.55	110.85	113.16
	115.4735	117.78	120.09	122.40	124.71	126.02	129.33	131.64	133.95	136.26
	138.5682	140.88	143.19	145.50	147.81	150.12	152.42	154.73	157.04	159.35
	161.6629	163.97	166.28	168.59	170.90	173.21	175.52	177.83	180.14	182.45
	184.7576	187.07	189.38	191.69	194.00	196.31	198.61	200.92	203.23	205.54
	207.8523	210.16	212.47	214.78	217.09	219.40	221.71	224.02	226.33	228.64

sure of Water due to its Weight.—The pressure of still n pounds per square inch against the sides of any pipe, channel, or f any shape whatever is due solely to the "head," or height of the rface of the water above the point at which the pressure is con-, and is equal to .43302 lb. per square inch for every foot of head, 5 lbs. per square foot for every foot of head (at 62° F.).

pressure per square inch is equal in all directions, downwards, up- or sideways, and is independent of the shape or size of the containing

pressure against a vertical surface, as a retaining-wall, at any point ect ratio to the head *above that point*, increasing from 0 at the l *to a maximum at the bottom.* The total pressure against a v *a unit's breadth increases as* the area of a right-angled t whose perpendicular represents the height of the strip and whose represents the pressure on a unit of surface at the bottom; that is, creases as the square of the depth. The sum of all the horizontal pre is represented by the area of the triangle, and the resultant of this equal to this sum exerted at a point one third of the height from the b (The centre of gravity of the area of a triangle is one third of its heig

The horizontal pressure is the same if the surface is inclined inst vertical.

(For an elaboration of these principles see Trautwine's Pocket-B the chapter on Hydrostatics in any work on Physics. For dams, ret walls, etc., see Trautwine.)

The amount of pressure on the interior walls of a pipe has no appr effect upon the amount of flow.

Buoyancy.—When a body is immersed in a liquid, whether it sink, it is buoyed up by a force equal to the weight of the bulk of th displaced by the body. The weight of a floating body is equal to the of the bulk of the liquid that it displaces. The upward pressure or ancy of the liquid may be regarded as exerted at the centre of gra the displaced water, which is called the centre of pressure or of buo A vertical line drawn through it is called the axis of buoyancy or tion. In a floating body at rest a line joining the centre of gravity centre of buoyancy is vertical, and is called the axis of equilibrium. an external force causes the axis of equilibrium to lean, if a vertical drawn upward from the centre of buoyancy to this axis, the point cuts the axis is called the *metacentre*. If the metacentre is above th of gravity the distance between them is called the metacentric heig the body is then said to be in stable equilibrium, tending to retur original position when the external force is removed.

Boiling-point.—Water boils at 212° F. (100° C.) at mean atmo pressure at the sea-level, 14.696 lbs. per square inch. The temper which water boils at any given pressure is the same as the tempera saturated steam at the same pressure. For boiling-point of water pressure than 14.696 lbs. per square inch, see table of the Prope Saturated Steam.

The Boiling-point of Water may be Raised.—When is entirely freed of air, which may be accomplished by freezing or the cohesion of its atoms is greatly increased, so that its temperatu be raised over 50° above the ordinary boiling-point before ebullitio place. It was found by Faraday that when such air-freed water d the rupture of the liquid was like an explosion. When water is surr by a film of oil, its boiling temperature may be raised considerab its normal standard. This has been applied as a theoretical explan the instance of boiler-explosions.

The freezing-point also may be lowered, if the water is perfectly – 10° C., or 18° Fahrenheit below the normal freezing-point. (H Smith, Jr., on Hydraulics, p. 13.) The density of water at 14° F. is . density at 39°. 1 being 1, and at 32°, .99987.

Freezing-point.—Water freezes at 32° F. at the ordinary atmo pressure, and ice melts at the same temperature. In the melting of of ice into water at 32° F. about 142 heat-units are absorbed, or latent: and in freezing 1 lb. of water into ice a like quantity of heat out to the surrounding medium.

Sea-water freezes at 27° F. The ice is fresh. (Trautwine.)

Ice and Snow. (From Clark.)—1 cubic foot of ice at 32° F. 57.50 lbs.; 1 pound of ice at 32° F. has a volume of .0174 cu. ft. = 30.0

Relative volume of ice to water at 32° F., 1.0855, the expansion in into the solid state being 8.55%. Specific gravity of ice = 0.922, w 62° F. being 1.

At high pressures the melting-point of ice is lower than 32° F., the rate of .0133° F. for each additional atmosphere of pressure

The specific heat of ice is .504, that of water being 1.

1 cubic foot of fresh snow, according to humidity of atmosphere: 12 lbs. 1 cubic foot of snow moistened and compacted by rain: 1 50 lbs. (Trautwine).

Specific Heat of Water. (From Clark's Steam-engine.)— lated by means of Regnault's formula, $c = 1 + 0.00004t + 0.0000$ *which c is the* specific heat of water at any temperature t in centigr *grees, the* specific heat at the freezing-point being 1.

r.	British Thermal Units per pound, above 32° F.	Specific Heat at the given Temperature.	Mean Specific Heat between 32° F. and the given Temp.	Temperatures. Cent.	Temperatures. Fahr.	British Thermal Units per pound, above 32° F.	Specific Heat at the given Temperature.	Mean Specific Heat between 32° F. and the given Temp.
	0.000	1.0000		120°	248°	217.449	1.0177	1.0067
	18.004	1.0005	1.0002	130	266	235.791	1.0204	1.0076
	36.018	1.0012	1.0005	140	284	254.187	1.0232	1.0087
	54.047	1.0020	1.0009	150	302	272.628	1.0262	1.0097
	72.090	1.0030	1.0013	160	320	291.132	1.0294	1.0109
	90.157	1.0042	1.0017	170	338	309.690	1.0328	1.0121
	108.247	1.0056	1.0023	180	356	328.320	1.0364	1.0133
	126.378	1.0072	1.0030	190	374	347.004	1.0401	1.0146
	144.508	1.0089	1.0035	200	392	365.760	1.0440	1.0160
	162.686	1.0109	1.0042	210	410	384.588	1.0481	1.0174
	180.900	1.0130	1.0050	220	428	403.488	1.0524	1.0189
	199.152	1.0153	1.0058	230	446	422.478	1.0568	1.0204

essibility of Water.—Water is very slightly compressible. ssibility is from .000040 to .000051 for one atmosphere, decreasing se of temperature. For each foot of pressure distilled water will hed in volume .0000015 to .0000013. Water is so incompressible t a depth of a mile a cubic foot of water will weigh only about d more than at the surface.

THE IMPURITIES OF WATER.

. E. Hunt and G. H. Clapp, Trans. A. I. M. E. xvii. 338.)

ial analyses are made to determine concerning a given water: cability for making steam; (2) its hardness, or the facility with ll "form a lather" necessary for washing; or (3) its adaptation nufacturing purposes.

ffalo meeting of the Chemical Section of the A. A. A. S. it was de- ort all water analyses in parts per thousand, hundred-thousand,

rt grains per imperial (British) gallons into parts per 100,000, di- To convert parts per 100,000 into grains per U. S. gallon, mul- 2 or .583.

common commercial analysis of water is made to determine its making steam. Water containing more than 5 parts per 100,000 huric or nitric acid is liable to cause serious corrosion, not only al of the boiler itself, but of the pipes, cylinders, pistons, and which the steam comes in contact.

residue in water used for making steam causes the interior lin- llers to become coated, and often produces a dangerous hard prevents the cooling action of the water from protecting the st burning.

magnesia bicarbonates in water lose their excess of carbonic ling, and often, especially when the water contains sulphuric ce, with the other solid residues constantly being formed by the , a very hard and insoluble scale. A larger amount than 100 00,000 of total solid residue will ordinarily cause troublesome should condemn the water for use in steam-boilers, unless a ly cannot be obtained.

wing is a tabulated form of the causes of trouble with water for oses, and the proposed remedies, given by Prof. L. M. Norton.

CAUSES OF INCRUSTATION.

tion of suspended matter.
tion of deposited salts from concentration.
tion of carbonates *of lime and magnesia* by boiling off *holds them in solution.*

4. Deposition of sulphates of lime, because sulphate of lime is b soluble in cold water, less soluble in hot water, insoluble above 27

5. Deposition of magnesia, because magnesium salts decompo temperature.

6. Deposition of lime soap, iron soap, etc., formed by saponif grease.

MEANS FOR PREVENTING INCRUSTATION.

1. Filtration.
2. Blowing off.
3. Use of internal collecting apparatus or devices for directin culation.
4. Heating feed-water.
5. Chemical or other treatment of water in boiler.
6. Introduction of zinc into boiler.
7. Chemical treatment of water outside of boiler.

TABULAR VIEW.

Troublesome Substance.	Trouble.	Remedy or Pal
Sediment, mud, clay, etc.	Incrustation.	Filtration; blowi
Readily soluble salts.	"	Blowing off.
Bicarbonates of lime, magnesia, iron.	"	Heating feed. A caustic soda, magnesia, etc.
Sulphate of lime.	"	Addition of ca barium chlori
Chloride and sulphate of magnesium.	Corrosion.	Addition of car soda, etc.
Carbonate of soda in large amounts.	Priming.	Addition of bar ride, etc.
Acid (in mine waters).	Corrosion.	Alkali.
Dissolved carbonic acid and oxygen.	"	Heating feed. A caustic soda, lime, etc.
Grease (from condensed water).	"	Slacked lime and Carbonate of Substitute mi
Organic matter (sewage).	Priming.	Precipitate with ferric chloride
Organic matter.	Corrosion.	Ditto.

The mineral matters causing the most troublesome boiler-scales bonates and sulphates of lime and magnesia, oxides of iron and and silica. The analyses of some of the most common and tro boiler-scales are given in the following table:

Analyses of Boiler-scale. (Chandler.)

	Sulphate of Lime.	Magnesia.	Silica.	Peroxide of Iron.	Water
N. Y. C. & H. R. Ry., No. 1	74.07	9.19	0.65	0.08	1.14
" " " No. 2	71.37	....	1.76	....	....
" " " No. 3	62.86	18.95	2.60	0.92	1.28
" " " No. 4	53.05	....	4.79	....	....
" " " No. 5	46.83	....	5.32	....	....
" " " No. 6	30.80	31.17	7.75	1.08	2.41
" " " No. 7	4.95	2.61	2.07	1.03	0.63
" " " No. 8	0.88	2.84	0.65	0.36	0.13
" " " No. 9	4.81	....	2.92	....	....
" " " No. 10	30.07	....	8.24	....	....

in Parts per 100,000 of Water giving Bad Results in Steam-boilers. (A. E. Hunt.)

	Bicarbonate of Lime deposited on Boiling	Bicarbonate of Magnesia depos'd on Boil'g	Total Lime.	Total Magnesia.	Sulphuric Acid.	Chlorine.	Iron.	Organic Matter.	Alumina.	Chloride of Sodium.
ater..............	110	25	119	39	890	590	780	30	640	......
....	151	38	1.90	48	360	990	38	21	30	13.10
....	75	89	95	120	310	21	75	10	80	36
a River..........	130	21	161	33	210	38	70	....	....	......
"	80	70	94	81	219	210	90	....	....	......
"	32	82	61	1.04	28	1.90	38	...	...	......
., near Oil-works	30	50	41	68	600	42	23	....	...	......

stances have been added with the idea of causing chemical h will prevent boiler-scale. As a general rule, these do more ;ood, for a boiler is one of the worst possible places in which to emical reaction, where it nearly always causes more or less the metal, and is liable to cause dangerous explosions.

vhere water containing large amounts of total solid residue is used, a heavy petroleum oil, free from tar or wax, which is not by acids or alkalies, not having sufficient wax in it to cause n, and which has a vaporizing-point at nearly 600° F., will give sults in preventing boiler-scale. Its action is to form a thin over the boiler linings, protecting them largely from the action ne water and greasing the sediment which is formed, thus pre- formation of scale and keeping the solid residue from the of the water in such a plastic suspended condition that it can ected from the boiler by the process of "blowing off." If the blown off sufficiently often, this sediment forms into a "putty" essitate cleaning the boilers. Any boiler using bad water should every twelve hours.

ss of Water.—The hardness of water, or its opposite quality, the ease with which it will form a lather with soap, depends ether upon the presence of compounds of lime and magnesia. oaps consist, chemically, of oleate, stearate, and palmitate, of ase, usually soda and potash. The more lime and magnesia in a ter, the more soap a given volume of the water will decompose, insoluble oleate, palmitate, and stearate of lime and magnesia, ently the more soap must be added to a gallon of water in order ssary quantity of soap may remain in solution to form the lather.

hardness of samples of water is generally expressed in terms er of standard soap-measures consumed by a gallon of water in rmanent lather.

ard soap-measure is the quantity required to precipitate one onate of lime.

only reckoned that one gallon of pure distilled water takes one e to produce a lather. Therefore one is deducted from the of soap-measures found to be necessary to use to produce a allon of water, in reporting the number of soap-measures, or f hardness of the water sample. In actually making tests for e "miniature gallon," or seventy cubic centimetres, is used the inconvenient larger amount. The standard measure is made ly dissolving ten grammes of pure castile soap (containing 60 per) in a litre of weak alcohol (of about 35 per cent alcohol). This tion containing exactly sufficient soap in one cubic centime[illegible] on to precipitate one milligramme of carbonate of lim[illegible] the standard soap *solution is* reduced to terms of the [illegible] *of water taken.*

harged with a bicarbonate of lime, magnesia, or iron it will, on the excess of the carbonic acid being expelled, deposit a co erable quantity of the lime, magnesia, or iron, and consequently the will be softer. The hardness of the water after this deposit of lime, long boiling, is called the *permanent hardness* and the difference betw and the total hardness is called *temporary hardness*.

Lime salts in water react immediately on soap-solutions, precipitatin oleate, palmitate, or stearate of lime at once. Magnesia salts, on the trary, require some considerable time for reaction. They are, how more powerful hardeners; one equivalent of magnesia salts consumi much soap as one and one-half equivalents of lime.

The presence of soda and potash salts softens rather than hardens Each grain of carbonate of lime per gallon of water causes an incr expenditure for soap of about 2 ounces per 100 gallons of water. (*E News*, Jan. 31, 1885.)

Purifying Feed-water for Steam-boilers.—To effect purification of water before and after being fed into a boiler, a device ufactured by the Albany Steam Trap Company, Albany, N. Y. re the impurities by the process of a continuous circulation of the water the boiler, through the filter and back into the boiler. The scale fo impurities that are held in suspension are thus brought in contact and "arrested" by the filtering agent contained in the filter while pressure, and at a temperature limited only by that contained in the b

It is sometimes desirable, in the removal of the sulphates and carbo from the feed-water, to heat the water up to nearly the same temper as it is in the boiler, and then to filter the same before feeding it in boiler. The operation in a general way is: The water is first forced in usual exhaust-heater by the feed-pump, and there it is heated by th haust from the engine, say to 200°, and at this temperature it enters th heater. The reheater consists of a vertical, cylindrical shell contain series of water pans or shelves, and so arranged that as the water en it delivered into the top pan, and then overflows into the second, and down the series to the bottom, and during its transit deposits the forming material. The circulating-pump takes the water from the bott the reheater and forces it through the filter on its way into the boiler.

Mr. W. B. Coggswell, of the Solvay Process Co.'s Soda Works in Syra N. Y., thus describes the system of purification of boiler feed-water at these works (Trans. A. S. M. E., xiii. 255):

For purifying, we use a weak soda liquor, containing about 12 to 15 Na_2Co_3 per litre. Say $1\frac{1}{2}$ to 2 M^3 (or 397 to 530 gals.) of this liquor into the precipitating tank. Hot water about 60° C. is then turned the reaction of the precipitation goes on while the tank is filling, whi quires about 15 minutes. When the tank is full the water is filtered thr the Hyatt (4), 5 feet diameter, and the Jewell (1), 10 feet diameter, filt 30 minutes. Forty tanks treated per 24 hours.

Charge of water purified at once............ ...	35 M^3, 9,275 gallon
Soda in purifying reagent.......................	15 kgs. Na_2CO_3.
Soda used per 1,000 gallons..	3.5 lbs.

A sample is taken from each boiler every other day and tested fo Baumé, soda and salt. If the deg. B is more than 2, that boiler is blo reduce it below 2 deg. B.

The following are some analyses given by Mr. Coggswell:

	Lake Water, grams per litre.	Mud from Hyatt Filter.	Scale from Boiler-tube.	Sc... fo... Pu...
Calcium sulphate.............	.261	3.70	51.24	[illegible]
Calcium chloride....	.186	....	....	.....
Calcium carbonate..........	.091	63.37	19.76	[illegible]
Magnesium carbonate........	.015	1.11	25.21	.....
Magnesium chloride..........	.087	....	....	.....
Salt, NaCl	.63	....	.14	.....
Silica..........	....	15.17	2.29	
Iron and aluminum oxide..	....	3.75	1.10	[illegible]
Tot[illegible]	1.270	87.10	99.74	[illegible]

Hard Water for Locomotive Use.—A water-soft- operation at Fossil, in Western Wyoming, on the Union Pa- is described in *Eng'g News*, June 9, 1892. It is the invention nell, of Kansas City. The general plan adopted is to first dis- icals in a closed tank, and then connect this to the supply main tents will be forced into the main tank, the supply-pipe being at thorough mixture of the solution with the water is ob- ste-pipe from the bottom of the tank is opened from time to ff the precipitate. The pipe leading to the tender is arranged ter from near the surface.

k 24 feet in diameter and 16 feet high will contain about 46,600 er. About three hours should be allowed for this amount of through the tank to insure thorough precipitation, giving a nsumption of about 15,000 gallons per hour. Should more quired, auxiliary settling-tanks should be provided.

ls added to precipitate the scale-forming impurities are so- e and quicklime, varying in proportions according to the rela- ns of sulphates and carbonates in the water to be treated. m carbonate is added to produce just enough sodium sulphate ith the remaining lime and magnesia sulphate and produce its corresponding magnesia salt, thereby to get rid of the te, which produces foaming, if allowed to accumulate.

YDRAULICS—FLOW OF WATER.

for Discharge of Water though Orifices and rectangular or circular orifices, with the head measured from rifice to the surface of the still water in the feeding reservoir.

$$Q = C\sqrt{2gH} \times a. \quad \ldots\ldots \quad (1)$$

no allowance for increased head due to velocity of approach:

$$Q = C\tfrac{2}{3}\sqrt{2gH} \times LH. \quad \ldots\ldots \quad (2)$$

ar and circular or other shaped vertical or inclined orifices; on the proposition that each successive horizontal layer of through the orifice has a velocity due to its respective head:

$$Q = cL\tfrac{2}{3}\sqrt{2g} \times (\sqrt{Hb^3} - \sqrt{Ht^3}). \quad \ldots\ldots \quad (3)$$

ar vertical weirs:

$$Q = c\tfrac{2}{3}\sqrt{2gH} \times Lh. \quad \ldots\ldots \quad (4)$$

y of water discharged in cubic feet per second; C = approxi- nt for formulas (1) and (2); c = correct coefficient for (3)

e coefficients c and C are given below.

$2g$ = 8.02; H = head in feet measured from centre of orifice ll water; Hb = head measured from bottom of orifice; Ht = d from top of orifice; $h = H$, corrected for velocity of ap- $H + \frac{4}{3}\frac{Va^2}{2g}$; a = area in square feet; L = length in feet.

Water from Orifices.—The theoretical velocity of water an orifice is the same as the velocity of a falling body which n a height equal to the head of water, $= \sqrt{2gH}$. The actual e smaller section of the *vena contracta* is substantially the theoretical, but the velocity at the plane of the

hich the coefficient C has the nearly constant value eter of the *vena contracta* is therefore about .79 e the *approximate coefficient* = .62, and c the c cient, the ratio $\frac{C}{c}$ varies with different ratios of the head to the dia

of the vertical orifice, or to $\frac{H}{D}$. Hamilton Smith, Jr., gives the follow

For $\frac{H}{D}$ = .5 .875 .1 1.5 2. 2.5 5. 1(

$\frac{C}{c}$ = .9604 .9849 .9918 .9965 .9980 .9987 .9997 1.

For vertical rectangular orifices of ratio of head to width W:

For $\frac{H}{W}$ = .5 .6 .8 .1 1.5 2. 3. 4. 5.

$\frac{C}{c}$ = .9428 .9657 .9823 .9890 .9953 .9974 .9988 .9993 .9996

For $H \div D$ or $H \div W$ over 8, $C = c$, practically.

Weisbach gives the following values of c for circular orifices in a thin
H = measured head from centre of orifice.

D ft.	H ft.						
	.066	.33	.82	2.0	3.0	45.	[illegible]
.033	.711	.665	.637	.628	.641	.632	[illegible]
.066			.629	.621			
.10			.622	.614			
.13			.614	.607			

For an orifice of D = .033 ft. and a well-rounded mouthpiece, H bein
effective head in feet,

H =	.066	1.64	11.5	56	338
c =	.959	.967	.975	.994	.994

Hamilton Smith, Jr., found that for great heads, 312 ft. to 336 ft., with
verging mouthpieces, c has a value of about one, and for small circ
orifices in thin plates, with full contraction, c = about .60. Some of
Smith's experimental values of c for orifices in thin plates discharging
air are as follows. All dimensions in feet.

Circular, in steel, D = .020,	H =	.739	2.43	3.19		
	c =	.6495	.6298	.6264		
Circular, in brass, D = .050,	H =	.185	.536	1.74	2.73	3.57
	c =	.6525	.6265	.6113	.6070	[illegible]
Circular, in brass, D = .100,	H =	.129	.457	.900	1.73	2.05
	c =	.6337	.6155	.6096	.6042	[illegible]
Circular, in iron, D = .100,	H =	1.80	1.81	2.81	4.68	
	c =	.6061	.6041	.6033	.6026	
Square, in brass, .05 × .05,	H =	.313	.877	1.79	2.81	3.70
	c =	.6410	.6238	.6157	.6127	[illegible]
Square, in brass, .10 × .10,	H =	.181	.939	1.71	2.75	3.74
	c =	.6292	.6139	.6084	.6076	[illegible]
Rectangular, in brass, L = .300, W = .050	H =	.261	.917	1.82	2.83	3.73
	c =	.6476	.6280	.6203	.6180	[illegible]

For the rectangular orifice, L, the length, is horizontal.

Mr. Smith, as the result of the collation of much experimental dat
others as well as his own, gives tables of the value of c for vertical orif
with full contraction, with a free discharge into the air, with the inner
of the plate, in which the orifice is pierced, plane, and with sharp
corners, so that the escaping vein only touches these inner edges. Th
tables are abridged below. The coefficient c is to be used in the formula
and (4) above. For formulæ (1) and (2) use the coefficient C found from
values of $\frac{C}{c}$ above.

[V]alues of Coefficient *c* for Vertical Orifices with Sharp Edges, Full Contraction, and Free Discharge into Air. (Hamilton Smith, Jr.)

[Head] of Orifice H.	Square Orifices. Length of the Side of the Square, in feet.												
	.02	.03	.04	.05	.07	.10	.12	.15	.20	.40	.60	.80	1.0
.4			.643	.637	.628	.621	.616	.611					
.6	.660	.645	.636	.630	.623	.617	.613	.610	.605	.601	.598	.596	
[illegible].0	.648	.636	.628	.622	.618	.613	.610	.608	.605	.603	.601	.600	.599
[illegible]	.632	.622	.616	.612	.609	.607	.606	.606	.605	.605	.604	.603	.603
[illegible]	.623	.616	.612	.609	.607	.605	.605	.605	.604	.604	.603	.602	.602
[illegible]	.616	.611	.608	.606	.605	.604	.604	.603	.603	.603	.602	.602	.601
[illegible]	.606	.605	.604	.603	.602	.602	.602	.602	.602	.601	.601	.601	.600
[illegible]	.599	.598	.598	.598	.598	.598	.598	.598	.598	.598	.598	.598	.598
[H].	Circular Orifices. Diameters, in feet.												
	.02	.03	.04	.05	.07	.10	.12	.15	.20	.40	.60	.80	1.0
.4				.637	.628	.618	.612	.606					
.6	.655	.640	.630	.624	.618	.613	.609	.605	.601	.596	.593	.590	
[illegible].0	.644	.631	.623	.617	.612	.608	.605	.603	.600	.598	.595	.593	.591
[illegible]	.632	.621	.614	.610	.607	.604	.601	.600	.599	.599	.597	.596	.595
[illegible]	.623	.614	.609	.605	.603	.602	.600	.599	.599	.598	.597	.597	.596
[illegible]	.618	.611	.607	.604	.602	.600	.599	.599	.598	.598	.597	.596	.596
[illegible]	.611	.606	.603	.601	.599	.598	.598	.597	.597	.597	.596	.596	.595
[illegible]	.601	.600	.599	.598	.597	.596	.596	.596	.596	.596	.596	.595	.594
[illegible]	.596	.596	.595	.595	.594	.594	.594	.594	.594	.594	.594	.593	.593
[illegible]	.593	.593	.592	.592	.592	.592	.592	.592	.592	.592	.592	.592	.592

HYDRAULIC FORMULÆ.—FLOW OF WATER IN OPEN AND CLOSED CHANNELS.

Flow of Water in Pipes.—The quantity of water discharged [thr]ough a pipe depends on the "head;" that is, the vertical distance be[twe]en the level surface of still water in the chamber at the entrance end of [the] pipe and the level of the centre of the discharge end of the pipe; [als]o upon the length of the pipe, upon the character of its interior surface [as] to smoothness, and upon the number and sharpness of the bends: but [it] is independent of the position of the pipe, as horizontal, or inclined [up]wards or downwards.

The head, instead of being an actual distance between levels, may be [cau]sed by pressure, as by a pump, in which case the head is calculated as a [ver]tical distance corresponding to the pressure 1 lb. per sq. in. = 2.309 ft. [he]ad, or 1 ft. head = .433 lb. per sq. in.

The total head operating to cause flow is divided into three parts: 1. The [vel]ocity-head, which is the height through which a body must fall *in vacuo* [to a]cquire the velocity with which the water flows into the pipe $= v^2 \div 2g$, in [wh]ich v is the velocity in ft. per sec. and $2g = 64.32$; 2. the *entry-head*, that [re]quired to overcome the resistance to entrance to the pipe. With sharp-[ed]ged entrance the entry-head = about ½ the velocity-head; with smooth [rou]nded entrance the entry-head is inappreciable; 3. the *friction-head*, due [to] the frictional resistance to flow within the pipe.

In ordinary cases of pipes of considerable length the sum of the entry and [vel]ocity heads required scarcely exceeds 1 foot. In the case of long pipes [wit]h low heads the sum of the velocity and entry heads is generally so small [th]at it may be neglected.

General Formula for Flow of Water in [...] Conduits.

Mean velocity in ft. per sec. $= c\sqrt{\text{mean hy[...]}}$ × slope

Do. for pipes running full $= c\sqrt{\frac{\text{diamete[...]}}{4}}$ [...]

[in] *which c is a coefficient determined by experim[...]*

The mean hydraulic radius $= \dfrac{\text{area of wet cross-section}}{\text{wet perimeter.}}$

In pipes running full, or exactly half full, and in semicircular open nels running full it is equal to ¼ diameter.

The slope = the head (or pressure expressed as a head, in feet) ÷ length of pipe measured in a straight line from end to

In open channels the slope is the actual slope of the surface, or its fa unit of length, or the sine of the angle of the slope with the horizon.

If r = mean hydraulic radius, s = slope = head ÷ length, v = veloc feet per second (all dimensions in feet), $v = c\sqrt{r}\sqrt{s} = c\sqrt{rs}$.

Quantity of Water Discharged.—If Q = discharge in cubic per second and a = area of channel, $Q = av = ac\sqrt{rs}$.

$a\sqrt{r}$ is approximately proportional to the discharge. It is a maxim 308°, corresponding to 19/20 of the diameter, and the flow of a conduit full is about 5 per cent greater than that of one completely filled.

Table giving Fall in Feet per Mile, the Distance on Sl corresponding to a Fall of 1 Ft., and also the Va of s and $\sqrt{s}$ for Use in the Formula $v = c\sqrt{rs}$.

$s = H \div L$ = sine of angle of slope = fall of water-surface (H), in any tance (L), divided by that distance.

Fall in Feet per Mi.	Slope, 1 Foot in	Sine of Slope, s.	$\sqrt{s}$.	Fall in Feet per Mi.	Slope, 1 Foot in	Sine of Slope, s.	$\sqrt{\ }$
0.25	21120	.0000473	.006881	17	310.6	.0032197	[illegible]
.30	17600	.0000568	.007538	18	293.3	.0034091	[illegible]
.40	13200	.0000758	.008704	19	277.9	.0035985	[illegible]
.50	10560	.0000947	.009731	20	264	.0037879	[illegible]
.60	8800	.0001136	.010660	22	240	.0041667	[illegible]
.702	7520	.0001330	.011532	24	220	.0045455	[illegible]
.805	6560	.0001524	.012347	26	203.1	.0049242	[illegible]
.904	5840	.0001712	.013085	28	188.6	.0053030	[illegible]
1.	5280	.0001894	.013762	30	176	.0056818	[illegible]
1.25	4224	.0002367	.015386	35.20	150	.0066667	[illegible]
1.5	3520	.0002841	.016854	40	132	.0075758	[illegible]
1.75	3017	.0003314	.018205	44	120	.0083333	[illegible]
2	2640	.0003788	.019463	48	110	.0090909	[illegible]
2.25	2347	.0004261	.020641	52.8	100	.010	.1
2.5	2112	.0004735	.021760	60	88	.0113636	[illegible]
2.75	1920	.0005208	.022822	66	80	.0125	[illegible]
3.	1760	.0005682	.023837	70.4	75	.0133333	[illegible]
3.25	1625	.0006154	.024807	80	66	.0151515	[illegible]
3.5	1508	.0006631	.025751	88	60	.0166667	[illegible]
3.75	1408	.0007102	.026650	96	55	.0181818	[illegible]
4	1320	.0007576	.027524	105.6	50	.02	[illegible]
5	1056	.0009470	.030773	120	44	.0227273	[illegible]
6	880	.0011364	.03371	132	40	.025	[illegible]
7	754.3	.0013257	.036416	160	33	.0303030	[illegible]
8	660	.0015152	.038925	220	24	.0416667	[illegible]
9	586.6	.0017044	.041286	264	20	.05	[illegible]
10	528	.0018939	.043519	330	16	.0625	[illegible]
11	443.6	.0020833	.045643	440	12	.0833333	[illegible]
12	440	.0022727	.047673	528	10	.1	[illegible]
13	406.1	.0024621	.04962	660	8	.125	[illegible]
14	377.1	.0026515	.051493	880	6	.1666667	[illegible]
15	352	.0028409	.0533	1056	5	.2	[illegible]
16	330	.0030303	.055048	1320	4	.25	.5

of $\sqrt{r}$ for Circular Pipes, Sewers, and Conduits of different Diameters.

n hydraulic depth $= \frac{\text{area}}{\text{perimeter}} = \frac{1}{4}$ diam. for circular pipes run- or exactly half full.

$\sqrt{r}$ in Feet.	Diam., ft.	in.	$\sqrt{r}$ in Feet.	Diam., ft.	in.	$\sqrt{r}$ in Feet.	Diam., ft.	in.	$\sqrt{r}$ in Feet.
.088	2		.707	4	6	1.061	9		1.500
.102	2	1	.722	4	7	1.070	9	3	1.521
.125	2	2	.736	4	8	1.080	9	6	1.541
.144	2	3	.750	4	9	1.089	9	9	1.561
.161	2	4	.764	4	10	1.099	10		1 581
.177	2	5	.777	4	11	1.109	10	3	1.601
.191	2	6	.790	5		1.118	10	6	1.620
.204	2	7	.804	5	1	1.127	10	9	1.639
.228	2	8	.817	5	2	1.137	11		1.658
.251	2	9	.829	5	3	1.146	11	3	1.677
.290	2	10	.842	5	4	1.155	11	6	1.696
.323	2	11	.854	5	5	1.164	11	9	1.714
.354	3		.866	5	6	1.173	12		1.732
.382	3	1	.878	5	7	1.181	12	3	1.750
.408	3	2	.890	5	8	1.190	12	6	1.768
.433	3	3	.901	5	9	1.199	12	9	1.785
.456	3	4	.913	5	10	1.208	13		1.083
.479	3	5	.924	5	11	1.216	13	3	1.820
.500	3	6	.935	6		1.225	13	6	1.837
.520	3	7	.946	6	3	1.250	14		1.871
.540	3	8	.957	6	6	1.275	14	6	1.904
.559	3	9	.968	6	9	1.299	15		1 936
.577	3	10	.979	7		1 323	15	6	1.968
.595	3	11	.990	7	3	1.346	16		2.
.612	4		1.	7	6	1.369	16	6	2.031
.629	4	1	1.010	7	9	1.392	17		2.061
.646	4	2	1.021	8		1.414	17	6	2.091
.661	4	3	1.031	8	3	1.436	18		2.121
.677	4	4	1.041	8	6	1.458	19		2.180
.692	4	5	1.051	8	9	1.479	20		2.236

s of the Coefficient *c*. (Chiefly condensed from P. J. Flynn f Water.)—Almost all the old hydraulic formulæ for finding the ocity in open and closed channels have constant coefficients, and are correct for only a small range of channels. They have often been give incorrect results with disastrous effects. Ganguillet and Kut- ughly investigated the American, French, and other experiments, ave as the result of their labors the formula now generally known 's formula. There are so many varying conditions affecting the ater, that all hydraulic formulæ are only approximations to the sult.

he surface-slope measurement is good, Kutter's formula will give dom exceeding 7½% error, provided the rugosity coefficient of the is known for the site. For small open channels D'Arcy's and rmulæ, and for cast-iron pipes D'Arcy's formulæ, are generally as being approximately correct.

r's Formula for measures in feet is

$$v = \left\{ \frac{\frac{1.811}{n} + 41.6 + \frac{.00281}{s}}{1 + \left(41.6 + \frac{.00281}{s}\right) \times \frac{n}{\sqrt{r}}} \right\} \times \sqrt{rs},$$

v = mean *velocity in feet per second*; $r = \frac{a}{p}$ = hydraulic m depth in feet = area of cross-section in square feet divided by wett eter in lineal feet; s = fall of water-surface (h) in any distance by that distance, $= \frac{h}{l}$, = sine of slope; n = the coefficient of rug pending on the nature of the lining or surface of the channel. If first term of the right-hand side of the equation equal c, we have formula, $v = c\sqrt{rs} = c \times \sqrt{r} \times \sqrt{s}$.

Values of n in Kutter's Formula.—The accuracy of Ku mula depends, in a great measure, on the proper selection of the c of roughness n. Experience is required in order to give the right this coefficient, and to this end great assistance can be obtained, i this selection, by consulting and comparing the results obtained periments on the flow of water already made in different channels.

In some cases it would be well to provide for the contingency deterioration of channel, by selecting a high value of n, as, for where a dense growth of weeds is likely to occur in small channels, where channels are likely not to be kept in a state of good repair.

The following table, giving the value of n for different material piled from Kutter, Jackson, and Hering, and this value of n applie each instance, to the surfaces of other materials equally rough.

VALUE OF n IN KUTTER'S FORMULA FOR DIFFERENT CHANNE

n = .009, well-planed timber, in perfect order and alignment; o perhaps .01 would be suitable.

n = .010, plaster in pure cement; planed timber; glazed, coate amelled stoneware and iron pipes; glazed surfaces of every sort i order.

n = .011, plaster in cement with one third sand, in good condition iron, cement, and terra cotta pipes, well joined, and in best order.

n = .012, unplaned timber, when perfectly continuous on the flumes.

n = .013, ashlar and well-laid brickwork; ordinary metal; ear stoneware pipe in good condition, but not new; cement and terra-c not well jointed nor in perfect order, plaster and planed wood in or inferior condition; and, generally, the materials mentioned with when in imperfect or inferior condition.

n = .015, second class or rough-faced brickwork; well-dressed sto foul and slightly tuberculated iron; cement and terra-cotta pipes, perfect joints and in bad order; and canvas lining on wooden fram

n = .017, brickwork, ashlar, and stoneware in an inferior condi berculated iron pipes; rubble in cement or plaster in good order; fi well rammed, ⅓ to ⅔ inch diameter; and, generally, the materi tioned with n = .013 when in bad order and condition.

n = .020, rubble in cement in an inferior condition; coarse rubb set in a normal condition; coarse rubble set dry; ruined brick masonry; coarse gravel well rammed, from 1 to 1½ inch diameter with beds and banks of very firm, regular gravel, carefully trim rammed in defective places; rough rubble with bed partially cove silt and mud; rectangular wooden troughs, with battens on the in inches apart; trimmed earth in perfect order.

n = .0225, canals in earth above the average in order and regimen

n = .025, canals and rivers in earth of tolerably uniform cross- slope and direction, in moderately good order and regimen, and f stones and weeds.

n = .0275, canals and rivers in earth below the average in order a men.

n = .030, canals and rivers in earth in rather bad order and regim ing stones and weeds occasionally, and obstructed by detritus.

n = .035, suitable for rivers and canals with earthen beds in bad o regimen, and having stones and weeds in great quantities.

n = .05, torrents encumbered with detritus.

Kutter's formula has the advantage of being easily adapted to a in the surface of the pipe exposed to the flow of water, by a chang value of n. For cast-iron pipes it is usual to use n = .013 to provide future deterioration of the surface.

Reducing Kutter's formula to the form $v = c \times \sqrt{r} \times \sqrt{s}$, and takin coefficient of roughness in the formula = .011, .012, and .013, and s = have the following values of the coefficient c for different diam conduit.

of c in Formula $v = c \times \sqrt{r} \times \sqrt{s}$ for Metal Pipes and Moderately Smooth Conduits Generally.

By KUTTER'S FORMULA. (s = .001 or greater.)

r.	n = .011	n = .012	n = .013	Diameter.	n = .011	n = .012	n = .013
	c =	c =	c =	ft.	c =	c =	c =
	47.1	....	....	7	152.7	139.2	127.9
	61.5	....	....	8	155.4	141.9	130.4
	77.4	....	....	9	157.7	144.1	132.7
	87.4	77.5	69.5	10	159.7	146	134.5
	105.7	94.6	85.3	11	161.5	147.8	136.2
	116.1	104.3	94.4	12	163	149.3	137.7
	123.6	111.3	101.1	14	165.8	152	140.4
	133.6	120.8	110.1	16	168	154.2	142.1
	140.4	127.4	116.5	18	169.9	156.1	144.4
	145.4	132.3	121.1	20	171.6	157.7	146
	149.4	136.1	124.8				

ular pipes the hydraulic mean depth r equals ¼ of the diameter.
ng to Kutter's formula the value of c, the coefficient of discharge,
e for all slopes greater than 1 in 1000; that is, within these limits
nt. We further find that up to a slope of 1 in 2640 the value of c
practical purposes, constant, and even up to a slope of 1 in 5000
ence in the value of c is very little. This is exemplified in the
:

of c for Different Values of $\sqrt{r}$ and s in Kutter's Formula, with n = .013.

$$v = c\sqrt{r} \times \sqrt{s}.$$

Slopes.				
in 1000	1 in 2500	1 in 3333.3	1 in 5000	1 in 10,000
93.6	91.5	90.4	88.4	83.3
116.5	115.2	114.4	113.2	109.7
142.6	142.8	143.0	143.1	143.8

ability of the values of the coefficient of Kutter's formula for
ess than 6 in. diameter is considered doubtful. (See note under
age 564.)

of c for Earthen Channels, by Kutter's Formula, for Use in Formula $v = c\sqrt{rs}$.

Coefficient of Roughness, n = .0225.					Coefficient of Roughness, n = .035.				
$\sqrt{r}$ in feet.					$\sqrt{r}$ in feet.				
0.4	1.0	1.8	2.5	4.0	0.4	1.0	1.8	2.5	4.0
c	c	c	c	c	c	c	c	c	c
35.7	62.5	80.3	89.2	99.9	19.7	37.6	51.6	59.3	69.2
35.5	62.3	80.3	89.3	100.2	19.6	37.6	51.6	59.4	69.4
35.2	62.1	80.3	89.5	100.6	19.4	37.4	51.6	59.5	69.8
34.6	61.7	80.3	89.8	101.4	19.1	37.1	51.6	59.7	70.4
34.	61.2	80.3	90.1	102.2	18.8	36.9	51.6	59.9	71.0
33.	60.5	80.3	90.7	103.7	18.3	36.4	51.6	60.4	72.2
31.6	59.4	80.3	91.5	106.0	17.6	35.8	51.6	60.9	73.9
30.5	58.5	80.3	92.3	107.9	17.1	35.3	51.6	60.5	75.4
28.5	*56.7*	*80.2*	*93.9*	*112.2*	16.2	34.3	51.6	[illegible]	[illegible]
27.4	*55.7*	*80.2*	*94.8*	*115.0*	15.6	33.8	51.5	[illegible]	[illegible]

Mr. Molesworth, in the 22d edition of his "Pocket-book of Eng[...] Formulæ," gives a modification of Kutter's formula as follows: For cast-iron pipes, $v = c\sqrt{rs}$, in which

$$c = \frac{181 + \frac{.00281}{s}}{1 + \frac{.026}{\sqrt{d}}\left(41.6 + \frac{.00281}{s}\right)},$$

in which d = diameter of the pipe in feet.

(This formula was given incorrectly in Molesworth's 21st edition.)

Molesworth's Formula.—$v = \sqrt{krs}$, in which the values o[...] as follows:

Nature of Channel.	Values of k for Velocities	
	Less than 4 ft. per sec.	More th[...] 4 ft. per [...]
Brickwork..............................	8800	8500
Earth..................................	7200	6800
Shingle................................	6400	5900
Rough, with bowlders...................	5300	4700

In very large channels, rivers, etc., the description of the channel [...] the result so slightly that it may be practically neglected, and k assu[...] from 8500 to 9000.

Flynn's Formula.—Mr. Flynn obtains the following expres[...] the value of Kutter's coefficient for a slope of .001 and a value of n =[...]

$$c = \frac{183.72}{1 + \left(44.41 \times \frac{.013}{\sqrt{r}}\right)}.$$

The following table shows the close agreement of the values of c ob[...] from Kutter's, Molesworth's, and Flynn's formulæ:

Diameter.	Slope.	Kutter.	Molesworth.
6 inches	1 in 40	71.50	71.48
6 inches	1 in 1000	69.50	69.79
4 feet	1 in 400	117.	117.
4 feet	1 in 1000	116.5	116.55
8 feet	1 in 700	130.5	130.68
8 feet	1 in 2600	129.8	129.93

Mr. Flynn gives another simplified form of Kutter's formula for us[...] different values of n as follows:

$$v = \left(\frac{K}{1 + \left(44.41 \times \frac{n}{\sqrt{r}}\right)}\right)\sqrt{rs}.$$

In the following table the value of K is given for the several values o[...]

n	K	n	K	n	K	n	K	n
.009	245.63	.012	195.33	.015	165.14	.018	145.03	.021
.010	225.51	.013	183.72	.016	157.6	.019	139.73	.022
.011	209.05	.014	137.77	.017	150.94	.020	134.96	.023

If in the application of Mr. Flynn's formula given above within the[...] of n as given in the table, we substitute for n, K, and $\sqrt{r}$ their va[...] have a [...]m of Kutter's formula.

e, when $n = .011$, and $d = 3$ feet, we have

$$v = \frac{209.05}{1 + \left(44.41 \times \frac{.011}{.866}\right)} \times \sqrt{rs}.$$

Formulæ:
ven surfaces, fine plastered sides and bed, planed planks, etc.,

$$v = \sqrt{1 \div .0000045\left(10.16 + \frac{1}{r}\right)} \times \sqrt{rs}.$$

rfaces such as cut-stone, brickwork, unplaned planking, mortar,

$$v = \sqrt{1 \div .000013\left(4.354 + \frac{1}{r}\right)} \times \sqrt{rs}.$$

uneven surfaces, such as rubble masonry:

$$v = \sqrt{1 \div .00006\left(1.219 + \frac{1}{r}\right)} \times \sqrt{rs}.$$

surfaces, such as earth:

$$v = \sqrt{1 \div .00035\left(0.2438 + \frac{1}{r}\right)} \times \sqrt{rs}.$$

tion of Bazin's formula, known as D'Arcy's Bazin's:

$$v = r\sqrt{\frac{1000s}{.08534r + 0.35}}.$$

l channels of less than 20 feet bed Bazin's formula for earthen good order gives very fair results, but Kutter's formula is super- almost all countries where its accuracy has been investigated. able on p. 561 shows the value of c, in Kutter's formula, for a wide hannels in earth, that will cover anything likely to occur in the ractice of an engineer.

's Formula for clean iron pipes under pressure is

$$v = \left\{\frac{rs}{.00007726 + \frac{.00000162}{r}}\right\}^{\frac{1}{2}}$$

nodification of D'Arcy's formula is

$$v = \left(\frac{155256d}{12d + 1}\right)^{\frac{1}{2}} \times \sqrt{rs}$$

= diameter in feet.

formula, as given by J. B. Francis, C.E., for old cast-iron pipe, deposit and under pressure, is

$$v = \left(\frac{144d^2s}{.0082(12d + 1)}\right)^{\frac{1}{2}}.$$

nodification of D'Arcy's *formula for* old cast-iron pipe is

$$v = \left(\frac{70243.9d}{12d + 1}\right)^{\frac{1}{2}} \times \sqrt{rs}.$$

For Pipes Less than 5 inches in Diameter, coeffici[…] in the formula $v = c\sqrt{rs}$, from the formula of D'Arcy, Kutter, and Fa[…]

Diam. in inches.	D'Arcy, for Clean Pipes.	Kutter, for $n = .011$ $s = .001$	Fanning, for Clean Iron Pipes	Diam. in inches	D'Arcy, for Clean Pipes.	Kutter, for $n = .011$ $s = .001$
3/8	59.4	32.		1¾	90.7	58.8
½	65.7	36.1		2	92.9	61.5
¾	74.5	42.6		2½	96.1	66.
1	80.4	47.4	80.4	3	98.5	70.1
1¼	84.8	51.9		4	101.7	77.4
1½	88.1	55.4	88.	5	103.8	82.9

Mr. Flynn, in giving the above table, says that the facts show that t[…] efficients diminish from a diameter of 5 inches to smaller diameters, […] is a safer plan to adopt coefficients varying with the diameter than […] stant coefficient. No opinion is advanced as to what coefficients sho[…] used with Kutter's formula for small diameters. The facts are s[…] stated, giving the results of well-known authors.

Older Formulæ.—The following are a few of the many formu[…] flow of water in pipes given by earlier writers. As they have constan[…] ficients, they are not considered as reliable as the newer formulæ.

Prony, $v = 97\sqrt{rs} - .08$;

Eytelwein, $v = 50\sqrt{\dfrac{dh}{l + 50d}}$, or $v = 108\sqrt{rs} - 0.13$;

Hawksley, $v = 48\sqrt{\dfrac{dh}{l + 54d}}$; Neville, $v = 140\sqrt{rs} - 11\sqrt[3]{r[…]}$

In these formulæ d = diameter in feet; h = head of water in fe[…] length of pipe in feet; s = sine of slope $= \frac{h}{l}$; r = mean hydraulic […] = area ÷ wet perimeter $= \frac{d}{4}$ for circular pipe.

Mr. Santo Crimp (*Eng'g*, August 4, 1893) states that observations o[…] in brick sewers show that the actual discharge is 33% greater than th[…] culated by Eytelwein's formula. He thinks Kutter's formula not su[…] to D'Arcy's for brick sewers, the usual coefficient of roughness i[…] former, viz., .013, being too low for large sewers and far too small in th[…] of small sewers.

D'Arcy's formula for brickwork is

$$v = \frac{\sqrt{2g}}{m} rs; \quad m = a\left(1 + \frac{B}{r}\right); \quad a = .0037285; \quad B = .229663.$$

VELOCITY OF WATER IN OPEN CHANNELS.

Irrigation Canals.—The minimum mean velocity required to pr[…] the deposit of silt or the growth of aquatic plants is in Northern […] taken at 1½ feet per second. It is stated that in America a higher ve[…] is required for this purpose, and it varies from 2 to 3½ feet per second. […] maximum allowable velocity will vary with the nature of the soil o[…] bed. A sandy bed will be disturbed if the velocity exceeds 3 fee[…] second. Good loam with not too much sand will bear a velocity of 4 […] per second. The Cavour Canal in Italy, over a gravel bed, has a veloci[…] about 5 per second. (Flynn's "Irrigation Canals.")

Mean Surface and Bottom Velocities.—According to the […] mula of Baz[…]

$$v = v_{max} - 25.4\sqrt{rs}; \quad v = v_b + 10.87\sqrt{rs}.$$

10.87 $\sqrt{rs}$, in which v = mean velocity in feet per second, um surface velocity in feet per second, vb = bottom velocity ond, r = hydraulic mean depth in feet = area of cross-section divided by wetted perimeter in feet, s = sine of slope.
locity, or that of the particles in contact with the bed, is ch less than the mean velocity as the greatest velocity is he mean.
es that in ordinary cases the velocities may be taken as bear- her nearly the proportions of 3, 4, and 5. In very slow cur- nearly as 2, 3, and 4.
om and Mean Velocities.—Ganguillet & Kutter give able of safe bottom and mean velocity in channels, calculated ula $v = vb + 10.87 \sqrt{rs}$:

ial of Channel.	Safe Bottom Veloc ity vb, in feet per second.	Mean Velocity v, in feet per second.
rth	0.249	0.328
. . . .	0.499	0.656
. . . .	1.000	1.312
. . . .	1.998	2.625
. . . .	2.999	3.938
flint	4.003	5.579
, soft slate.	4.988	6.564
. . . .	6.006	8.204
. . . .	10.009	13.127

& Kutter state that they are unable for want of observations far these figures are trustworthy. They consider them to be portionately small than too large, and therefore recommend onfidently.
ing at a high velocity and carrying large quanties of silt is very channels, even when constructed of the best masonry.
ce of Soils to Erosion by Water.—W. A. Burr, *Eng'g* 1894, gives a diagram showing the resistance of various soils to wing water.
ts show that a velocity greater than 1.1 feet per second will while pure clay will stand a velocity of 7.35 feet per second. he proportion of clay carried by any soil, the higher the per- ity. Mr. Burr states that experiments have shown that the line e power of soils to resist erosion is parabolic. From his dia- lowing figures are selected representing different classes of

nd resists erosion by flow of	1.1	feet per second.
soil, 15% clay	1.2	" "
oam, 40% clay	1.8	" "
soil, 65% clay	3.0	" "
am, 85% clay	4.8	" "
tural clay, 95% clay	6.2	" "
. . . .	7.35	" "

g and Transporting Power of Water.—Prof. J. is "Elements of Geology," states:
power of water, or its power of overcoming cohesion, varies as the velocity of the current.
orting power of a current varies as the sixth power of the ve- If the velocity therefore be increased ten times, the transport- increased 1,000,000 times. A current running three feet per bout two miles per hour, will bear fragments of stone of the s egg, or about three ounces weight. A current of ten miles an ar fragments of one and a half tons, and a torrent of twenty r will carry fragments of 100 tons.
orting power of water must not be confounded with its erosive resistance to be overcome in the one case is weight, ir ion; the latter *varies as the* square: the former as the *velocity.*
ses of removal of slightly cohering material, the resista

mixture of these two resistances, and the power of removing ma[...] vary at some rate between v^2 and v^6.

Baldwin Latham has found that in order to prevent deposits of [...] in small sewers or drains, such as those from 6 inches to 9 inches [...] a mean velocity of not less than 3 feet per second should be [...] Sewers from 12 to 24 inches diameter should have a velocity of not [...] 2½ feet per second, and in sewers of larger dimensions in no case [...] velocity be less than 2 feet per second.

The specific gravity of the materials has a marked effect upon [...] velocities necessary to move them. T. E. Blackwell found that [...] sp. gr. of 1.26 was moved by a current of from 1.25 to 1.50 ft. per [...] while stones of a sp. gr. of 2.32 to 3.00 required a velocity of 2.5 to [...] second.

Chailly gives the following formula for finding the velocity req[...] move rounded stones or shingle :

$$v = 5.67 \sqrt{ag},$$

in which v = velocity of water in feet per second. a = average dia[...] feet of the body to be moved, g = its specific gravity.

Geo. Y. Wisner, *Eng'g News*, Jan 10, 1895, doubts the general ac[...] statements made by many authorities concerning the rate of flow [...] rent and the size of particles which different velocities will move.

The scouring action of any river, for any given rate of current, [...] inverse function of the depth. The fact that some engineer has f[...] a given velocity of current on some stream of unknown depth [...] sand or gravel has no bearing whatever on what may be expecte[...] rents of the same velocity in streams of greater depths. In chann[...] ft. deep a mean velocity of 3 to 5 ft. per second may produce rapid [...] while in depths of 18 ft. and upwards current velocities of 6 to [...] second often have no effect whatever on the channel bed.

Grade of Sewers.—The following empirical formula is give[...] meister's "Cleaning and Sewerage of Cities," for the minimum gr[...] sewer of clear diameter equal to d inches, and either circular o[...] section :

$$\text{Minimum grade, in per cent,} = \frac{100}{5d + 50}.$$

As the lowest limit of grades which can be flushed, 0.1 to 0.2 per [...] be assumed for sewers which are sometimes dry, while 0.3 per cent [...] able for the trunk sewers in large cities. The sewers should r[...] rarely as possible.

Relation of Diameter of Pipe to Quantity Discha[...] In many cases which arise in practice the information sought is th[...] ter necessary to supply a given quantity of water under a given he[...] diameter is commonly taken to vary as the two-fifth power of [...] charge. This is almost certainly too large. Hagen's formula, wi[...] Unwin's coefficients, give $d = c\left(\frac{Q}{\left(\frac{h}{l}\right)^{\frac{1}{2}}}\right)^{.387}$, where c = .239 when [...] are in feet and cubic feet per second.

Mr. Thrupp has proposed a formula which makes d vary as the .3[...] of the discharge, and the formula of M. Vallot, a French engineer, [...] vary as the .375 power of the discharge. (*Engineering.*)

FLOW OF WATER—EXPERIMENTS AND TAB[...]

The Flow of Water through New Cast-iron Pi[...] recently measured by S. Bent Russell, of the St. Louis, Mo., Water[...] The pipe was 12 inches in diameter, 1631 feet long, and laid on a [...] grade from end to end. Under an average total head of 3.36 feet [...] was 43,200 cubic feet in seven hours; under an average head of 3.3[...] flow was the same; under an average total head of 3.41 feet the [...] 46,700 cubic feet in 8 hours and 35 minutes. Making allowance [...] of head due to entrance and to curves, it was found that the value [...] the formula $v = c\sqrt{rs}$ was from 88 to 93. (*Eng'g Record*, April 14, [...]

[...] Water in a 20-inch Pipe 75,000 Feet Lo[...] [...]xperimental data with calculations by different [...]

s. B. Brush, Trans. A. S. C. E., 1888. The pipe experimented t supplying the city of Hoboken, N. J.

TAINED BY THE HACKENSACK WATER COMPANY, FROM 1882–1887, ING THROUGH A 20-IN. CAST-IRON MAIN 75,000 FEET LONG.

bs. per sq. in. at pumping-station:						
100	105	110	115	120	125	130
e head in feet:						
66	77	89	100	112	123	135
U. S. gallons in 24 hours, 1 = 1000:						
3,165	3,354	3,566	3,804	3,904	4,116	4,255
ity in main in feet per second:						
2.24	2.36	2.52	2.68	2.76	2.92	3.00
consumed in delivering each million gals. at given velocities:						
$8.15	$8.00	$8.10	$8.30	$8.60	$9.00	$9.60
discharge by D'Arcy's formula:						
3,004	3,244	3,488	3,699	3,915	4,102	4,297

s in Smooth Cast-iron Water-pipes from 1 Foot eet in Diameter, on Hydraulic Grades of 0.5 o 8 Feet per Mile; with Corresponding Values $V = c\sqrt{rs}$. (D. M. Greene, in *Eng'g News*, Feb. 24, 1894.)

Hydraulic Grade; Feet per Mile = h.

$h = 0.5$ = 0.0000947	1.0 0.0001894	1.5 0.0002841	2.0 0.0003788	3.0 0.0005682	4.0 0.0007576
$V =$ 0.4542	0.6673	0.8356	0.9803	1.2277	1.4402
$c =$ 92.7	97.0	99.1	100.7	103.0	104.7
$V =$ 0.7359	·1.0793	1.3516	1.5856	1.9857	2.3294
$c =$ 106.6	110.9	113.4	115.2	117.9	119.7
$V =$ 0.9733	1.4298	1.7906	2.1017	2.6306	3.0860
$c =$ 115.5	119.9	122.6	124.4	127.5	129.5
$V =$ 1.1883	1.7456	2.1861	2.5645	3.2116	3.7676
$c =$ 122.1	126.8	129.7	131.8	134.7	136.9
$V =$ 1.3872	2.0379	2.5521	2.9939	3.7493	4.3983
$c =$ 127.5	132.4	135.5	137 6	140.7	142.9
$V =$ 1.5742	2.3126	2.8961	3.3975	4.2548	4.9913
$c =$ 132.1	137.8	140.3	142.6	145.8	148.1
$V =$ 1.7518	2.5736	3.2230	3.7809	4.7350	5.5546
$c =$ 135.9	141.4	146.0	146.8	150.2	152.5
$V =$ 1.9218	2.8234	3.5358	4.1479	5.1945	6.0936
$c =$ 139.7	145 1	148.4	150.7	154.1	156.5
$V =$ 2.0854	3.0638	3.8368	4.5010	5.6368	6.6125
$c =$ 142.9	148.4	151.7	154.2	157.6	160.1

ties in this table have been calculated by Mr. Greene's modifi- e Chezy formula, which modification is found to give results by from 1.29 to – 2.65 per cent (average 0.9 per cent) from very asured flows in pipes from 16 to 48 inches in diameter, on grades et to 10.296 feet per mile, and in which the velocities ranged o 6.195 feet per second. The only assumption made is that the mula for V gives correct results in conduits from 4 feet to 9 eter, as it is known to do in conduits less than 4 feet in diameter. cles on Flow of Water in long tubes are to be found in *Eng'g* ows: G. B. Pearsons, *Sept. 23, 18.6;* E. Sherman Gould, Feb. 16, 16, *and 23, 1889; J. L. Fitzgerald*, Sept. 6 and 13, 1890; Jas. Duane *J. T. Fanning, July 14, 1892;* A. N. Talbot, Aug. 11, 1892.

Flow of Water in Circular Pipes, [illegible] Full, Based on Kutter's Formula, [illegible]

Discharge in cubic feet per second.

Diameter.	Slope, or Head Divided by Length					
	1 in 40	1 in 70	1 in 100	1 in 200	[illegible]	[illegible]
5 in.	.456	.344	.288	[illegible]	[illegible]	[illegible]
6 "	.762	.576	.482	[illegible]	[illegible]	[illegible]
7 "	1.17	.889	.744	[illegible]	[illegible]	[illegible]
8 "	1.70	1.29	1.08	[illegible]	[illegible]	[illegible]
9 "	2.37	1.79	1.50	[illegible]	[illegible]	[illegible]
Slope....	1 in 60	1 in 80	1 in 100	1 in 200	[illegible]	[illegible]
10 in.	[illegible]	2.24	2.01	1.42	[illegible]	[illegible]
11 "	3.84	2.94	2.63	1.86	[illegible]	[illegible]
12 "	4.82	3.74	3.35	2.37	[illegible]	[illegible]
13 "	5.88	4.66	4.16	2.95	2.40	[illegible]
14 "	6.60	5.72	5.15	3.62	[illegible]	[illegible]
Slope....	1 in 100	1 in 200	1 in 300	1 in 400	[illegible]	[illegible]
15 in.	6.18	4.37	3.57	3.09	[illegible]	[illegible]
16 "	7.68	5.32	4.26	3.69	[illegible]	[illegible]
18 "	10.21	7.22	5.89	5.10	[illegible]	[illegible]
20 "	13.65	9.65	7.88	6.82	6.10	[illegible]
22 "	17.71	12.52	10.22	8.85	[illegible]	[illegible]
Slope....	1 in 200	1 in 400	1 in 600	1 in 800	1 in 1000	[illegible]
2 ft.	15.88	11.23	9.17	7.94	7.10	[illegible]
2 ft. 2 in.	19.73	13.96	11.39	9.87	8.82	[illegible]
2 " 4 "	24.15	17.07	13.94	12.07	10.80	[illegible]
2 " 6 "	29.08	20.56	16.79	14.54	13.00	[illegible]
2 " 8 "	34.71	24.54	20.04	17.35	15.52	[illegible]
Slope....	1 in 500	1 in 750	1 in 1000	1 in 1250	1 in 1500	[illegible]
2 ft. 10 in.	25.84	21.10	18.27	16.34	14.92	[illegible]
3 "	30.14	24.61	21.31	19.06	17.40	[illegible]
3 " 2 in.	34.90	28.50	24.68	22.07	20.15	[illegible]
3 " 4 "	40.08	32.72	28.34	25.35	23.14	[illegible]
3 " 6 "	45.66	37.28	32.28	28.87	26.35	[illegible]
Slope....	1 in 500	1 in 750	1 in 1000	1 in 1250	1 in 1500	1 in 17[illegible]
3 ft. 8 in.	51.74	42.52	36.59	32.72	29.87	27.66
3 " 10 "	58.36	47.65	41.27	36.91	33.69	31.20
4 "	65.47	53.46	46.30	41.41	37.80	34.50
4 " 6 in.	89.75	73.28	63.47	56.76	51.82	47.97
5 "	118.9	97.09	84.08	75.21	68.65	63.56
Slope....	1 in 750	1 in 1000	1 in 1500	1 in 2000	1 in 2500	1 in 3000
5 ft. 6 in.	125.2	108.4	88.54	76.67	68.58	62.60
6 "	157.8	136.7	111.6	96.66	86.45	78.92
6 " 6 "	195.0	168.8	137.9	119.4	106.8	97.49
7 "	237.7	205.9	168.1	145.6	130.2	118.8
7 " 6 "	285.3	247.1	201.7	174.7	156.3	142.6
Slope....	1 in 1500	1 in 2000	1 in 2500	1 in 3000	1 in 3500	1 in 4000
8 ft.	239.4	207.3	195.4	169.3	156.7	146.6
8 " 6 in.	281.1	243.5	217.8	198.8	184.0	172.2
9 "	327.0	283.1	253.3	231.2	214.0	200.2
9 " 6 "	376.9	326.4	291.9	266.5	246.7	230.8
10 "	431.4	373.6	334.1	305.0	282.4	264.2

For U. S. gallons multiply the figures in the table by 7.48
[illegible] a given diameter the quantity of flow varies as the [illegible]
[illegible] the slope. From this principle the flow for other [illegible]

in the table may be found. Thus, what is the flow for a pipe 8 feet ter, slope 1 in 125? From the table take $Q = 207.3$ for slope 1 in 2000. ven slope 1 in 125 is to 1 in 2000 as 16 to 1, and the square root of this s 4 to 1. Therefore the flow required is $207.3 \times 4 = 829.2$ cu. ft.

Circular Pipes, Conduits, etc., Flowing Full.

es of the factor $ac\sqrt{r}$ in the formula $Q = ac\sqrt{r} \times \sqrt{s}$ correspond- different values of the coefficient of roughness, n. (Based on Kutter's la.)

Value of $ac\sqrt{r}$.					
$n = .010.$	$n = .011.$	$n = .012.$	$n = .013.$	$n = .015.$	$n = .017.$
6.906	6.0627	5.3800	4.8216	3.9604	3.329
21.25	18.742	16.708	15.029	12.421	10.50
46.93	41.487	37.149	33.497	27.803	23.60
86.05	76.347	68.44	61.867	51.600	43.93
141.2	125.60	112.79	102.14	85.496	72.99
214.1	190.79	171.66	155.68	130.58	111.8
307.6	274.50	247.33	224.63	188.77	164
421.9	377.07	340.10	309.23	260.47	223.9
559.6	500.78	452.07	411.27	347.28	299.3
722.4	647.18	584.90	532.76	451.23	388.8
911.8	817.50	739.59	674.09	570.90	493.3
1128.9	1013.1	917.41	836.69	709.56	613.9
1374.7	1234.4	1118.6	1021.1	866.91	750.8
1652.1	1484.2	1345.9	1229.7	1045	906
1962.8	1764.3	1600.9	1463.9	1245.3	1080.7
2682.1	2413.3	2193	2007	1711.4	1487.3
3543	3191.8	2903.6	2659	2272.7	1977
4557.8	4111.9	3742.7	3429	2934.8	2557.2
5731.5	5176.3	4713.9	4322	3702.3	3232.5
7075.2	6394.9	5825.9	5339	4588.3	4010
8595.1	7774.3	7087	6510	5591.6	4893
10296	9318.3	8501.8	7814	6717	5884.2
12196	11044	10083	9272	7978.3	6995.3
14298	12954	11832	10889	9377.9	8226.3
16604	15049	13751	12663	10917	9580.7
19118	17338	15847	14597	12594	11061
21858	19834	18134	16709	14426	12678
24823	22534	20612	18996	16412	14424
28020	25444	23285	21464	18555	16333
31482	28593	26179	24139	20879	18395
35156	31987	29254	26981	23352	20584
39104	35529	32558	30041	26012	22938
43307	39358	36077	33301	28850	25451
47751	43412	39802	36752	31860	28117
52491	47739	43773	40432	35073	30965
57496	52308	47969	44322	38454	33975
62748	57103	52382	48413	42040	37147
74191	67557	62008	57343	49823	44073
86769	79050	72594	67140	58387	51669
100617	91711	84247	77932	67839	60067
115769	105570	96991	89759	78201	69301
132133	120570	110905	102559	89423	79259

v of Water in Circular Pipes, Conduits, etc., Flowing under Pressure.

d on D'Arcy's formulæ for the flow of water through cast-iron pipes. comparison of results obtained by Kutter's formula, with $n = .013$. ensed from Flynn on Water Power.)

es of a, and also the values of the factors $c\sqrt{r}$ and $ac\sqrt{r}$ for use in mulæ $Q = av$; $v = c\sqrt{r} \times \sqrt{s}$, and $Q = ac\sqrt{r} \times \sqrt{s}$.

Q = discharge in cubic feet per second, a = area in square feet, v = vel-
ity in feet per second, r = mean hydraulic depth, ¼ diam. for pipes run-
full, s = sine of slope.

(For values of $\sqrt{s}$ see page 558.)

Size of Pipe.			Clean Cast-iron Pipes.		Value of $ac\sqrt{r}$ by Kutter's Formula, when $n = .013$.	Old Cast-iron Pipes Lined with Deposit.	
d = diam. in ft.	in.	a = area in square feet.	For Velocity, $c\sqrt{r}$.	For Discharge, $ac\sqrt{r}$.		For Velocity, $c\sqrt{r}$.	For Discharge, $ac\sqrt{r}$.
	⅜	.00077	5.251	.00403		3.532	[illegible]
	½	.00136	6.702	.00914		4.507	[illegible]
	¾	.00307	9.309	.02855		6.261	[illegible]
	1	.00545	11.61	.06334		7.811	[illegible]
	1¼	.00852	13.68	.11659		9.255	[illegible]
	1½	.01227	15.58	.19115		10.48	[illegible]
	1¾	.01670	17.32	.28936		11.65	[illegible]
	2	.02182	18.96	.41357		12.75	[illegible]
	2½	.0341	21.94	.74786		14.76	[illegible]
	3	.0491	24.63	1.2089		16.56	[illegible]
	4	.0873	29.37	2.5630		19.75	[illegible]
	5	.136	33.54	4.5610		22.56	[illegible]
	6	.196	37.28	7.3068	4.822	25 07	[illegible]
	7	.267	40.65	10 852		27.34	[illegible]
	8	.349	43.75	15.270		29.43	[illegible]
	9	.442	46.73	20.652	15.03	31.42	[illegible]
	10	.545	49.45	26.952		33.26	[illegible]
	11	.660	52.16	34.428		35.09	[illegible]
1		.785	54 65	42.918	33.50	36.75	[illegible]
1	2	1.000	59 34	63.435		39 91	[illegible]
1	4	1.396	63.67	88.886		42.53	[illegible]
1	6	1.767	67.75	119.72	102.14	45.57	[illegible]
1	8	2.182	71.71	156.46		48.34	[illegible]
1	10	2.640	75.32	198.83		50.658	[illegible]
2		3.142	78.80	247.57	224.63	52.961	[illegible]
2	2	3.687	82.15	302.90		55.258	[illegible]
2	4	4.276	85.39	365.14		57.436	[illegible]
2	6	4.909	88.39	433.92	411.37	59.455	[illegible]
2	8	5.585	91.51	511.10		61.55	343.8
2	10	6.305	94.40	595.17		63.49	[illegible]
3		7.068	97.17	686.76	674.09	65.35	[illegible]
3	2	7.875	99.93	786.94		67.21	[illegible]
3	4	8.726	102.6	895.7		69	[illegible]
3	6	9.621	105.1	1011.2	1021.1	70.70	[illegible]
3	8	10 559	107.6	1136.5		72.40	[illegible]
3	10	11.541	110.2	1271.4		74.10	[illegible]
4		12.566	112.6	1414.7	1463.9	75.73	[illegible]
4	3	14.186	116.1	1647.6		78.12	1108.2
4	6	15.904	119.6	1901.9	2007	80.43	1279.2
4	9	17.721	122.8	2176.1		82.20	[illegible]
5		19.635	126.1	2476.4	2659	84 83	[illegible]
5	3	21.648	129.3	2799.7		86.99	[illegible]
5	6	23.758	132.4	3146.3	3429	89.07	[illegible]
5	9	25.967	135.4	3516		91.08	[illegible]
6		28.274	138.4	3912.8	4322	93.08	[illegible]
6	6	33.183	144.1	4782.1	5339	96.93	[illegible]
7		38.485	149.6	5757.5	6510	100.61	[illegible]
7	6	44.179	154 9	6841.6	7814	104.11	[illegible]
8		50.266	160	8043	9272	107.61	[illegible]
8	6	56.745	165	9364.7	10889	111	[illegible]
9		[illegible]	169.8	10804	12663	114.2	[illegible]
9	6	[illegible]	174.5	12370	14597	117.4	[illegible]
10		[illegible]	179.1	14066	16709	120.4	[illegible]

e.	Clean Cast-iron Pipes.		Value of $ac\sqrt{r}$ by Kutter's Formula, when $n = .013$	Old Cast-iron Pipes Lined with Deposit.	
area in uare eet.	For Velocity, $c\sqrt{r}$.	For Discharge, $ac\sqrt{r}$.		For Velocity, $c\sqrt{r}$.	For Discharge, $ac\sqrt{r}$.
590	183.6	15893	18996	123.4	10690
033	187.9	17855	21464	126.3	12010
869	192.2	19966	24139	129.3	13429
098	196.3	22204	26981	132	14935
719	200.4	24598	30041	134.8	16545
733	204.4	27134	33301	137.5	18252
139	208.3	29818	36752	140.1	20056
938	212.2	32664	40432	142.7	21971
130	216.0	35660	44322	145.2	23986
715	219.6	38807	48413	147.7	26103
692	223.3	42125	52753	150.1	28335
062	226.9	45621	57343	152.6	30686
825	230.4	49273	62132	155	33144
981	233.9	53082	67140	157.3	35704
529	237.3	57074	72409	159.6	38389
470	240.7	61249	77932	161.9	41199
529	247.4	70154	89759	166.4	47186
159	253.8	79736	102559	170.7	53633

ater in Circular Pipes from ⅜ inch to 12 inches Diameter.

D'Arcy's formula for clean cast-iron pipes. $Q = ac\sqrt{r}\sqrt{s}$.

ia. n.	Slope, or Head Divided by Length of Pipe.							
	1 in 10.	1 in 20.	1 in 40.	1 in 60.	1 in 80.	1 in 100.	1 in 150.	1 in 200.
	Quantity in cubic feet per second.							
⅜	.00127	.00090	.00064	.00052	.00045	.00040	.00033	.00028
½	.00289	.00204	.00145	.00118	.00102	.00091	.00075	.00065
¾	.00903	.00638	.00451	.00369	.00319	.00286	.00233	.00202
1	.02003	.01416	.01001	.00818	.00708	.00633	.00517	.00448
1¼	.03687	.02607	.01843	.01505	.01303	.01166	.00952	.00824
1½	.06044	.04274	.03022	.02468	.02137	.01912	.01561	.01352
1¾	.09140	.06470	.04575	.03736	.03235	.02894	.02363	.02046
2	.13077	.09247	.06539	.05339	.04624	.04136	.03377	.02927
2½	.23647	.16722	.11824	.09655	.08361	.07479	.06106	.05288
3	.38225	.27031	.19113	.15607	.13515	.12089	.09871	.08548
4	.81042	.57309	.40521	.33088	.28654	.25630	.20927	.18123
5	1.4422	1.0198	.72109	.58882	.50992	.45610	.37241	.32251
6	2.3104	1.6338	1.1552	.94331	.81690	.73068	.59660	.51666
7	3.4314	2.4265	1.7157	1.4110	1.2132	1.0852	.88607	.76734
8	4.8284	3.4143	2.4141	1.9713	1.7072	1.5270	1.2468	1.0797
9	6.5302	4.6178	3.2651	2.6662	2.3089	2.0652	1.6862	1.4603
0	8.5222	6.0265	4.2611	3.4795	3.0132	2.6952	2.2006	1.9058
1	10.886	7.6981	5.4431	4.4447	3.8491	3.4428	2.8110	2.4344
2	13.571	9.5965	6.7853	5.5407	4.7982	4.2918	3.5043	3.0347
=	.3162	.2236	.1581	.1291	.1118	.1	.08165	.07071

Value of $ac\sqrt{r}$.	Dia. in.	Slope, or Head Divided by Length of Pipe.						
		1 in 250.	1 in 300.	1 in 350.	1 in 400.	1 in 450.	1 in 500.	1 in 550.
.00403	⅜	.00025	.00023	.00022	.00020	.00019	.00018	.00017
.00914	½	.00058	.00053	.00049	.00046	.00043	.00041	.00039
.02855	¾	.00181	.00165	.00153	.00143	.00134	.00128	.00122
.06334	1	.00400	.00366	.00339	.00317	.00298	.00283	.00270
.11659	1¼	.00737	.00673	.00623	.00583	.00549	.00521	.00497
.19115	1½	.01209	.01104	.01022	.00956	.00901	.00855	.00815
.28936	1¾	.01830	.01671	.01547	.01447	.01363	.01294	.01234
.41357	2	.02615	.02388	.02211	.02068	.01948	.01849	.01763
.74786	2½	.04730	.04318	.03997	.03739	.03523	.03344	.03189
1.2089	3	.07645	.06980	.06462	.06045	.05695	.05406	.05155
2.5630	4	.16208	.14799	.13699	.12815	.12074	.11461	.10929
4.5610	5	.28843	.26335	.24379	.22805	.21487	.20397	.19448
7.3068	6	.46208	.42189	.39055	.36534	.34422	.32676	.31156
10.852	7	.68628	.62660	.58005	.54260	.51124	.48530	.46273
15.270	8	.96567	.88158	.81617	.76350	.71936	.68286	.65111
20.652	9	1.3060	1.1924	1.1038	1.0326	.97292	.92356	.88060
26.952	10	1.7044	1.5562	1.4405	1.3476	1.2697	1.2053	1.1492
34.428	11	2.1772	1.9878	1.8402	1.7214	1.6219	1.5396	1.4680
42.918	12	2.7141	2.4781	2.2940	2.1459	2.0219	1.9193	1.8300
Value of $\sqrt{s}$ =		.06324	.05774	.05345	.05	.04711	.04472	.04264

For U. S. gals. per sec., multiply the figures in the table by......
" " " " min., " " " " ...... [illegible]
" " " " hour, " " " " ...... [illegible]
" " " " 24 hrs., " " " " ...... [illegible]

For any other slope the flow is proportional to the square root [illegible] slope; thus, flow in slope of 1 in 100 is double that in slope of 1 in 400.

Flow of Water in Pipes from ⅜ Inch to 12 In[illegible] Diameter for a Uniform Velocity of 100 Ft. per [illegible]

Diameter in Inches.	Area in Square Feet.	Flow in Cubic Feet per Minute.	Flow in U. S Gallons per Minute.	Flow in Gall[illegible] Ho[illegible]
⅜	.00077	0.077	.57	[illegible]
½	.00136	0.136	1.02	[illegible]
¾	.00307	0.307	2.30	[illegible]
1	.00545	0.545	4.08	[illegible]
1¼	.00852	0.852	6.38	[illegible]
1½	.01227	1.227	9.18	[illegible]
1¾	.01670	1.670	12.50	[illegible]
2	.02182	2.182	16.32	[illegible]
2½	.0341	3.41	25.50	1,5[illegible]
3	.0491	4.91	36.72	2,2[illegible]
4	.0873	8.73	65.28	3,9[illegible]
5	.136	13.6	102.00	6,1[illegible]
6	.196	19.6	146.88	8,8[illegible]
7	.267	26.7	199.92	11,9[illegible]
8	.349	34.9	261.12	15,6[illegible]
9	.442	44.2	330.48	19,8[illegible]
10	.545	54.5	408.00	24,4[illegible]
11	.660	66.0	493.68	29,6[illegible]
12	.785	78.5	587.52	35,2[illegible]

Given the diameter of a pipe, to find the quantity in gallons it will d[illegible] the velocity of flow being 100 ft. per minute. Square the diameter in [illegible] *and multiply by 4.08.*

Q' = quantity in gallons per minute and d = diameter in inches, then

$$Q' = \frac{d^2 \times .7854 \times 100 \times 7.4805}{144} = 4.08d^2.$$

[F]or any other velocity, V', in feet per minute, $Q' = 4.08d^2\frac{V'}{100} = .0408d^2V'$.

[G]iven diameter of pipe in inches and velocity in feet per second, to find [dis]charge in cubic feet and in gallons per minute.

$$Q' = \frac{d^2 \times .7854 \times v \times 60}{144} = 0.32725d^2v \text{ cubic feet per minute.}$$

$$= .32725 \times 7.4805 \text{ or } 2.448d^2v \text{ U. S. gallons per minute.}$$

[T]o find the capacity of a pipe or cylinder in gallons, multiply the square [of] the diameter in inches by the length in inches and by .0034. Or multiply [the] square of the diameter in inches by the length in feet and by .0408.

$$Q = \frac{.7854d^2l}{231} = .0034d^2l \text{ (exact) } .0034 \times 12 = .0408.$$

LOSS OF HEAD.

[T]he loss of head due to friction when water, steam, air, or gas of any kind [flo]ws through a straight tube is represented by the formula

$$h = f\frac{4l}{d}\frac{v^2}{2g}; \qquad \text{whence } v = \sqrt{\frac{64.4}{4f}\frac{hd}{l}},$$

[in] which l = the length and d = the diameter of the tube, both in feet; v = [vel]ocity in feet per second, and f is a coefficient to be determined by experi[me]nt. According to Weisbach, $f = .00644$, in which case

$$\sqrt{\frac{64.4}{4f}} = 50, \text{ and } v = 50\sqrt{\frac{hd}{l}},$$

[wh]ich is one of the older formulæ for flow of water (Downing's). Prof. Un[win] says that the value of f is possibly too small for tubes of small bore, [an]d he would put f = .006 to .01 for 4-inch tubes, and f = .0084 to .012 for 2-[inc]h tubes. Another formula by Weisbach is

$$h = \left(.0144 + \frac{.01716}{\sqrt{v}}\right)\frac{l}{d}\frac{v^2}{2g}.$$

[R]ankine gives

$$f = .005\left(1 + \frac{1}{12d}\right).$$

[F]rom the general equation for velocity of flow of water $v = c\sqrt{r}\sqrt{s}$, = [for] round pipes $c\sqrt{\frac{d}{4}}\sqrt{\frac{h}{l}}$, we have $v^2 = c^2\frac{d}{4}\frac{h}{l}$ and $h = \frac{4lv^2}{c^2d}$, in which [c is] the coefficient c of D'Arcy's, Bazin's, Kutter's, or other formula, as found [by] experiment. Since this coefficient varies with the condition of the inner [sur]face of the tube, as well as with the velocity, it is to be expected that [val]ues of the loss of head given by different writers will vary as much as those [of q]uantity of flow. Two tables for loss of head per 100 ft. in length in pipes [of] different diameters with different velocities are given below. The first [is g]iven by Clark, based on Ellis' and Howland's experiments; the second is [fro]m the Pelton Water-wheel Co.'s catalogue, authority not stated. The [los]s of head as given in these two tables for any given diameter and velocity [diff]ers considerably. Either table should be used with caution and the re[sult]s compared with the quantity of flow for the given diameter and head [as g]iven in the tables of flow based on Kutter's and D'Arcy's formulæ.

Relative Loss of Head by Friction for each 10[illegible] Length of Clean Cast-iron Pipe.

(Based on Ellis and Howland's experiments.)

Velocity in Feet per Second.	Diameter of Pipes in Inches.								
	3	4	5	6	7	8	9	10	12
	Loss of Head in Feet, per 100 Feet Long.								
Feet	Feet of Head	Feet of Head	Feet of Head	Feet of Head	Feet of Head	Feet of Head	Feet of Head	Feet of Head	Feet of Head
2	.97	.55	.41	.32	.27	.23	.19	.18	[illegible]
2.5	1.49	.92	.64	.50	.43	.36	.30	.27	[illegible]
3	1.9	1.2	.82	.72	.61	.51	.44	.39	[illegible]
3.5	2.6	1.6	1.2	1.0	.7	.71	.61	.52	[illegible]
4	3.3	2.2	1.7	1.3	.9	.92	.79	.69	[illegible]
4.5	.....	.....	.....	1.6	1.2	1.2	1.01	.87	[illegible]
5	.....	.....	.....	.....	.....	.....	1.2	1.1	[illegible]
5.5	.....	.....	.....	.....	.....	.....	.....	.....	.....
6	.....	.....	.....	.....	.....	.....	.....	.....	.....
	15	18	21	24	27	30	33	36	42
2	.11	.095	.075	.065	.055	.052	.049	.047	[illegible]
2.5	.17	.147	.117	.109	.088	.085	.076	.067	[illegible]
3	.25	.21	.17	.15	.13	.12	.108	.10	[illegible]
3.5	.34	.29	.23	.20	.18	.16	.15	.14	[illegible]
4	.44	.36	.31	.27	.23	.22	.20	.17	[illegible]
4.5	.56	.46	.39	.34	.30	.28	.25	.22	[illegible]
5	.70	.58	.48	.41	.37	.34	.30	.27	[illegible]
5.5	.84	.70	.59	.50	.44	.39	.36	.32	[illegible]
6	.....	.....	.....	.59	.53	.49	43	.4	[illegible]

Loss of Head in Pipe by Friction.—Loss of head by f[illegible] each 100 feet in length of different diameters of pipe when dischar[illegible] following quantities of water per minute (Pelton Water-wheel Co.):

Velocity in Feet per Second. V	Inside Diameter of Pipe in Inches.									
	1		2		3		4		5	
	Loss of Head in Feet. h	Cubic Feet per Minute. Q	Loss of Head in Feet. h	Cubic Feet per Minute. Q	Loss of Head in Feet. h	Cubic Feet per Minute. Q	Loss of Head in Feet. h	Cubic Feet per Minute. Q	Loss of Head in Feet. h	Cubic Feet per Minute. Q
2.0	2.37	.65	1.185	2.62	.791	5.89	.593	10.4	.474	16.3
3.0	4.89	.99	2.44	3.92	1.62	8.83	1.22	15.7	.978	24.5
4.0	8.20	1.32	4.10	5.23	2.73	11.80	2.05	20.9	1.64	32.7
5.0	12.33	1.65	6.17	6.54	4.11	14.70	3.08	26.2	2.46	40.9
6.0	17.23	1.98	8.61	7.85	5.74	17.70	4.31	31.4	3.45	49.1
7.0	22.89	2.31	11.45	9.16	7.62	20.6	5.72	36.6	4.57	57.2

Inside Diameter of Pipe in Inches.

7		8		9		10		11		12	
h	Q	h	Q	h	Q	h	Q	h	Q	h	Q
.338	32.0	.296	41.9	.264	53	.237	65.4	.216	79.2	.198	94.2
.698	48.1	.611	62.8	.544	79.5	.488	98.2	.444	119	.407	141
1.175	64.1	1.027	83.7	.913	106	.822	131	.747	158	.685	188
1.76	80.2	1.54	105	1.37	132	1.23	163	1.122	198	1.028	235
2.46	96.2	2.15	125	1.92	159	1.71	196	1.56	237	1.43	283
3.26	112.0	2.85	146	2.52	185	2.28	229	2.07	277	1.91	330

Inside Diameter of Pipe in Inches.

13		14		15		16		18		20	
h	Q	h	Q	h	Q	h	Q	h	Q	h	Q
.183	110	.169	128	.158	147	.147	167	.132	212	.119	262
.375	166	.349	192	.325	221	.306	251	.271	318	.245	393
.632	221	.587	256	.548	294	.513	335	.456	424	.410	528
.949	276	.881	321	.822	368	.770	419	.685	530	.617	654
1.325	332	1.229	385	1.148	442	1.076	502	.957	636	.861	785
1.75	387	1.63	449	1.52	515	1.43	586	1.27	742	1.143	916

Inside Diameter of Pipe in Inches.

22		24		26		28		30		36	
h	Q	h	Q	h	Q	h	Q	h	Q	h	Q
.108	316	.098	377	.091	442	.084	513	.079	589	.066	848
.222	475	.204	565	.188	663	.174	770	.163	883	.135	1273
.373	633	.342	754	.315	885	.293	1026	.273	1178	.228	1697
.561	792	.513	942	.474	1106	.440	1283	.411	1472	.342	2121
.782	950	.717	1131	.662	1327	.615	1539	.574	1767	.479	2545
1.040	1109	.953	1319	.879	1548	.817	1796	.762	2061	.636	2868

EXAMPLE.—Given 200 ft. head and 600 ft. of 11-inch pipe, carrying 119 cubic feet of water per minute. To find effective head : In right-hand column, under 11-inch pipe, find 119 cubic ft.; opposite this will be found the loss by friction in 100 ft. of length for this amount of water, which is .444. Multiply this by the number of hundred feet of pipe, which is 6, and we have 2.66 ft., which is the loss of head. Therefore the effective head is 200 − 2.66 = 197.34.

EXPLANATION.—The loss of head by friction in pipe depends not only upon diameter and length, but upon the quantity of water passed through it. The head or pressure is what would be indicated by a pressure-gauge attached to the pipe near the wheel. Readings of gauge should be taken while the water is flowing from the nozzle.

To reduce heads in feet to pressure in pounds multiply by .433. To reduce pounds pressure to feet multiply by 2.309.

Cox's Formula.—Weisbach's formula for loss of head caused by the friction of water in pipes is as follows :

$$\text{Friction-head} = \left(0.0144 + \frac{0.01716}{\sqrt{V}}\right)\frac{L.V^2}{5.367d},$$

where L = length of pipe in feet;
V = velocity of the water in feet per second;
d = diameter of pipe in inches.

William Cox (*Amer. Mach., Dec. 28, 1893*) gives a simpler formula which gives almost *identical results :*

$$H = \textit{friction-head in feet} = \frac{L}{d}\,\frac{4V^2 + 5V - 2}{1200}. \quad \ldots$$

$$\frac{Hd}{L} = \frac{4V^2 + 5V - 2}{1200}. \quad \ldots$$

He gives a table by means of which the value of $\frac{4V^2+5V-2}{1200}$ is at o
obtained when V is known, and *vice versa*.

VALUES OF $\frac{4V^2+5V-2}{1200}$.

V	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
1	.00583	.00695	.00813	.00938	.01070	.01208	.01353	.01505	.01663	[illegible]
2	.02000	.02178	.02363	.02555	.02753	.02958	.03170	.03388	.03613	[illegible]
3	.04083	.04328	.04580	.04838	.05103	.05375	.05653	.05938	.06230	[illegible]
4	.06833	.07145	.07463	.07788	.08120	.08458	.08803	.09155	.09513	[illegible]
5	.10250	.10628	.11013	.11405	.11803	.12208	.12620	.13038	.13463	[illegible]
6	.14333	.14778	.15230	.15688	.16153	.16625	.17103	.17588	.18080	[illegible]
7	.19083	.19595	.20113	.20638	.21170	.21708	.22253	.22805	.23363	[illegible]
8	.24500	.25078	.25663	.26255	.26853	.27458	.28070	.28688	.29313	[illegible]
9	.30583	.31228	.31880	.32538	.33203	.33875	.34553	.35238	.35930	[illegible]
10	.37333	.38045	.38763	.39488	.40220	.40958	.41703	.42455	.43213	[illegible]
11	.44750	.45528	.46313	.47105	.47903	.48708	.49520	.50338	.51163	[illegible]
12	.52833	.53678	.54530	.55388	.56253	.57125	.58003	.58888	.59780	[illegible]
13	.61583	.62495	.63413	.64338	.65270	.66208	.67153	.68105	.69063	[illegible]
14	.71000	.71978	.72963	.73955	.74953	.75958	.76970	.77988	.79013	[illegible]
15	.81083	.82128	.83180	.84238	.85303	.86375	.87453	.88538	.89630	[illegible]
16	.91833	.92945	.94063	.95188	.96320	.97458	.98603	.99755	1.00913	[illegible]
17	1.03250	1.04428	1.05613	1.06805	1.08003	1.09208	1.10420	1.11638	1.12863	[illegible]
18	1.15333	1.16578	1.17830	1.19088	1.20353	1.21625	1.22903	1.24188	1.25480	[illegible]
19	1.28083	1.29395	1.30713	1.32038	1.33370	1.34708	1.36053	1.37405	1.38763	[illegible]
20	1.41500	1.42878	1.44263	1.45655	1.47053	1.48458	1.49870	1.51288	1.52713	[illegible]
21	1.55583	1.57028	1.58480	1.59938	1.61403	1.62875	1.64353	1.65838	1.67330	[illegible]

The use of the formula and table is illustrated as follows:

Given a pipe 5 inches diameter and 1000 feet long, with 49 feet head, w
will the discharge be?

If the velocity V is known in feet per second, the discharge is 0.327
cubic foot per minute.

By equation 2 we have

$$\frac{4V^2+5V-2}{1200} = \frac{Hd}{L} = \frac{49 \times 5}{1000} = 0.245;$$

whence, by table, V = real velocity = 8 feet per second.

The discharge in cubic feet per minute, if V is velocity in feet per se
and d diameter in inches, is $0.32725d^2V$, whence, discharge

$$= 0.32725 \times 25 \times 8 = 65.45 \text{ cubic feet per minute.}$$

The velocity due the head, if there were no friction, is $8.025\sqrt{H} = 5$
feet per second, and the discharge at that velocity would be

$$0.32725 \times 25 \times 56.175 = 460 \text{ cubic feet per minute.}$$

Suppose it is required to deliver this amount, 460 cubic feet, at a vel
of 2 feet per second, what diameter of pipe will be required and what wi
the loss of head by friction?

$$d = \text{diameter} = \sqrt{\frac{Q}{V \times 0.32725}} = \sqrt{\frac{460}{2 \times 0.32725}} = \sqrt{703} = 26.5 \text{ inc}$$

Having now the diameter, the velocity, and the discharge, the friction-l
is calculated by equation 1 and use of the table; thus,

$$H = \frac{L}{d}\,\frac{4V^2+5V-2}{1200} = \frac{1000}{26.5} \times 0.02 = \frac{20}{26.5} = 0.75 \text{ foot,}$$

thus leaving $49 - 0.75$ = say 48 feet effective head applicable to power-
ducing purposes.

Problems of the loss of head may be solved rapidly by means of C
Pipe Computer, a mechanical device on the principle of the slide-rule
sale by Keuffel & Esser, New York.

Heads at Given Rates of Discharge in Clean
on Pipes for Each 1000 Feet of Length.

nsed from Ellis and Howland's Hydraulic Tables.)

	6-inch Pipe.		8-inch Pipe.		10-inch Pipe.		12-inch Pipe.		14-inch Pipe.	
head, feet.	Velocity in ft. per sec.	Friction-head, feet.	Velocity in ft. per sec.	Friction-head, feet.	Velocity in ft. per sec.	Friction-head, feet.	Velocity, in ft. per sec.	Friction-head, feet.	Velocity, in ft. per sec.	Friction-head, feet.
.59	.28	.11	.16	.04	.10	.02	.07	.01	....	.....
2 01	.57	.32	.32	.10	.20	.04	.14	.02	.10	.01
7.36	1.13	1.08	.64	.29	.41	.11	.28	.05	.21	.03
5.05	1.70	2.28	.96	.60	.61	.22	.43	.10	.31	.05
3.09	2.27	3.92	1.28	1.01	.82	.36	.57	.16	.42	.08
3.47	2.84	6.00	1.60	1.52	1.02	.54	.71	.24	.52	.12
2 20	3.40	8.52	1.91	2.13	1.23	.75	.85	.32	.63	.16
4.26	3.97	11.48	2.23	2.85	1.43	.99	.99	.43	.73	.21
9.68	4.54	14.89	2.55	3.68	1.63	1.27	1.13	.54	.83	.27
0.53	5.67	23.01	3.19	5.64	2.04	1.93	1.42	.81	1.04	.40
4 76	6.81	32.89	3.83	8.03	2.45	2.72	1.70	1.14	1.25	.55
2.36	7.94	44.54	4 47	10.83	2.86	3.66	1.98	1.52	1.46	.73
....	9.08	57.95	5.09	14.05	3.27	4.73	2.27	1.96	1.67	.94
....	10 21	73.12	5.74	17.68	3.68	5.93	2.55	2.45	1 88	1.17
....	11 35	90.05	6.38	21.74	4.08	7.28	2.84	3.00	2.08	1.43
....	13.61	129.20	7.66	31.10	4 90	10.38	3.40	4.26	2.50	2.02
....	15 88	175.38	8.94	42.13	5.72	14.02	3.97	5.74	2 91	2.72
....	18 15	228 62	10.21	54.84	6.53	18.22	4.54	7.44	3.33	3.51
....	20.42	288.90	11.47	69.22	7.35	22.96	5.11	9.36	3.75	4.41
....	22.69	356.22	12.77	85 27	8.17	28.25	5.67	11.50	4.17	5.41
....	....	....	15.96	132.70	10.21	43.87	7.09	17.82	5.21	8.35
....	....	......	.....	......	12.25	62.92	8.51	25.51	6.25	11.93
....	.....	.....	....	.....	.....	.....	....	....	8.34	21.00

h	18-inch Pipe.		20-inch Pipe.		24-inch Pipe.		30-inch Pipe.		36-inch Pipe.	
Friction-head, feet.	Velocity in ft. per sec.	Friction-head, feet.	Velocity in ft. per sec.	Friction-head, feet.	Velocity in ft. per sec.	Friction-head, feet.	Velocity in ft. per sec.	Friction-head, feet.	Velocity in ft. per sec.	Friction-head, feet.
.22	.63	.13	.51	.08	.35	.04	.23	.01	.16	.01
.76	1.26	.44	1.02	.27	.71	.12	.45	.04	.32	.02
1.68	1.89	.93	1.53	.56	1.06	.24	.68	.09	.47	.04
2.82	2.52	1.60	2.04	.96	1.42	.41	.91	.15	.63	.06
4.34	3.15	2 45	2.55	1.47	1.77	.62	1.13	.22	.79	.09
6.19	3.78	3.48	3.06	2.09	2.13	.87	1.36	.30	.95	.13
8.37	4.41	4.70	3.57	2.81	2.48	1.16	1.59	.40	1.10	.17
10.87	5.04	6.09	4.08	3.64	2.84	1.50	1.82	.52	1.26	.22
13.70	5.67	7.67	4.59	4.58	3.19	1.88	2.04	.64	1.42	.27
16.85	6.30	9.43	5.11	5.62	3.55	2.31	2.27	.78	1.58	.33
.....	7.57	13.49	6.13	8.03	4.26	3.28	2.72	1.11	1.89	.46
.....	.....	....	7.15	10.86	4.96	4.43	3.18	1.49	2 21	.62
.....	.....	.....	.....	.....	5.67	5.75	3.63	1.93	2.52	.80
.....	.....	.....	.....	.....	6.38	7 25	4.08	2.43	2 84	1.00
.....	.....	.....	.....	.....	.....	.....	4.54	2.98	3.15	1.23
.....	.....	.....	.....	.....	.....	.....	5.44	4.25	3.78	[illegible]
.....	.....	.....	.....	.....	.....	.....	6.36	5.75	[illegible]	[illegible]
.....	.....	.....	.....	.....	.....	.....	.....	.....	[illegible]	[illegible]
.....	.....	.....	.....	.....	.....	.....	.....	.....	[illegible]	[illegible]
.....	.....	.....	.....	.....	.....	.....	.....	.....	[illegible]	[illegible]

Effect of Bends and Curves in Pipes.—Weisbach's

bends: Loss of head in feet $= .131 + 1.847\left(\frac{r}{R}\right)^{\frac{7}{2}} \times \frac{v^2}{64.4} \times \frac{a}{180}$, in

= internal radius of pipe in feet, R = radius of curvature of axis
= velocity in feet per second, and a = the central angle, or angle
by the bend.

Hamilton Smith, Jr., in his work on Hydraulics, says: The expe data at hand are entirely insufficient to permit a satisfactory an this quite complicated subject; in fact, about the only experiments are those made by Bossut and Dubuat with small pipes.

Curves.—If the pipe has easy curves, say with radius not les diameters of the pipe, the flow will not be materially diminished, the tops of all curves are kept below the hydraulic grade-line and be made for escape of air from the tops of all curves. (Trautwine.

Hydraulic Grade-line.—In a straight tube of uniform throughout, running full and discharging freely into the air, the grade-line is a straight line drawn from the discharge end to a poi diately over the entry end of the pipe and at a depth below th equal to the entry and velocity heads. (Trautwine.)

In a pipe leading from a reservoir, no part of its length should the hydraulic grade-line.

Flow of Water in House-service Pipes.

Mr. E. Kuichling, C.E., furnished the following table to the Meter Co.:

Condition of Discharge.	Pressure in Main, pounds per square inch.	Discharge, or Quantity capable of being deliv Cubic Feet per Minute, from the Pipe, under the conditions specified in the first col							
		Nominal Diameters of Iron or Lead Service-Inches.							
		½	⅝	¾	1	1½	2	3	[illegible]
Through 35 feet of service-pipe, no back pressure.	30	1.10	1.92	3.01	6.13	16.58	33.34	88.16	[illegible]
	40	1.27	2.22	3.48	7.08	19.14	38.50	101.80	[illegible]
	50	1.42	2.48	3.89	7.92	21.40	43.04	113.82	[illegible]
	60	1.56	2.71	4.26	8.67	23.44	47.15	124.68	[illegible]
	75	1.74	3.08	4.77	9.70	26.21	52.71	139.39	[illegible]
	100	2.01	3.50	5.50	11.20	30.27	60.87	160.96	[illegible]
	130	2.29	3.99	6.28	12.77	34.51	69.40	183.52	[illegible]
Through 100 feet of service-pipe, no back pressure.	30	0.66	1.16	1.84	3.78	10.40	21.30	58.19	[illegible]
	40	0.77	1.34	2.12	4.36	12.01	24.59	67.19	[illegible]
	50	0.86	1.50	2.37	4.88	13.43	27.50	75.13	[illegible]
	60	0.94	1.65	2.60	5.34	14.71	30.12	82.30	[illegible]
	75	1.05	1.84	2.91	5.97	16.45	33.68	92.01	[illegible]
	100	1.22	2.13	3.36	6.90	18.99	38.89	106.24	[illegible]
	130	1.39	2.42	3.83	7.86	21.66	44.34	121.14	[illegible]
Through 100 feet of service-pipe and 15 feet vertical rise.	30	0.55	0.96	1.52	3.11	8.57	17.55	47.90	[illegible]
	40	0.66	1.15	1.81	3.72	10.24	20.95	57.20	[illegible]
	50	0.75	1.31	2.06	4.24	11.67	23.87	65.18	[illegible]
	60	0.83	1.45	2.29	4.70	12.94	26.48	72.28	[illegible]
	75	0.94	1.64	2.59	5.32	14.64	29.96	81.79	[illegible]
	100	1.10	1.92	3.02	6.21	17.10	35.00	95.55	[illegible]
	130	1.26	2.20	3.48	7.14	19.66	40.23	109.82	[illegible]
Through 100 feet of service-pipe, and 30 fe[illegible]	30	0.44	0.77	1.22	2.50	6.80	14.11	38.63	[illegible]
	40	0.55	0.97	1.53	3.15	8.68	17.79	48.68	[illegible]
	50	0.65	1.14	1.79	3.69	10.16	20.82	56.98	[illegible]
	60	0.73	1.28	2.02	4.15	11.45	23.47	64.22	[illegible]
	[illegible]	0.84	1.47	2.32	4.77	13.15	26.95	73.76	[illegible]
	[illegible]	[illegible]	1.74	2.75	5.65	15.58	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	2.02	3.19	6.55	18.07	[illegible]	[illegible]	[illegible]

is assumed that the pipe is straight and smooth inside; that he main and meter are disregarded; that the inlet from the nary character, sharp, not flaring or rounded, and that the diameter of pipe. The deliveries given will be increased if, etween the meter and the main is of larger diameter than the if the main is tapped, say for 1-inch pipe, but is enlarged 1¼ or 1½ inch; or, third, if pipe on the outlet is larger than t side of the meter. The exact details of the conditions given n practice; consequently the quantities of the table may be decreased, because the pipe is liable to be throttled at the l bends may interpose, or stop-cocks may be used, or the nay be increased.

Pipes.—A pipe is said to be air-bound when, in conse-ing entrapped at the high points of vertical curves in the not flow out of the pipe, although the supply is higher than e remedy is to provide cocks or valves at the high points, the air may be discharged. The valve may be made auto-s of a float.

ets. (Molesworth.)—H = head of water, h = height of jet, f jet, K = coefficient, varying with ratio of diameter of jet = KH.

300	600	1000	1500	1800	2800	3500	4500,
.96	.9	.85	.8	.7	.6	.5	.25

livered through Meters. (Thomson Meter Co.).—The ractice limits the velocity in water-pipes to 10 lineal feet per e this as a basis of delivery, and we find, for the several sizes y metered, the following approximate results:

ter of pipe in inches:

½	¾	1	1½	2	3	4	6

ered, in cubic feet per minute, due to said velocity:

.28	1.85	3.28	7.36	13.1	29.5	52.4	117.9

arged for Water in Different Cities (National

ium price for 1000 gallons in 163 places....	9.4 cents.
num " " " " " " "	28 "
cents to	100 "

FIRE-STREAMS.

arge from Nozzles at Different Pressures.

g, Am. Water-works Ass'n, 1892, *Eng'g News*, July 14, 1892.)

Pressure at Play-pipe, lbs.	Horizontal Projection of Streams, ft.	Gallons per minute.	Gallons per 24 hours.	Friction per 100 ft. Hose, lbs.	Friction per 100 ft. Hose, Net Head, ft.
46.5	59.5	203	292,298	10.75	24.77
59.0	67.0	230	331,200	13.00	31.10
79.0	76.6	267	384,500	17.70	40.78
130 0	88.0	311	447.900	22.50	54.14
44.5	61.3	249	358,520	15.50	35.71
55.5	69.5	281	404,700	19.40	44.70
72.0	78.5	324	466,600	25.40	58.52
103.0	89.0	376	541,500	33.80	77.88
43.0	66.0	306	440,613	22.75	52.42
53.5	72.4	343	493.900	28.40	65.43
68.5	81.0	388	558,800	35.90	82.71
93.0	92.0	460	662,500	57.75	86.98
41.5	77.0	368	530,149	32.50	[illegible]
51.5	74.4	410	590,500	40.00	[illegible]
65.5	*82.6*	*468*	674,000	51.40	[illegible]
88.0	*92.0*	*540*	777,700	72.00	[illegible]

Friction Losses in Hose.—In the above table the vol[illegible] water discharged per jet were for stated pressures at the play-pipe[illegible]
In providing for this pressure due allowance is to be made for [illegible] losses in each hose, according to the streams of greatest discharge w[illegible] to be used.

The loss of pressure or its equivalent loss of head (h) in the hose [illegible] found by the formula $h = v^2(4m)\frac{l}{2gd}$.

In this formula, as ordinarily used, for friction per 100 ft. of 2½-[illegible] there are the following constants: 2½ in. diameter of hose $d =$ .2[illegible] length of hose $l = 100$ ft., and $2g = 64.4$. The variables are: $v =$ ve[illegible] feet per second; $h =$ loss of head in feet per 100 ft. of hose; $m =$ [illegible] cient found by experiment; the velocity v is found from the gi[illegible] charges of the jets through the given diameter of hose.

Head and Pressure Losses by Friction in 100 Lengths of Rubber-lined Smooth 2½-in. Hose.

Discharge per minute, gallons.	Velocity per second, ft.	Coefficient, m.	Head Lost, ft.	Pressure Lost, lbs. per sq. in.	Gall[illegible] 24 h[illegible]
200	13.072	.00450	22.89	9.93	[illegible]
250	16.388	.00446	35.55	15.43	[illegible]
300	18.858	.00442	46.80	20.31	[illegible]
347	21.677	.00439	61.53	26.70	[illegible]
350	22.873	.00439	68.48	29.73	[illegible]
400	26.144	.00436	88.83	38.55	[illegible]
450	29.408	.00434	111.80	48.52	[illegible]
500	32.675	.00432	137.50	59.67	[illegible]
520	33.982	.00431	148.40	64.40	[illegible]

These frictions are for given volumes of flow in the hose and th[illegible] ties respectively due to those volumes, and are independent of [illegible] nozzle. The changes in nozzle do not affect the friction in the hose [illegible] is no change in velocity of flow, but a larger nozzle with equal pre[illegible] the nozzle augments the discharge and velocity of flow, and thus n[illegible] increases the friction loss in the hose.

Loss of Pressure (p) and Head (h) in Rubber Smooth 2½-in. Hose may be found approximately by the [illegible] $p = \frac{lq^2}{4150d^5}$ and $h = \frac{lq^2}{1801d^5}$, in which $p =$ pressure lost by fri[illegible] pounds per square inch; $l =$ length of hose in feet; $q =$ gallons [illegible] discharged per minute; $d =$ diam. of the hose in inches, 2½ in.; $h =$ [illegible] head in feet. The coefficient of d^5 would be decreased for rougher [illegible]

The loss of pressure and head for a 1⅛-in. stream with power to [illegible] height of 80 ft. is, in each 100 ft. of 2½-in. hose, approximately 20 [illegible] ft. net, or, say, including friction in the hydrant, ½ ft. loss of head [illegible] foot of hose.

If we change the nozzles to 1¼ or 1⅜ in. diameter, then for the sa[illegible] height of stream we increase the friction losses on the hose to [illegible] mately ⅝ ft. and 1 ft. head, respectively, for each foot-length of ho[illegible]

These computations show the great difficulty of maintaining [illegible] stream through large nozzles unless the hose is very short, especial[illegible] gravity or direct-pressure system.

This single 1⅛-in. stream requires approximately 56 lbs. pressure, [illegible] lent to 129 ft. head, at the play-pipe, and 45 to 50 ft. head for eac[illegible] length of smooth 2½-in. hose, so that for 100, 200, and 300 ft. of [illegible] must have available heads at the hydrant or fire-engine of 1[illegible]6, 1[illegible] ft., respectively. If we substitute 1¼-in. nozzles for same height of [illegible] we must have available heads at the hydrants or engine of 185, 2[illegible] ft., respectively, or we must increase the diameter of a portion a[illegible] the long hose and save friction-loss of head.

Rated Capacities of Steam Fire-engines, which is [illegible] *one third greater than their ordinary rate of work at fires, are* [illegible] *as follows:*

3d size,	550 gals. per min., or	792,000 gals. per 24 hours.	
2d "	700 " "	1,008,000 " "	
1st "	[illegible] " "	1,296,000 " "	
1 e[illegible]	[illegible] "	1,584,000	

s required at Nozzle and at Pump, with Quantity Pressure of Water Necessary to throw Water us Distances through Different-sized Nozzles— 2½-inch Rubber Hose and Smooth Nozzles.

xperiments of Ellis & Leshure, Fanning's "Water Supply.")

Size of Nozzles.	1 Inch.				1⅛ Inch.			
nozzle, lbs. per sq. in.......	40	60	80	100	40	60	80	100
at pump or hydrant with inch rubber hose...........	48	73	97	121	54	81	108	135
minute....................	155	189	219	245	196	240	277	310
distance thrown, feet.......	109	142	168	186	113	148	175	193
tance thrown, feet...........	79	108	131	148	81	112	137	157
Size of Nozzles.	**1¼ Inch.**				**1⅜ Inch.**			
nozzle, lbs. per sq. in.......	40	60	80	100	40	60	80	100
at pump or hydrant with ½-inch rubber hose..........	61	92	123	154	71	107	144	180
minute....................	242	297	342	383	293	358	413	462
distance thrown, feet........	118	156	186	207	124	166	200	224
stance thrown, feet...........	82	115	142	164	85	118	146	169

eater length of 2½-inch hose the increased friction can be ob-noting the differences between the above given "pressure at d "pressure at pump or hydrant with 100 feet of hose." For it requires at hydrant or pump eight pounds more pressure s at nozzle to overcome the friction when pumping through 100 -inch hose (using 1-inch nozzle, with 40-pound pressure at said n it requires 16-pounds pressure to overcome the friction in ough 200 feet of same size hose.

se of Flow due to Increase of Length of Hose. an's Experiments, Trans. A. S. C. E. 1889.)—If the static pres-bs. and the hydrant-pipes of such size that the pressure at the hy-lbs., the hose 2½ in. nominal diam., and the nozzle 1⅛ in. diam., of effective fire-stream obtainable and the quantity in gallons per l be:

	Linen Hose. Height, feet.	Linen Hose. Gals. per min.	Best Rubber-lined Hose. Height, feet.	Best Rubber-lined Hose. Gals. per min.
50 ft. of 2½-in. hose.......	73	261	81	282
50 " " " "	42	184	61	229
00 " " " "	27	146	46	192

) ft. of smoothest and best rubber-lined hose, if diameter be in., effective height of stream will be 39 ft. (177 gals.); if diameter rger, effective height of stream will be 46 ft. (192 gals.)

THE SIPHON.

on is a bent tube of unequal branches, open at both ends, and onvey a liquid from a higher to a lower level, over an intermedi-igher than either. Its parallel branches being in a vertical plane d into two bodies of liquid whose upper surfaces are at different fluid will stand at the same level both within and without each the tube when a vent or small opening is made at the bend. If withdrawn from the siphon through this vent, the water will rise nches by the atmospheric pressure without, and when the two nite and the *vent is closed, the liquid will flow from the* upper *s long as the end of the shorter* branch of the siphon is below the *the liquid in the reservoir.*

ter was free from air the height of the bend above tl *e as great as 33 feet.*

If A = area of cross-section of the tube in square feet, H = the diffe… in level between the two reservoirs in feet, D the density of the… pounds per cubic foot, then ADH measures the intensity of the force… causes the movement of the fluid, and $V = \sqrt{2gH} = 8.02\sqrt{H}$ is the theo… velocity, in feet per second, which is reduced by the loss of head… and friction, as in other cases of flow of liquids through pipes. In the… of the difference of level being greater than 33 feet, however, the… the water in the shorter leg is limited to that due to a height of 33 f… that due to the difference between the atmospheric pressure at the… and the vacuum at the bend.

Leicester Allen (*Am. Mach.*, Nov. 2, 1893) says: The supply of liquid… siphon must be greater than the flow which would take place from the… charge end of the pipe, provided the pipe were filled with the liquid… supply end stopped, and the discharge end opened when the discharge… is left free, unregulated, and unsubmerged.

To illustrate this principle, let us suppose the extreme case of a… having a calibre of 1 foot, in which the difference of level, or betwe… point of supply and discharge, is 4 inches. Let us further suppose… siphon to be at the sea-level, and its highest point above the level of… supply to be 27 feet. Also suppose the discharge end of this siphon… regulated, unsubmerged. It would be inoperative because the water… longer leg would not be held solid by the pressure of the atmosphere… it, and it would therefore break up and run out faster than it could… placed at the inflow end under an effective head of only 4 inches.

Long Siphons.—Prof. Joseph Torrey, in the *Amer. Mach.*… describes a long siphon which was a partial failure.

The length of the pipe was 1792 feet. The pipe was 3 inches diame… rose at one point 9 feet above the initial level. The final level was… below the initial level. No automatic air valve was provided. The… point in the siphon was about one third the total distance from the… nearest the pond. At this point a pump was placed, whose mission… fill the pipe when necessary. This siphon would flow for about two… and then cease, owing to accumulation of air in the pipe. When… operation it discharged 43½ gallons per minute. The theoretical… from such a sized pipe with the specified head is 55½ gallons per min…

Siphon on the Water-supply of Mount Vernon, N.… (*Eng'g News*, May 4, 1893.)—A 12-inch siphon, 925 feet long, with a max… lift of 22.12 feet and a 45° change in alignment, was put in use in 18… New York City Suburban Water Co., which supplies Mount Vernon, N…

At its summit the siphon crosses a supply main, which is tapped to… the siphon.

The air-chamber at the siphon is 12 inches by 16 feet long. A ½-inch… and cock at the top of the chamber provide an outlet for the collected…

It was found that the siphon with air-chamber as described would… until 125 cubic feet of air had gathered, and that this took place only… soon with a 14-foot lift as with the full lift of 22.12 feet. The siphon… operate about 12 hours without being recharged, but more water… gotten over by charging every six hours. It can be kept running 23 h… out of 24 with only one man in attendance. With the siphon as descr… above it is necessary to close the valves at each end of the siphon… recharge it.

It has been found by weir measurements that the discharge of the si… before air accumulates at the summit is practically the same as throu… straight pipe.

MEASUREMENT OF FLOWING WATER.

Piezometer.—If a vertical or oblique tube be inserted into a pipe… taining water under pressure, the water will rise in the former, and the… tical height to which it rises will be the head producing the pressure at… point where the tube is attached. Such a tube is called a piezometer… pressure measure. If the water in the piezometer falls below its… level it shows that the pressure in the main pipe has been reduced by… obstruction between the piezometer and the reservoir. If the water… *above its* proper level, it indicates that the pressure there has been… *creased* by an obstruction beyond the piezometer.

If we imagine a pipe full of water to be provided with a number of… zometers, then a line joining the tops of the columns of water is the… *the hydraulic* …-line.

Tube Gauge.—The Pitot tube is used for measuring the veloc-
iids in motion. It has been used with great success in measuring
of natural gas. (S. W. Robinson, Report Ohio Geol. Survey, 1890.)
Van Nostrand's Mag., vol. xxxv.) It is simply a tube so bent that
eg extends into the current of fluid flowing from a tube, with the
the entering orifice opposed at right angles to the direction of the
The pressure caused by the impact of the current is transmitted
the tube to a pressure-gauge of any kind, such as a column of
of mercury, or a Bourdon spring-gauge. From the pressure thus
and the known density and temperature of the flowing gas is ob-
e head corresponding to the pressure, and from this the velocity.
ification of the Pitot tube described by Prof. Robinson, there are
s inserted into the pipe conveying the gas, one of which has the
the orifice at right angles to the current, to receive the static pres-
the pressure due to impact; the other has the plane of its orifice
to the current, so as to receive the static pressure only. These
connected to the legs of a U tube partly filled with mercury, which
isters the difference in pressure in the two tubes, from which the
may be calculated. Comparative tests of Pitot tubes with gas-
or measurement of the flow of natural gas, have shown an agree-
hin 3%.

Venturi Meter, invented by Clemens Herschel, and described in
let issued by the Builders' Iron Foundry of Providence, R. I., is
rom Venturi, who first called attention, in 1796, to the relation be-
e velocities and pressures of fluids when flowing through converging
rging tubes.

ists of two parts—the tube, through which the water flows, and the
, which registers the quantity of water that passes through the

be takes the shape of two truncated cones joined in their smallest
s by a short throat-piece. At the up-stream end and at the throat
air-chambers, at which points the pressuees are taken.

tion of the tube is based on that property which causes the small
f a gently expanding frustum of a cone to receive, without material
t loss of head, as much water at the smallest diameter as is dis-
at the large end, and on that further property which causes the
of the water flowing through the throat to be less, by virtue of its
velocity, than the pressure at the up-stream end of the tube, each
being at the same time a function of the velocity at that point and
ydrostatic pressure which would obtain were the water motionless
he pipe.

corder is connected with the tube by pressure-pipes which lead to
the chambers surrounding the up-stream end and the throat of the
t may be placed in any convenient position within 1000 feet of the
t is operated by a weight and clockwork.

ifference of pressure or head at the entrance and at the throat of the
balanced in the recorder by the difference of level in two columns
ury in cylindrical receivers, one within the other. The inner carries
the position of which is indicative of the quantity of water flowing
the tube. By its rise and fall the float varies the time of contact
an integrating drum and the counters by which the successive
s are registered.

is no limit to the sizes of the meters nor the quantity of water that
measured. Meters with 24-inch, 36-inch, 48-inch, and even 20-foot
an be readily made.

surement by Venturi Tubes. (Trans. A. S. C. E., Nov., 1887,
., 1888.)—Mr. Herschel recommends the use of a Venturi tube, in-
n the force-main of the pumping engine, for determining the quantity
r discharged. Such a tube applied to a 24-inch main has a total
of about 20 feet. At a distance of 4 feet from the end nearest the
the inside diameter of the tube is contracted to a throat having a
r of about 8 inches. A pressure-gauge is attached to each of two
rs, the one surrounding and communicating with the entrance or
pe, the other *with the throat. According* to experiments made upon
s of this kind, one 4 in. in diameter at the throat and 12 in. at the en-
d the other about 36 in. in diameter at the throat and 9 fe...
the quantity of water which passes through the tube is ver...
tical discharge through an opening having an area equa...
t, and a velocity which is that due to the difference in he...

by the two gauges. Mr. Herschel states that the coefficient for widely-varying sizes of tubes and for a wide range of velocity th pipe, was found to be within two per cent, either way, of 98 words, the quantity of water flowing through the tube per sec pressed within two per cent by the formula $W = 0.98 \times A \times \sqrt{2gh}$, A is the area of the throat of the tube, h the head, in feet, co ing to the difference in the pressure of the water entering the tube found at the throat, and $g = 32.16$.

Measurement of Discharge of Pumping-engi Means of Nozzles. (Trans. A. S. M. E., xiii, 557).—The mea of water by computation from its discharge through orifices, or th nozzles of fire-hose, furnishes a means of determining the quantity delivered by a pumping-engine which can be applied without much John R. Freeman, Trans. A. S. C. E., Nov., 1889, describes a series ments covering a wide range of pressures and sizes, and the results that the coefficient of discharge for a smooth nozzle of ordinary was within one half of one per cent, either way, of 0.977; the dia the nozzle being accurately calipered, and the pressures being de by means of an accurate gauge attached to a suitable piezometer at of the play-pipe.

In order to use this method for determining the quantity of charged by a pumping-engine, it would be necessary to provide a box, to which the water would be conducted, and attach to the box nozzles as would be required to carry off the water. Accordin Freeman's estimate, four 1¼-inch nozzles, thus connected, with a of 80 lbs. per square inch, would discharge the full capacity of a t half-million engine. He also suggests the use of a portable appar a single opening for discharge, consisting essentially of a Siames so-called, the water being carried to it by three or more lines of fir

To insure reliability for these measurements, it is necessary that off valve in the force-main, or the several shut-off valves, should so that all the water discharged by the engine may pass through th

Flow through Rectangular Orifices. (Approximate. S

CUBIC FEET OF WATER DISCHARGED PER MINUTE THROUGH AN ORI INCH SQUARE, UNDER ANY HEAD OF WATER FROM 3 TO 72 INC

For any other orifice multiply by its area in square inches

Formula, $Q' = .624\sqrt{h''} \times a$. Q' = cu. ft. per min.; a = area in

Heads in inches.	Cubic Feet Discharged per min.	Heads in inches.	Cubic Feet Discharged per min.	Heads in inches.	Cubic Feet Discharged per min.	Heads in inches.	Cubic Feet Discharged per min.	Heads in inches.	Cubic Feet Discharged per min.	Heads in inches.	Cubic Feet Discharged per min.
3	1.12	13	2.20	23	2.90	33	3.47	43	3.95	53	4.39
4	1.27	14	2.28	24	2.97	34	3.52	44	4.00	54	4.42
5	1.40	15	2.36	25	3.03	35	3.57	45	4.05	55	4.46
6	1.52	16	2.43	26	3.08	36	3.62	46	4.09	56	4.52
7	1.64	17	2.51	27	3.14	37	3.67	47	4.12	57	4.55
8	1.75	18	2.58	28	3.20	38	3.72	48	4.18	58	4.58
9	1.84	19	2.64	29	3.25	39	3.77	49	4.21	59	4.63
10	1.94	20	2.71	30	3.31	40	3.81	50	4.27	60	4.65
11	2.03	21	2.78	31	3.36	41	3.86	51	4.30	61	4.72
12	2.12	22	2.84	32	3.41	42	3.91	52	4.34	62	4.74

Measurement of an Open Stream by Velocity and section.—Measure the depth of the water at from 6 to 12 poin the stream at equal distances between. Add all the depths in feet and divide by the number of measurements made; this will be the depth of the stream, which multiplied by its width will give its area section. Multiply this by the velocity of the stream in feet per mi the result will be the discharge in cubic feet per minute of the stre

The velocity [illegible] stream can be found by laying off 100 feet of *throwi*[illegible] the middle, noting the time taken to pass *100 ft.* [illegible] ber of times and take the average; then ce by the time gives the velocity at the surface. As the top of the ws faster than the bottom or sides—the average velocity being of the surface velocity at the middle—it is convenient to measure of 120 feet for the float and reckon it as 100.

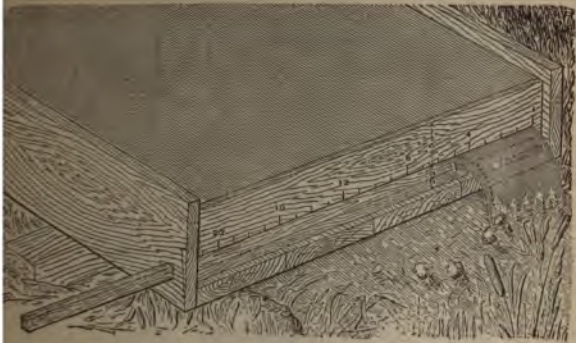

FIG. 130.

ers' Inch Measurements. (Pelton Water Wheel Co.)

Fig. 130, shows the form of measuring-box ordinarily used, and the table gives the discharge in cubic feet per minute of a miner's inch as measured under the various heads and different lengths and f apertures used in California.

Openings 2 Inches High.			Openings 4 Inches High.		
Head to Centre, 5 inches	Head to Centre, 6 inches.	Head to Centre, 7 inches.	Head to Centre, 5 inches.	Head to Centre, 6 inches.	Head to Centre, 7 inches.
Cu. ft.	Cu. ft.	Cu. ft.	Cu. ft.	Cu. ft.	Cu. ft.
1.348	1.473	1.589	1.320	1.450	1.570
1.355	1.480	1.596	1.336	1.470	1.595
1.359	1.484	1.600	1.344	1.481	1.608
1.361	1.485	1.602	1.349	1.487	1.615
1.363	1.487	1.604	1.352	1.491	1.620
1.364	1.488	1.604	1.354	1.494	1.623
1.365	1.489	1.605	1.356	1.496	1.626
1.365	1.489	1.606	1.357	1.498	1.628
1.365	1.490	1.606	1.359	1.499	1.630
1.366	1.490	1.607	1.359	1.500	1.631
1.366	1.490	1.607	1.360	1.501	1.632
1.366	1.490	1.607	1.361	1.502	1.633
1.367	1.491	1.607	1.361	1.503	1.634
1.367	1.491	1.608	1.362	1.503	1.635
1.367	1.492	1.608	1.363	1.505	1.637
1.368	1.493	1.609	1.364	1.507	1.639
1.368	1.493	1.609	1.365	1.508	1.640
1.368	1.493	1.609	1.365	1.508	1.641
1.368	1.493	1.609	1.366	1.509	[illegible]
1.369	1.493	1.610	1.366	1.509	[illegible]
1.369	1.494	1.610	1.366	1.509	[illegible]

—*The apertures from which the above measurements wer* were through material 1¼ inches thick, and the lower edge 2 inches the bottom of the measuring-box, thus giving full contraction.

Flow of Water Over Weirs. Weir Dam Measure (Pelton Water Wheel Co.)—Place a board or plank in the stream, as

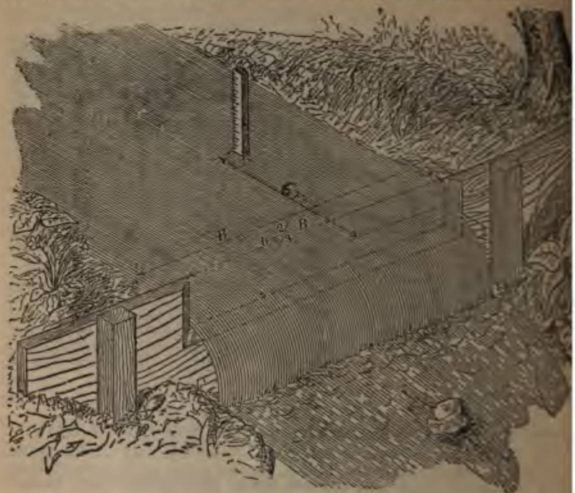

FIG. 131.

in the sketch, at some point where a pond will form above. The len the notch in the dam should be from two to four times its depth for quantities and longer for large quantities. The edges of the notch be bevelled toward the intake side, as shown. The overfall below the should not be less than twice its depth. [Francis says a fall below th equal to one-half the head is sufficient, but there must be a free ac air under the sheet.]

In the pond, about 6 ft. above the dam, drive a stake, and then obstr water until it rises precisely to the bottom of the notch and mark the at this level. Then complete the dam so as to cause all the water through the notch, and, after time for the water to settle, mark the again for this new level. If preferred the stake can be driven with precisely level with the bottom of the notch and the depth of the w measured with a rule after the water is flowing free, but the marks ar ferable in most cases. The stake can then be withdrawn; and the d between the marks is the theoretical depth of flow corresponding quantities in the table on the following page.

Francis's Formulæ for Weirs.

	As given by Francis.	As modif Smith
Weirs with both end contractions suppressed	$Q = 3.33lh^{\frac{3}{2}}$	$3.29\left(l+\right.$
Weirs with one end contraction suppressed..........	$Q = 3.33(l - .1h)h^{\frac{3}{2}}$	$3.29lh^{\frac{3}{2}}$
Weirs with full contraction.......	$Q = 3.33(l - .2h)h^{\frac{3}{2}}$	$3.29\left(l-\right.$

...reatest variation of the Francis formulæ from the values of c gi ...ounts to 3½%. The modified Francis formulæ, says Smith, w ...icient ... n great accuracy is not required, wit fro ... being not less than 3 h.

n cubic feet per second, l = length of weir in feet, h = effec- measured from the level of the crest to the level of still weir.

rge in cubic feet per *minute*, and l' and h' are taken in of the above formulæ reduces to $Q' = 0.4l'h'^{\frac{3}{2}}$. From this wing table is calculated. The values are sufficiently accu- computations of water-power for weirs without end con- for a weir the full width of the channel of approach, and also for weirs with end contraction when l = at least $10h$, xcess of the truth when $l = 4h$.

Weir Table.

EET OF WATER PER MINUTE THAT WILL FLOW OVER A WEIR INCH WIDE AND FROM 1/8 TO 20 7/8 INCHES DEEP.

other widths multiply by the width in inches.

	1/8 in.	1/4 in.	3/8 in.	1/2 in.	5/8 in.	3/4 in.	7/8 in.
	cu. ft.	cu. ft.	cu. ft.	cu. ft.	cu. ft.	cu. ft.	cu. ft.
	.01	.05	.09	.14	.19	.26	.32
	.47	.55	.64	.73	.82	.92	1.02
	1.23	1.35	1.46	1.58	1.70	1.82	1.95
	2.21	2.34	2.48	2.61	2.76	2.90	3.05
	3.35	3.50	3.66	3.81	3.97	4.14	4.30
	4.64	4.81	4.98	5.15	5.33	5.51	5.69
	6.06	6.25	6.44	6.62	6.82	7.01	7.21
	7.60	7.80	8.01	8.21	8.42	8.63	8.83
	9.26	9.47	9.69	9.91	10.13	10.35	10.57
	11.02	11.25	11.48	11.71	11.94	12.17	12.41
	12.88	13.12	13.36	13.60	13.85	14.09	14.34
	14.84	15.09	15.34	15.59	15.85	16.11	16.36
	16.88	17.15	17.41	17.67	17.94	18.21	18.47
	19.01	19.29	19.56	19.84	20.11	20.39	20.67
	21.23	21.51	21.80	22.08	22.37	22.65	22.94
	23.52	23.82	24.11	24.40	24.70	25.00	25.30
	25.90	26.20	26.50	26.80	27.11	27.42	27.72
	28.34	28.65	28.97	29.28	29.59	29.91	30.22
	30.86	31.18	31.50	31.82	32.15	32.47	32.80
	33.45	33.78	34.11	34.44	34.77	35.10	35.44
	36.11	36.45	36.78	37.12	37.46	37.80	38.15

curate computations, the coefficients of flow of Hamilton f Bazin should be used. In Smith's hydraulics will be found esults of experiments on orifices and weirs of various shapes different authorities, together with a discussion of their e. (See also Trautwine's Pocket Book.)

xperiments.—M. Bazin (*Annales des Ponts et Chaussées*, lated by Marichal and Trautwine, Proc. Engrs. Club of Phila., le an extensive series of experiments with a sharp-crested iteral contraction, the air being admitted freely behind the nd found values of m varying from 0.42 to 0.50, with varia- gth of the weir from 19 3/4 to 78 3/4 in., of the height of the crest m of the channel from 0.79 to 2.46 ft., and of the head from From these experiments he deduces the following formula:

$$Q = \left[0.425 + 0.21\left(\frac{H}{P+H}\right)^2\right] LH\sqrt{2gH},$$

ie height in feet of the crest of the weir above the bottom approach, L the length of the weir, H the head, both in fe harge in cu. ft. per sec. This formula, says M. Bazin, is where errors of 2% to 3% are admis followi sed from M. Bazin's *paper* :

VALUES OF THE COEFFICIENT m IN THE FORMULA $Q = mLH\sqrt{2}$
SHARP-CRESTED WEIR WITHOUT LATERAL CONTRACTION; THE
ADMITTED FREELY BEHIND THE FALLING SHEET.

Head, H.		Height of Crest of Weir Above Bed of Chann							
		Feet ...0.66 Inches 7.87	0.98 11.81	1.31 15.75	1.64 19.69	1.97 23.62	2.62 31.50	3.28 39.38	4.92 59.07
Ft.	In.	m	m	m	m	m	m	m	m
.164	1.97	0.458	0.453	0.451	0.450	0.449	0.449	0.449	0.448
.230	2.76	0.455	0.448	0.445	0.443	0.442	0.441	0.440	0.440
.295	3.54	0.457	0.447	0.442	0.440	0.438	0.436	0.436	0.435
.394	4.72	0.462	0.448	0.442	0.438	0.436	0.433	0.432	0.430
.525	6.30	0.471	0.453	0.444	0.438	0.435	0.431	0.429	0.427
.656	7.87	0.480	0.459	0.447	0.440	0.436	0.431	0.428	0.425
.787	9.45	0.488	0.465	0.452	0.444	0.438	0.432	0.428	0.424
.919	11.02	0.496	0.472	0.457	0.448	0.441	0.433	0.429	0.424
1.050	12.60	....	0.478	0.462	0.452	0.444	0.436	0.430	0.424
1.181	14.17	....	0.483	0.467	0.456	0.448	0.438	0.432	0.424
1.312	15.75	... .	0.489	0.472	0.459	0.451	0.440	0.433	0.424
1.444	17.32	....	0.494	0.476	0.463	0.454	0.442	0.435	0.425
1.575	18.90	... .	.....	0.480	0.467	0.457	0.444	0.436	0.425
1.706	20.47	... .	.....	0.483	0.470	0.460	0.446	0.438	0.426
1.837	22.05	....	.....	0.487	0.473	0.463	0.448	0.439	0.427
1.969	23.62	....	.....	0.490	0.476	0.466	0.451	0.441	0.427

A comparison of the results of this formula with those of e
says M. Bazin, justifies us in believing that, except in the unusu
very low weir (which should always be avoided), the precedin
give the coefficient m in all cases within 1%; provided, however,
rangements of the standard weir are exactly reproduced. It i
important that the admission of the air behind the falling sheet
assured. If this condition is not complied with, m may vary
wider limits. The type adopted gives the least possible vari
coefficient.

WATER-POWER.

Power of a Fall of Water—Efficiency.—The gro
a fall of water is the product of the weight of water discharged
time into the total head, i.e., the difference of vertical eleva
upper surface of the water at the points where the fall in ques
and ends. The term "head" used in connection with water-w
difference in height from the surface of the water in the whee
surface in the pen-stock when the wheel is running.

If Q = cubic feet of water discharged per second, D = weigh
foot of water = 62.36 lbs. at 60° F., H = total head in feet; then

DQH = gross power in foot-pounds per second,
and $DQH \div 550 = .1134QH$ = gross horse-power.

If Q' is taken in cubic feet per minute, H. P. $= \frac{Q'H \times 62.36}{33,000} =$

A water-wheel or motor of any kind cannot utilize the whole
H, since there are losses of head at both the entrance to and th
the wheel. There are also losses of energy due to friction of t
its passage through the wheel. The ratio of the power develo
wheel to the gross power of the fall is the efficiency of the whe

efficien... -power $= .00142Q'H = \frac{Q'H}{706}$.

...ad of water can be made use of in one or other of the following ways

By its weight, as in the water-balance and overshot-wheel.
By its pressure, as in turbines and in the hydraulic engine, hydraulic crane, etc.
By its impulse, as in the undershot-wheel, and in the Pelton wheel.
By a combination of the above.

...rse-power of a Running Stream.—The gross horse-power ...P. $= QH \times 62.36 \div 550 = .1134QH$, in which Q is the discharge in cubic ...er second actually impinging on the float or bucket, and $H =$ theoret... ...ead due to the velocity of the stream $= \frac{v^2}{2g} = \frac{v^2}{64.4}$, in which v is the ...ty in feet per second. If Q' be taken in cubic feet per minute, $= .00189Q'H$.

...s, if the floats of an undershot-wheel driven by a current alone be 5 $\times$ 1 foot, and the velocity of stream = 210 ft. per minute, or 3½ ft. per ...of which the theoretical head is .19 ft., $Q = 5$ sq. ft. $\times 210 = 1050$ cu. ft. ...inute; $H = .19$ ft.; H. P. $= 1050 \times .19 \times .00189 = .377$ H. P.

...e wheels would realize only about .4 of this power, on account of friction ...slip, or .151 H. P., or about .03 H.P. per square foot of float, which is ...lent to 33 sq. ft. of float per H. P.

...urrent Motors.—A current motor could only utilize the whole power ...running stream if it could take all the velocity out of the water, so that ...ould leave the floats or buckets with no velocity at all; or in other words, ...ould require the backing up of the whole volume of the stream until the ...al head was equivalent to the theoretical head due to the velocity of the ...eam. As but a small fraction of the velocity of the stream can be taken ...by a current motor, its efficiency is very small. Current motors may be ...to obtain small amounts of power from large streams, but for large ...ms they are not practicable.

Horse-power of Water Flowing in a Tube.—The head due to ...velocity is $\frac{v^2}{2g}$; the head due to the pressure is $\frac{f}{w}$; the head due to actual ...ht above the datum plane is h feet. The total head is the sum of these $= ... + h + \frac{f}{w}$, in feet, in which $v =$ velocity in feet per second, $f =$ pressure ...lbs. per sq. ft., $w =$ weight of 1 cu. ft. of water = 62.36 lbs. If $p =$ pres...re in lbs. per sq. in., $\frac{f}{w} = 2.309p$. In hydraulic transmission the velocity ...d the height above datum are usually small compared with the pressure-...ad. The work or energy of a given quantity of water under pressure = ...volume in cubic feet $\times$ its pressure in lbs. per sq. ft.; or if $Q =$ quantity ...cubic feet per second, and $p =$ pressure in lbs. per square inch, $W = ...pQ$, and the H. P. $= \frac{144pQ}{550} = .2618pQ$.

Maximum Efficiency of a Long Conduit.—A. L. Adams and ...G. Gemmel (*Eng'g News*, May 4, 1893), show by mathematical analysis that ...conditions for securing the maximum amount of power through a long ...duit of fixed diameter, without regard to the economy of water, is that ...draught from the pipe should be such that the frictional loss in the pipe ...be equal to one third of the entire static head.

Mill-Power.—A "mill-power" is a unit used to rate a water-power for ...purpose of renting it. The value of the unit is different in different ...alities. The following are examples (from Emerson):

Holyoke, Mass.—Each mill-power at the respective falls is declared to be ...right during 16 hours in a day to draw 38 cu. ft. of water per second at ...upper fall when the head there is 20 feet, or a quantity proportionate to ...height at the falls. This is equal to 86.2 horse-power as a maximum.

Lowell, Mass.—The right to draw during 15 hours in the day so much water ...shall give a power equal to 25 cu. ft. a second at the great fall, when the ...there is 30 feet. Equal to 85 H. P. maximum.

Lawrence, Mass.—The right to draw during 16 hours in a day so much ...ter as shall give a horse-power equal to 30 cu. ft. per second when the ...ad is 25 feet. Equal to 85 H. P. maximum.

Minneapolis, Minn.—30 cu. ft. of water per second with head of ...
...al to 74.8 H. P.

Manchester, N. H.—Divide 725 by the number of feet of fall min...

a_1 = the arc subtending one bucket at entrance. (In practice a_1 is la
than a.)

$a_2 = gh$, the arc subtending one bucket at exit,

$K = bf$, normal section of passage, it being assumed that the pass
and buckets are very narrow,

$k_1 = bd$, initial normal section of bucket,

$k_2 = gi$, terminal normal section,

ωr_1 = velocity of initial rim,

ωr_2 = velocity of terminal rim,

$\theta = HFI$, angle between the terminal rim and actual direction of
water at exit,

Y = depth of K, y_1 of a_1, and y_2 of K_2, then

$K = Ya \sin \alpha$; $K_1 = y_1 a_1 \sin \gamma_1$; $K_2 = y_2 a_2 \sin \gamma_2$.

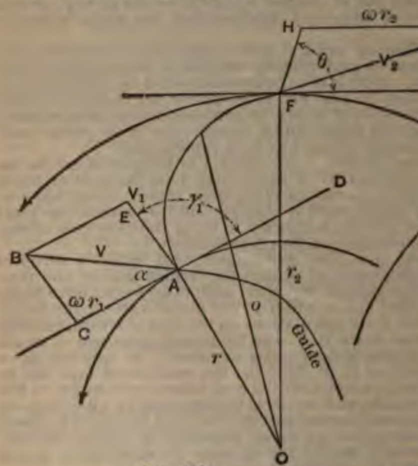

FIG. 182.

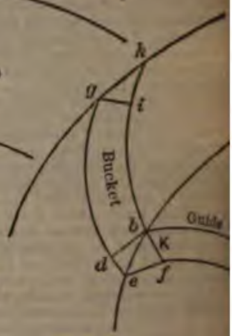

FIG. 183.

Three simple systems are recognized, $r_1 < r_2$, called outward flow; r_1 r_2
called inward flow; $r_1 = r_2$, called parallel flow. The first and second
be combined with the third, making a mixed system.

Value of γ_2 *(the quitting angle)*.—The efficiency is increased as γ_2
creases, and is greatest for $\gamma_2 = 0$. Hence, theoretically, the terminal
ment of the bucket should be tangent to the quitting rim for best effi
This, however, for the discharge of a finite quantity of water, w
require an infinite depth of bucket. In practice, therefore, this angle
have a finite value. The larger the diameter of the terminal rim the
may be this angle for a given depth of wheel and given quantity of w
discharged. In practice γ_2 is from 10° to 20°.

In a wheel in which all the elements except γ_2 are fixed, the veloc
the wheel for best effect must increase as the quitting angle of the
decreases.

Values of $\alpha + \gamma_1$ must be less than 180°, but the best relation can
determined by analysis. However, since the water should be deflected
its course as much as possible from its entering to its leaving the wheel
angle α for this reason should be as small as practicable.

In practice, α cannot be zero, and is made from 20° to 30°.

The value $r_1 = 1.4 r_2$ makes the width of the crown for internal flow
the same as for $r_1 = r_2\sqrt{\frac{1}{2}}$ for outward flow, being approximately 0.3 of
external radius.

Values of a_1 *and* a_2.—The frictional resistances depend upon the
tion of the ... smoothness of the surfaces, sharpness of the w

f the curved parts, and also upon the speed it is run. These ot be definitely assigned beforehand, but Weisbach gives for ons $\mu_1 = \mu_2 = 0.05$ to 0.10.
not necessarily equal, and μ_1 may be from 0.05 to 0.075, and μ_2 0.10 or even larger.
γ_1 must be less than $180° - a$.
he safe side, γ_1 may be 20 or 30 degrees less than $180°-2a$, giving

$$\gamma_1 = 180° - 2a - 25 \quad \text{(say)} \quad = 155 - 2a.$$

$= 30°$, $\gamma_1 = 95°$. Some designers make γ_1 90°; others more, and less, than that amount. Weisbach suggests that it be less, so cket will be shorter and friction less. This reasoning appears to for the inflow wheel, but not for the outflow wheel. In the Tre- nes, described in the Lowell Hydraulic Experiments, this angle angle a 20°, and γ_2 10°, which proportions insured a positive the wheel. Fourneyron made $\gamma_1 = 90°$, and a from 30° to 33°, s made the initial pressure in the wheel near zero.
Bucket.—The form of the bucket cannot be determined analytic- the initial and terminal directions and the volume of the water ough the wheel, the area of the normal sections may be found. al section of the buckets will be:

$$K = \frac{Q}{V}; \quad k_1 = \frac{Q}{v_1}; \quad k_2 = \frac{Q}{v_2}.$$

hs of those sections will be:

$$Y = \frac{K}{a \sin a}; \quad y_1 = \frac{k_1}{a_1 \sin \gamma_1}; \quad y_2 = \frac{k_2}{a_2 \sin \gamma_2}.$$

ges of curvature and section must be gradual, and the general ar, so that eddies and whirls shall not be formed. For the same wheel must be run with the correct velocity to secure the best practice the buckets are made of two or three arcs of circles, angential.
ue of ω.—So far as analysis indicates, the wheel may run at any t in order that the stream shall flow smoothly from the supply nto the bucket, the velocity V should be properly regulated.
$\mu_2 = 0.10$, $r_2 \div r_1 = 1.40$, $a = 25°$, $\gamma_1 = 90°$, $\gamma_2 = 12°$, the velocity of rim for outward flow will be for maximum efficiency 0.614 of the ue to the head, or $\omega r_1 = 0.614 \sqrt{2gH}$.
city due to the head would be $\sqrt{2gH} = 1.414 \sqrt{gH}$.
nflow wheel for the case in which $r_1^2 = 2r_2^2$, and the other dimen iven above, $\omega r_1 = 0.682 \sqrt{2gH}$.
hest efficiency of the Tremont turbine, found experimentally, was d the corresponding velocity, 0.62645 of that due to the head, and ocities above and below this value the efficiency was less.
Tremont wheel $a = 20°$ instead of 25°, and $\gamma_2 = 10°$ instead of 12°. uld make the theoretical efficiency and velocity of the wheel some- ater. Experiment showed that the velocity might be considerably smaller than this amount without much diminution of the efficiency. found that if the velocity of the initial (or interior) rim was not less nor more than 75% of that due to the fall, the efficiency was 75% or his wheel was allowed to run freely without any brake except its tion, and the velocity of the initial rim was observed to be H, half of which is $0.6675 \sqrt{2gH}$, which is not far from the velocity aximum effect; that is to say, when the gate is fully raised the coeffi- ffect is a maximum when the wheel is moving with about half its m velocity.
r of Buckets.—Successful wheels have been made in which the dis- tween the buckets was as small as 0.75 of an inch, and others as 2.75 inches. Turbines at the Centennial Exposition had buckets inches to 9 inches from centre to centre. If too large they will not operly. Neither should they be too deep. Horizontal partitions times introduced. These secure more efficient wor[illegible] only partly opened. The form and number of [illegible] urposes are chiefly the result of experience.

Ratio of Radii.—Theory does not limit the dimensions of the wheel. In practice,

for outward flow, $r_2 \div r_1$ is from 1.25 to 1.50;
for inward flow, $r_2 \div r_1$ is from 0.66 to 0.80.

It appears that the inflow-wheel has a higher efficiency than the outward-flow wheel. The inflow-wheel also runs somewhat slower for best effect. The centrifugal force in the outward-flow wheel tends to force the water outward faster than it would otherwise flow; while in the inward flow wheel it has the contrary effect, acting as it does in opposition to the velocity in the buckets.

It also appears that the efficiency of the outward-flow wheel increases slightly as the width of the crown is less and the velocity for maximum efficiency is slower; while for the inflow-wheel the efficiency slightly increases for increased width of crown, and the velocity of the outer rim at the same time also increases.

Efficiency.—The exact value of the efficiency for a particular wheel must be found by experiment.

It seems hardly possible for the effective efficiency to equal, much less exceed, 86%, and all claims of 90 or more per cent for these motors should be discarded as improbable. A turbine yielding from 75% to 80% is extremely good. Experiments with higher efficiencies have been reported.

The celebrated Tremont turbine gave 79¼% without the "diffuser," which might have added some 2%. A Jonval turbine (parallel flow) was reported as yielding 0.75 to 0.90, but Morin suggested corrections reducing it to 0.63 to 0.71. Weisbach gives the results of many experiments, in which the efficiency ranged from 50% to 84%. Numerous experiments give $E = 0.60$ to 0.65. The efficiency, considering only the energy imparted to the wheel, will exceed by several per cent the efficiency of the wheel, for the latter will include the friction of the support and leakage at the joint between the sluice and wheel, which are not included in the former; also as a plant the resistances and losses in the supply-chamber are to be still further deducted.

The Crowns.—The crowns may be plane annular disks, or conical, or curved. If the partitions forming the buckets be so thin that they may be discarded, the law of radial flow will be determined by the form of the crowns. If the crowns be plane, the radial flow (or radial component) will diminish, for the outward flow-wheel, as the distance from the axis increases—the buckets being full—for the angular space will be greater.

Prof. Wood deduces from the formulæ in his paper the tables on page 595.

It appears from these tables: 1. That the terminal angle, α, has frequently been made too large in practice for the best efficiency.

2. That the terminal angle, α, of the guide should be for the inflow less than 10° for the wheels here considered, but when the initial angle of the bucket is 90°, and the terminal angle of the guide is 5° 28′, the gain of efficiency is not 2% greater than when the latter is 25°.

3. That the initial angle of the bucket should exceed 90° for best effect for outflow-wheels.

4. That with the initial angle between 60° and 120° for best effect on inflow wheels the efficiency varies scarcely 1%.

5. In the outflow-wheel, column (9) shows that for the outflow for best effect the direction of the quitting water in reference to the earth should be nearly radial (from 76° to 97°), but for the inflow wheel the water is thrown forward in quitting. This shows that the velocity of the rim should somewhat exceed the relative final velocity backward in the bucket, as shown in columns (4) and (5).

6. In these tables the velocities given are in terms of $\sqrt{2gh}$, and the coefficients of this expression will be the part of the head which would produce that velocity if the water issued freely. There is only one case, column (5), where the coefficient exceeds unity, and the excess is so small it may be discarded; and it may be said that in a properly proportioned turbine with the conditions here given none of the velocities will equal that due to the head in the supply-chamber when running at best effect.

7. The inflow turbine presents the best conditions for construction for producing a given effect, the only apparent disadvantage being an increased first cost due to an increased depth, or an increased diameter for producing a given amount of work. The larger efficiency should, however, more than neutralize the increased first cost.

$r_1 = r_2\sqrt{\tfrac{1}{2}}$. $\mu_1 = \mu_2 = 0.10$. $\gamma_2 = 12°$. Parallel Crowns. $k_1v_1 = k_2v_2 = KV = Q = 1$.

Initial Angle. γ_1	Efficiency. E	Velocity Outer Rim. $r_2\omega'$	Velocity Inner Rim. $r_1\omega' = \sqrt{\tfrac{1}{2}}\,r_2\omega'$	Relative Velocity of Exit. v_2	Relative Velocity of Entrance. v_1	Velocity of Exit from supply-Chamber. V	Terminal Angle of Guide. α	Direction of quitting Water. θ	Head Equivalent of Energy in quitting Water. $\frac{w^2}{2g}$	$k_2\sqrt{gH}$
1	2	3	4	5	6	7	8	9	10	11
60°	0.804	$0.972\sqrt{2gH}$	$0.687\sqrt{2gH}$	$1.048\sqrt{2gH}$	$0.356\sqrt{2gH}$	$0.595\sqrt{2gH}$	31° 17′	76°	$0.051H$	0.67
90°	0.828	$0.874\sqrt{2gH}$	$0.619\sqrt{2gH}$	$0.931\sqrt{2gH}$	$0.274\sqrt{2gH}$	$0.676\sqrt{2gH}$	23° 56′	79°	$0.039H$	0.76
120°	0.839	$0.798\sqrt{2gH}$	$0.565\sqrt{2gH}$	$0.843\sqrt{2gH}$	$0.286\sqrt{2gH}$	$0.749\sqrt{2gH}$	19° 5′	82°	$0.031H$	0.84
150°	0.921	$0.709\sqrt{2gH}$	$0.501\sqrt{2gH}$	$0.707\sqrt{2gH}$	$0.416\sqrt{2gH}$	$0.886\sqrt{2gH}$	13° 31′	97°	$0.022H$	1.00

Inward-flow Turbine.

$r_1 = \sqrt{2}r_2$. $\mu_1 = \mu_2 = 0.10$. $\gamma_2 = 12°$. Parallel Crowns. $k_1v_1 = k_2v_2 = KV = Q = 1$.

γ_1	E.	Velocity Outer Rim. $r_1\omega'$	Velocity Inner Rim. $r_2\omega'$	v_2	v_1	V	α	θ	$\frac{w^2}{2g}$	$k_2\sqrt{gH}$
60°	0.920	$0.709\sqrt{2gH}$	$0.501\sqrt{2gH}$	$0.476\sqrt{2gH}$	$0.089\sqrt{2gH}$	$0.672\sqrt{2gH}$	7° 0′	110°	$0.010H$	1.48
90°	0.920	$0.688\sqrt{2gH}$	$0.487\sqrt{2gH}$	$0.470\sqrt{2gH}$	$0.069\sqrt{2gH}$	$0.691\sqrt{2gH}$	5° 28′	106°	$0.010H$	1.50
120°	0.919	$0.668\sqrt{2gH}$	$0.473\sqrt{2gH}$	$0.456\sqrt{2gH}$	$0.077\sqrt{2gH}$	$0.709\sqrt{2gH}$	4° 46′	105°	$0.010H$	1.55
150°	0.918	$0.634\sqrt{2gH}$	$0.448\sqrt{2gH}$	$0.429\sqrt{2gH}$	$0.126\sqrt{2gH}$	$0.743\sqrt{2gH}$	3° 08′	107°	$0.009H$	1.65

1876. (From a paper by R. H. Thurston on The Systen
Turbine Wheels in the United States, Trans. A. S. M. E., v
the judges at the International Exhibition conducted a se
turbines. Many of the wheels offered for tests were foun
less defective in fitting and workmanship. The following is
the results of all turbines entered which gave an efficien
Seven other wheels were tested, giving results between 65% r

Maker's Name, or Name the Wheel is Known By.	Per Cent at Full Gate or Discharge.	Per Cent at about 9/10 of Full Discharge.	Per Cent at about ⅞ of Full Discharge.	Per Cent at about ¾ of Full Discharge.	Per Cent at about
Risdon	87.68	......	86.20	82.41	...
National	83.79	......	... ..	70.79	...
Geyelin (single)	83.30	......	......	......	...
Thos. Tait	82.13	.. ..	......	70.40	66
Goldie & McCullough	81.21	... ..	71.01	55.90	...
Rodney Hunt Mach. Co.	78.70	71.66	......	68.60	51
Tyler Wheel	79.59	......	81.24	79.92	67
Geyelin (duplex)	77.57	......	......	......	...
Knowlton & Dolan	77.43	74.25	......	......	62
E. T. Cope & Sons	76.94	......	69.92	......	...
Barber & Harris	76.16	73.33	......	......	70
York Manufacturing Co.	75.70	......	67.08	67.57	62
W. F. Mosser & Co.	75.15	74.89	71.90	70.52	...

The limits of error of the tests, says Prof. Thurston, were
they are undoubtedly considerable as compared with the lat
the permanent flume at Holyoke—possibly as much as 4% or
Experiments with "draught-tubes," or "suction-tubes
actually "diffusers" in their effect, so far as Prof. Thursto
them, indicate the loss by friction which should be antic
cases, this loss decreasing as the tube increased in size, a

ith maximum tightness and transmitting power. A "quarter
about 10% as a maximum, and a "quarter twist" about 5%.

ns of Turbines.—For dimensions, power, etc., of stand-
turbines consult the catalogues of different manufacturers.
different makers vary greatly in their proportions for any

n Water-wheel.—Mr. Ross E. Browne (*Eng'g News*, Feb.
outlines the principles upon which this water-wheel is

of a water-wheel, operated by a jet of water escaping from
convert the energy of the jet, due to its velocity, into useful
er to utilize this energy fully the wheel-bucket, after catching
bring it to rest before discharging it, without inducing turbu-
tion of the particles.
be fully effected, and unavoidable difficulties necessitate the
tion of the energy. The principal losses occur as follows:
or angular diversion of the jet in entering, or in its course
ucket, causing impact, or the conversion of a portion of the
at instead of useful work. Second, in the so-called frictional
red to the motion of the water by the wetted surfaces of the
ing also the conversion of a portion of the energy into heat
ful work. Third, in the velocity of the water, as it leaves the
enting energy which has not been converted into work.
eking a high efficiency: 1. The bucket-surface at the entrance
proximately parallel to the relative course of the jet, and
ould be curved in such
o avoid sharp angular de-
stream. If, for example,
surface at an angle and
flected, a portion of the
d, the smoothness of the
turbed, and there results
loss by impact and other-
trance and deflection in
cket are such as to avoid
the main. (See Fig. 136.)

FIG. 134.

FIG. 135.

ber of buckets should be small, and the path of the jet in the
in other words, the total wetted surface should be small, as
ction will be proportional to this.
large end of the bucket should be as nearly tangential to the
ery as compatible with the clearance of the bucket which
great differences of velocity in the parts of the escaping water
ided. In order to bring the water to rest at the discharge end
, it is shown, mathematically, that the velocity of the bucket
half the velocity of the jet.
ich as shown in Fig. 135, will cause the heaping of more or less

dead or turbulent water at the point indicated by dark
shading. This dead water is subsequently thrown from
the wheel with considerable velocity, and represents a
large loss of energy. The introduction of the wedge in
the Pelton bucket (see Fig. 134) is an efficient means of
avoiding this loss.

A wheel of the form of the Pelton conforms closely in
construction to each of these requirements.

In a test made by the proprietors of the Idaho mine,
near Grass Valley, Cal., the dimensions and results were
as follows: Main supply-pipe, 22 in. diameter, 6900 ft.
ead of 386½ feet above centre of nozzle. The loss by friction
as 1.8 ft., reducing the effective head to 384.7 ft. The Pelton
n the test was 6 ft. in diameter and the nozzle was 1.89 in.
he work done was measured by a Prony brake, and the mean
wed a useful effect of 87.3%.
wheel is also used as a motor for small powers. A test by
f a 12-inch wheel, with a ⅝-inch nozzle, under 100 lbs. pressure,
e-power. The theoretical discharge was .0935 cu... ...per
the theoretical horse-power 2.45; the efficiency
ther *styles of water-motor* tested at the same t
55 per cent.

Pelton Water-wheel Tables. (Abridged.)

The smaller figures under those denoting the various heads [illegible] spouting velocity of the water in feet per minute. The cubic-feet [illegible] ment is also based on the flow per minute.

Head in ft.	Size of Wheels.	6 in. No. 1	12 in. No. 2	18 in. No. 3	18 in. No. 4	24 in. No. 5	3 ft.	4 ft.	5 ft.
20	Horse-power.	.05	.12	.20	.37	.66	1.50	2.64	[illegible]
	Cubic feet....	1.67	3.91	6.62	11.72	20.83	46.93	83.32	[illegible]
2151.97	Revolutions..	684	342	228	228	171	114	85	[illegible]
30	Horse-power.	.10	.23	.38	.69	1.22	2.76	4.88	[illegible]
	Cubic feet....	2.05	4.79	8.11	14.36	25.51	57.44	102.04	[illegible]
2635.62	Revolutions..	837	418	279	279	209	139	104	[illegible]
40	Horse-power.	.15	.35	.59	1.06	1.89	4.24	7.58	[illegible]
	Cubic feet....	2.37	5.53	9.37	16.59	29.46	66.36	107.84	[illegible]
3043.39	Revolutions..	969	484	323	323	242	161	121	[illegible]
50	Horse-power.	.21	.49	.84	1.49	2.65	5.98	10.60	[illegible]
	Cubic feet....	2.64	6.18	10.47	18.54	32.93	74.17	131.72	[illegible]
3402.61	Revolutions..	1083	541	361	361	270	180	135	[illegible]
60	Horse-power.	.28	.65	1.10	1.96	3.48	7.84	13.94	21.77
	Cubic feet....	2.90	6.77	11.47	20.31	36.08	81.25	144.32	225.80
3727.37	Revolutions..	1185	592	395	395	296	197	148	118
70	Horse-power.	.35	.82	1.39	2.47	4.39	9.88	17.58	27.51
	Cubic feet....	3.13	7.31	12.39	21.94	38.97	87.76	155.88	243.89
4026.00	Revolutions..	1281	640	427	427	320	213	160	[illegible]
80	Horse-power.	.43	1.00	1.70	3.01	5.36	12.04	21.44	33.54
	Cubic feet....	3.35	7.82	13.25	23.46	41.66	93.84	166.64	260.73
4303.99	Revolutions..	1368	684	456	456	342	228	171	137
90	Horse-power.	.51	1.20	2.03	3.60	6.39	14.40	25.59	40.04
	Cubic feet....	3.55	8.29	14.05	24.88	44.19	99.52	176.75	[illegible]
4565.04	Revolutions..	1452	726	484	484	363	242	181	[illegible]
100	Horse-power.	.60	1.40	2.32	4.21	7.49	16.84	29.93	46.85
	Cubic feet....	3.74	8.74	14.81	26.22	46.58	104.88	186.32	291.51
4812.00	Revolutions..	1530	765	510	510	382	255	191	[illegible]
120	Horse-power.	.79	1.84	3.12	5.54	9.85	22.18	39.41	[illegible]
	Cubic feet....	4.10	9.57	16.21	28.72	51.02	114.91	204.10	[illegible]
5271.30	Revolutions..	1677	838	559	559	419	279	209	[illegible]
140	Horse-power.	.99	2.33	3.94	6.99	12.41	27.96	49.64	77.71
	Cubic feet....	4.43	10.34	17.53	31.03	55.11	124.12	220.44	[illegible]
5693.65	Revolutions..	1812	906	604	604	453	302	[illegible]	181
160	Horse-power.	1.22	2.84	4.82	8.54	15.17	34.16	60.68	94.94
	Cubic feet....	4.73	11.05	18.74	33.17	58.92	132.68	235.68	[illegible]
6086.74	Revolutions..	1938	969	646	646	484	323	242	[illegible]
180	Horse-power.	1.45	3.39	5.75	10.19	18.10	40.77	72.41	[illegible]
	Cubic feet....	5.02	11.72	19.87	35.18	62.49	140.74	249.97	[illegible]
6455.97	Revolutions..	2049	1024	683	683	513	342	256	[illegible]
200	Horse-power.	1.70	3.97	6.74	11.93	21.20	47.75	84.81	132.79
	Cubic feet....	5.29	12.36	20.94	37.08	65.87	148.35	263.49	[illegible]
6805.17	Revolutions..	2160	1080	720	720	540	360	270	216
250	Horse-power.	2.38	5.56	9.42	16.68	29.63	66.74	118.54	185.47
	Cubic feet....	5.92	13.82	23.42	41.46	73.64	165.86	[illegible]	[illegible]
[illegible]	Revolutions..	2418	1209	806	806	605	403	302	[illegible]

Pelton Water-wheel Tables.—*Continued.*

	Size of Wheels.	6 in. No. 1	12 in. No. 2	18 in. No. 3	18 in. No. 4	24 in. No. 5	3 ft.	4 ft.	5 ft.	6 ft.
	Horse-pow'r	3.13	7.31	12.38	21.93	38.95	87.73	155.83	243.82	350.94
	Cubic feet...	6.48	15.13	25.66	45.42	80.67	181.69	322.71	504.91	726.76
2	Revolutions	2652	1326	884	884	663	442	331	265	221
	Horse-pow'r	3.94	9.21	15.61	27.64	49.09	110.56	196.38	307.25	442.27
	Cubic feet...	7.00	16.35	27.71	49.06	87.14	196.25	348.57	545.36	785.00
3	Revolutions	2865	1432	955	955	716	477	358	285	238
	Horse-pow'r	4.82	11.25	19.0	33.77	59.98	135.08	239.94	375.40	540.35
	Cubic feet...	7.49	17.48	29.63	52.45	93.16	209.80	372.64	583.02	839.20
00	Revolutions	3063	1531	1021	1021	765	510	382	306	255
	Horse-pow'r	5.75	13.43	22.76	40.29	71.57	161.19	286.31	447.95	644.78
	Cubic feet...	7.94	18.54	31.42	55.63	98.81	222.52	395.24	618.38	890.11
79	Revolutions	3249	1624	1083	1083	812	541	406	324	270
	Horse-pow'r	6.74	15.73	26.66	47.20	83.83	188.80	335.34	524.66	755.20
	Cubic feet...	8.37	19.54	33.12	58.64	104.15	234.56	416.62	651.83	938.25
96	Revolutions	3426	1713	1142	1142	856	571	428	342	285
	Horse-pow'r	....	....	....	62.04	110.19	248.16	440.77	689.63	992.65
	Cubic feet...	....	....	....	64.24	114.09	256.95	456.38	714.05	1027.80
94	Revolutions	....	....	....	1251	938	625	469	375	312
	Horse-pow'r	....	....	....	69.95	124.25	279.82	497.01	777.62	1119.29
	Cubic feet...	....	....	....	66.86	118.75	267.44	475.02	743.21	1069.77
.24	Revolutions	....	....	....	1302	976	651	488	390	325
	Horse-pow'r	....	....	....	78.18	138.86	312.73	555.46	869.06	1250.92
	Cubic feet...	....	....	....	69.38	123.23	277.54	492.95	771.26	1110.16
.34	Revolutions	....	....	....	1351	1013	675	506	405	337
	Horse-pow'r	....	....	....	86.70	154.00	346.83	616.03	963.82	1387.34
	Cubic feet...	....	....	....	71.82	127.56	287.28	510.25	798.33	1149.13
.19	Revolutions	....	....	....	1399	1049	699	524	419	349
	Horse-pow'r	....	....	....	95.52	169.66	382.09	678.66	1061.81	1528.36
	Cubic feet...	....	....	....	74.17	131.74	296.70	526.99	824.51	1186.81
.40	Revolutions	....	....	....	1444	1083	722	542	433	361
	Horse-pow'r	....	....	....	113.98	202.45	455.94	809.82	1267.02	1823.76
	Cubic feet...	....	....	....	78.67	139.74	314.70	558.96	874.53	1258.81
.00	Revolutions	....	....	....	1532	1149	766	574	459	383
00	Horse-pow'r	....	....	....	133.50	237.12	534.01	948.48	1483.97	2136.04
	Cubic feet...	....	....	....	82.93	147.30	331.72	589.19	921.83	1326.91
.89	Revolutions	....	....	....	1615	1210	807	605	484	403

THE POWER OF OCEAN WAVES.

bert W. Stahl, U. S. N. (Trans. A. S. M. E., xiii. 438), gives the following ıulæ and table, based upon a theoretical discussion of wave motion:
e total energy of one whole wave-length of a wave H feet high, L feet , and one foot in breadth, the length being the distance between succes- crests, and the height the vertical distance between the crest and the gh, is $E = 8LH^2\left(1 - 4.935\,\dfrac{H^2}{L^2}\right)$ foot-pounds.

ıe time required for each wave to travel through a distance equal to its length is $P = \sqrt{\dfrac{L}{5.123}}$ *seconds, and the number of waves* [illegible] given point in one minute is $N = \frac{60}{P} = 60\sqrt{\frac{5.123}{L}}$. Hence the total energy of an indefinite series of such waves, expressed in horse-power per foot of breadth, is

$$\frac{E \times N}{33000} = .0329H^2L\left(1 - 4.935\frac{H^2}{L^2}\right).$$

By substituting various values for $H \div L$, within the limits of such values actually occurring in nature, we obtain the following table of

TOTAL ENERGY OF DEEP-SEA WAVES IN TERMS OF HORSE-POWER PER FOOT OF BREADTH.

Ratio of Length of Waves to Height of Waves.	Length of Waves in Feet.							
	25	50	75	100	150	200	300	[illegible]
50	.04	.23	.64	1.31	3.62	7.43	20.46	[illegible]
40	.06	.36	1.00	2.05	5.65	11.59	31.95	[illegible]
30	.12	.64	1.77	3.64	10.02	20.57	56.70	[illegible]
20	.25	1.44	3.96	8.13	21.79	45.98	[illegible]	[illegible]
15	.42	2.83	6.97	14.31	39.43	80.94	223.06	[illegible]
10	.98	5.53	15.24	31.29	86.22	177.00	487.75	[illegible]
5	3.30	18.68	51.48	105.68	291.20	597.78	[illegible]	[illegible]

The figures are correct for trochoidal deep-sea waves only, but they give a close approximation for any nearly regular series of waves in deep water, and a fair approximation for waves in shallow water.

The question of the practical utilization of the energy which exists in ocean waves divides itself into several parts:

1. The various motions of the water which may be utilized for power purposes.

2. The wave motor proper. That is, the portion of the apparatus in direct contact with the water, and receiving and transmitting the energy thereof, together with the mechanism for transmitting this energy to the machinery for utilizing the same.

3. Regulating devices, for obtaining a uniform motion from the irregular and more or less spasmodic action of the waves, as well as for adjusting the apparatus to the state of the tide and condition of the sea.

4. Storage arrangements for insuring a continuous and uniform output of power during a calm, or when the waves are comparatively small.

The motions that may be utilized for power purposes are the following: 1. Vertical rise and fall of particles at and near the surface. 2. Horizontal to-and-fro motion of particles at and near the surface. 3. Varying slope of surface of wave. 4. Impetus of waves rolling up the beach in the form of breakers. 5. Motion of distorted verticals. All of these motions, except the last one mentioned, have at various times been proposed to be utilized for power purposes; and the last is proposed to be used in apparatus described by Mr. Stahl.

The motion of distorted verticals is thus defined: A set of particles, originally in the same vertical straight line when the water is at rest, does not remain in a vertical line during the passage of the wave; so that the line connecting a set of such particles, while vertical and straight in still water, becomes distorted, as well as displaced, during the passage of the wave, its upper portion moving farther and more rapidly than its lower portion.

Mr. Stahl's paper contains illustrations of several wave-motors designed upon various principles. His conclusions as to their practicability is as follows: "Possibly none of the methods described in this paper may ever prove commercially successful; indeed the problem may not be susceptible of a financially successful solution. My own investigations, however, so far as I have yet been able to carry them, incline me to the belief that wave-power can and will be utilized on a paying basis."

Continuous U[illegible]tion of Tidal Power. (P. Decoeur, Proc. [illegible] C. E. 1890.)—[illegible] with the training-walls to be constructed ary of the Seine, it is proposed to construct large basins, by means h the power available from the rise and fall of the tide could be util- he method proposed is to have two basins separated by a bank rising igh water, within which turbines would be placed. The upper basin be in communication with the sea during the higher one third of the ange, rising, and the lower basin during the lower one third of the ange, falling. If H be the range in feet, the level in the upper would never fall below $\frac{2}{3}H$ measured from low water, and the the lower basin would never rise above $\frac{1}{3}H$. The available head between $0.53H$ and $0.80H$, the mean value being $\frac{2}{3}H$. If S square feet area of the lower basin, and the above conditions are fulfilled, a y $1/3SH$ cu. ft. of water is delivered through the turbines in the space hours. The mean flow is, therefore, $SH \div 99,900$ cu. ft. per sec., and, an fall being $\frac{2}{3}H$, the available gross horse-power is about $1/30S'H^2$, S' is measured in acres. This might be increased by about one third riation of level in the basins amounting to $\frac{1}{6}H$ were permitted. But ch this end the number of turbines would have to be doubled, the head being reduced to $\frac{1}{3}H$, and it would be more difficult to transmit tant power from the turbines. The turbine proposed is of an improved designed to utilize a large flow with a moderate diameter. One has lesigned to produce 300 horse-power, with a minimum head of 5 ft. 3 a speed of 15 revolutions per minute, the vanes having 13 ft. internal ter. The speed would be maintained constant by regulating sluices.

PUMPS AND PUMPING ENGINES.

eoretical Capacity of a Pump.—Let Q' = cu. ft. per min.; Amer. gals. per min. = $7.4805Q'$; d = diam. of pump in inches; l = in inches; N = number of single strokes per min.

$$\text{Capacity in cu. ft. per min.} = Q' = \frac{\pi}{4} \cdot \frac{d^2}{144} \cdot \frac{lN}{12} = .0004545Nd^2l;$$

$$\text{Capacity in gals. per min. } G' = \frac{\pi}{4} \cdot \frac{Nd^2l}{231} \ldots\ldots\ldots = .0034Nd^2l;$$

$$\text{Capacity in gals. per hour } \ldots\ldots\ldots\ldots\ldots = .204Nd^2l.$$

$$\left.\begin{array}{l}\text{Diameter required for a}\\ \text{given capacity per min.}\end{array}\right\} d = 46.9\sqrt{\frac{Q'}{Nl}} = 17.15\sqrt{\frac{G'}{Nl}}.$$

v = piston speed in feet per min., $d = 13.54\sqrt{\frac{Q'}{v}} = 4.95\sqrt{\frac{G'}{v}}$.

he piston speed is 100 feet per min.:

$Nl = 1200$, and $d = 1.354\sqrt{Q'} = .495\sqrt{G'}$; $G' = 4.08d^2$ per min.

e actual capacity will be from 60% to 95% of the theoretical, according to ightness of the piston, valves, suction-pipe, etc.

heoretical Horse-power required to raise Water to a en Height.—Horse-power =

$$\frac{\text{me in cu. ft. per min.} \times \text{pressure per sq. ft.}}{33,000} = \frac{\text{Weight} \times \text{height of lift}}{33,000}.$$

= cu. ft. per min.; G' = gals. per min.; W = wt. in lbs.; P = pressure s. per sq. ft.; p = pressure in lbs. per sq. in.; H = height of lift in ft.; $62.36Q'$, $P = 144p$, $p = .433H$, $H = 2.309p$, $G' = 7.4805Q'$.

$$\text{HP} = \frac{Q'P}{33,000} = \frac{Q'H \times 144 \times .433}{33,000} = \frac{Q'H}{529.2} = \frac{G'H}{3958.7};$$

$$\text{HP} = \frac{WH}{33,000} = \frac{Q' \times 62.36 \times 2.309p}{33,000} = \frac{Q'p}{229.2} = \frac{G'p}{1714.5}.$$

r the actual *horse-power required* an allowance must be made for the *ion, slips, etc.,* of engine, pump, valves, and passages.

Depth of Suction.—Theoretically a perfect pump will draw from a height of nearly 34 feet, or the height corresponding to a vacuum (14.7 lbs. × 2.309 = 33.95 feet); but since a perfect vacuum cannot be obtained, on account of valve-leakage, air contained in the water, and vapor of the water itself, the actual height is generally less than When the water is warm the height to which it can be lifted by suction decreases, on account of the increased pressure of the vapor. In pumping water, therefore, the water must flow into the pump by gravity. The following table shows the theoretical maximum depth of suction for different temperatures, leakage not considered:

Temp. F.	Absolute Pressure of Vapor, lbs. per sq. in.	Vacuum in Inches of Mercury.	Max. Depth of Suction, feet.	Temp. F.	Absolute Pressure of Vapor, lbs. per sq. in.	Vacuum in Inches of Mercury.
101.4	1	27.88	31.6	183.0	8	13.63
126.2	2	25.85	29.3	188.4	9	11.59
144.7	3	23.81	27.0	193.2	10	9.55
153.3	4	21.77	24.7	197.6	11	7.51
162.5	5	19.74	22.4	201.9	12	5.48
170.3	6	17.70	20.1	205.8	13	3.44
177.0	7	15.66	17.8	209.6	14	1.40

Amount of Water raised by a Single-acting Lift-p
—It is common to estimate that the quantity of water raised single-acting bucket-valve pump per minute is equal to the num strokes in one direction per minute, multiplied by the volume trave the piston in a single stroke, on the theory that the water rises in th only when the piston or bucket ascends; but the fact is that the col water does not cease flowing when the bucket descends, but flows tinuously through the valve in the bucket, so that the discharge pump, if it is operated at a high speed, may amount to nearly dou calculated from the displacement multiplied by the number of single in one direction.

Proportioning the Steam-cylinder of a Direct-
Pump.—Let

A = area of steam-cylinder; a = area of pump-cylinder;
D = diameter of steam-cylinder; d = diameter of pump-cylind
P = steam-pressure, lbs. per sq. in.; p = resistance per sq. in. on
H = head = $2.309p$; $p = .433H$;

$$E = \text{efficiency of the pump} = \frac{\text{work done in pump-cylinder}}{\text{work done by the steam-cylinder}},$$

$$A = \frac{ap}{EP};\quad a = \frac{EAP}{p};\quad D = d\sqrt{\frac{p}{EP}};\quad d = D\sqrt{\frac{EP}{p}};\quad P = \frac{ap}{EA};\ p$$

$$\frac{A}{a} = \frac{p}{EP} = \frac{.433H}{EP};\quad H = 2.309EP\frac{A}{a};\quad \text{If } E = 75\%,\ H = 1.732P$$

E is commonly taken at 0.7 to 0.8 for ordinary direct-acting pum the highest class of pumping-engines it may amount to 0.9. Th pressure P is the mean effective pressure, according to the indica gram; the water-pressure p is the mean total pressure acting on th plunger or piston, including the suction, as could be shown by an in diagram of the water-cylinder. The pressure on the pump-pisto quently much greater than that due to the height of the lift, on acc the friction of the valves and passages, which increases rapidly with of flow.

Speed of Water through Pipes and Pump-pass
The speed of the water is commonly from 100 to 200 feet per minute feet per minute is exceeded, the loss from friction may be consider

The diameter of pipe required is $4.95\sqrt{\frac{\text{gallons per minute}}{\text{velocity in feet per mi}}}$

a ve² per minute, diameter = 35 × √gall

zes of Direct-acting Pumps.—The two following tables are se-
d from catalogues of manufacturers, as representing the two common
s of direct-acting pump, viz., the single-cylinder and the duplex.
types are now made by most of the leading manufacturers.

The Deane Direct-acting Pump.

STANDARD SIZES FOR ORDINARY SERVICE.

Diameter of Water-cylinder in In.	Length of Stroke in In.	Gallons per Stroke.	Strokes per Minute.	Capacity per Minute at Given Speed. Stks.	Gals.	Extreme Length in Inches.	Extreme Width in Inches.	Size of Steam Supply-pipe.	Size of Steam Exhaust-pipe.	Size of Suction.	Size of Discharge.
3½	5	.14	1 to 300	130	18	33	9½	½	¾	2	1½
4	5	.27	1 to 300	130	35	33	9½	½	¾	2	1½
4	7	.39	1 to 300	125	49	45½	15	¾	1	3	2½
5	7	.51	1 to 275	125	64	45½	15	¾	1	3	2½
5½	7	.72	1 to 275	125	90	45½	15	¾	1	3	2½
7	10	1.64	1 to 250	110	180	58	17	1	1½	5	4
7½	10	1.91	1 to 250	110	210	58	17	1	1½	5	4
8	10	2.17	1 to 250	110	239	58	17	1	1½	5	4
6	12	1.47	1 to 250	100	147	67	20½	1	1½	4	4
7	12	2.00	1 to 250	100	200	67	20½	1	1½	5	4
8	12	2.61	1 to 250	100	261	68	30	1	1½	5	5
10	12	4.08	1 to 250	100	408	68	20½	1	1½	8	8
8	12	2.61	1 to 250	100	261	68½	30	1½	2	5	5
10	12	4.08	1 to 250	100	408	68½	30	1½	2	8	8
12	12	5.87	1 to 250	100	587	68½	30	1½	2	8	8
10	12	4.08	1 to 250	100	408	64	24	2	2½	8	8
10	18	6.12	1 to 200	70	428	68½	30	2	2½	8	8
12	12	5.87	1 to 250	100	587	64	28½	2	2½	8	8
12	18	8.80	1 to 175	70	616	88	28½	2	2½	8	8
14	18	12.00	1 to 175	70	840	88	28½	2	2½	8	8
10	12	4.08	1 to 250	100	408	69	30	2	2½	8	8
10	18	6.12	1 to 175	70	428	93	25	2	2½	8	8
10	24	8.16	1 to 150	50	408	112	26	2	2½	8	8
12	12	5.87	1 to 250	100	587	69	30	2	2½	8	8
12	18	8.80	1 to 175	70	616	88	28½	2	2½	8	8
12	24	11.75	1 to 150	50	587	112	26	2	2½	10	8
14	24	15.99	1 to 150	50	800	112	34	2	2½	12	10
16	16	13.92	1 to 175	80	1114	84	34	2	2½	12	10
16	24	20.88	1 to 150	50	1044	112	38	2	2½	12	10
14	18	12.00	1 to 175	70	840	89	27	2	2½	8	8
14	24	15.99	1 to 150	50	800	109	34	2	2½	12	10
16	16	13.92	1 to 175	80	1114	85	34	2	2½	12	10
16	24	20.88	1 to 150	50	1044	115	34	2	2½	12	10
18	24	26.43	1 to 125	50	1322	115	40	2	2½	14	12
16	24	20.88	1 to 125	50	1044	118	38	3	3½	12	10
18	24	26.43	1 to 125	50	1322	118	40	3	3½	14	12
20	24	32.64	1 to 125	50	1632	118	40	3	3½	16	14
18	24	26.43	1 to 125	50	1322	118	40	3	3½	14	12
20	24	32.64	1 to 125	50	1632	118	40	3	3½	16	14
22	24	39.50	1 to 125	50	1975	120	40	3	3½	18	14

fficiency of Small Direct-acting Pumps.—Chas. E. Emery, in
orts of Judges of Philadelphia Exhibition, 1876, Group xx., says: "Ex-
ments made with steam-pumps at the American Institute Exhibition of
showed that average sized *steam-pumps* do not, on the average, utilize
e than 50 *per cent of the indicated* power in the steam-cylinders, the re-
nder being absorbed in the friction of the engine, but more particularly
passage of the water through the pump. Again, all ordinary steam-
for miscellaneous uses require that the steam-cylinder [illegible]
four times the area of the water-cylinder to give suffi[illegible]

when the steam is accidentally low; hence as such pumps usually w[ork] against the atmospheric pressure, the net or effective pressure for[ms a] small percentage of the total pressure, which, with the large ext[ernal] radiating surface exposed and the total absence of expansion, makes [the] expenditure of steam very large. One pump tested required 120 po[unds] weight of steam per indicated horse-power per hour, and it is believed [that] the cost will rarely fall below 60 pounds; and as only 50 per cent of the [in]dicated power is utilized, it may be safely stated that ordinary steam-pu[mps] rarely require less than 120 pounds of steam per hour for each horse-po[wer] utilized in raising water, equivalent to a duty of only 15,000,000 foot-pou[nds] per 100 pounds of coal. With larger steam-pumps, particularly when t[hey] are proportioned for the work to be done, the duty will be materially [in]creased."

The Worthington Duplex Pump.

STANDARD SIZES FOR ORDINARY SERVICE.

Diameter of Steam-cylinders.	Diameter of Water-plungers.	Length of Stroke.	Displacement in Gallons per Stroke of One Plunger.	Proper Strokes per Minute of One Plunger, varying with kind of work and pressure.	Gallons delivered per Minute by both Plungers at stated Number of Strokes.	Diameter of Plunger required in any single-cylinder pump to do the same work at same speed.	Sizes of Pipes for Short Lengths. To be increased as length increases. Steam-pipe.	Exhaust-pipe.	Suction-pipe.
3	2	3	.04	100 to 250	8 to 20	2⅞	⅜	½	1¼
4½	2¾	4	.10	100 to 200	20 to 40	4	½	¾	2
5¼	3½	5	.20	100 to 200	40 to 80	5	¾	1¼	2½
6	4	6	.33	100 to 150	70 to 100	5⅝	1	1½	3
7½	4½	6	.42	100 to 150	85 to 125	6⅜	1¼	2	4
7½	5	6	.51	100 to 150	100 to 150	7	1½	2	4
7½	4½	10	.69	75 to 125	100 to 170	6⅜	1½	2	4
9	5¼	10	.93	75 to 125	135 to 230	7½	2	2½	4
10	6	10	1.22	75 to 125	180 to 300	8½	2	2½	5
10	7	10	1.66	75 to 135	245 to 410	9⅞	2	2½	6
12	7	10	1.66	75 to 125	245 to 410	9⅞	2½	3	6
14	7	10	1.66	75 to 125	215 to 410	9⅞	2½	3	6
12	8½	10	2.45	75 to 125	365 to 610	12	2½	3	6
14	8½	10	2.45	75 to 125	365 to 610	12	2½	3	6
16	8½	10	2.45	75 to 125	365 to 610	12	2½	3	6
18½	8½	10	2.45	75 to 125	365 to 610	12	3	3½	6
20	8½	10	2.45	75 to 125	365 to 610	12	4	5	6
12	10¼	10	3.57	75 to 125	530 to 890	14¼	2½	3	8
14	10¼	10	3.57	75 to 125	530 to 890	14¼	2½	3	8
16	10¼	10	3.57	75 to 125	530 to 890	14¼	2½	3	8
18½	10¼	10	3.57	75 to 125	530 to 890	14¼	3	3½	8
20	10¼	10	3.57	75 to 125	530 to 890	14¼	4	5	8
14	12	10	4.89	75 to 125	730 to 1220	17	2½	3	10
16	12	10	4.89	75 to 125	730 to 1220	17	2½	3	10
18½	12	10	4.89	75 to 125	730 to 1220	17	3	3½	10
20	12	10	4.89	75 to 125	730 to 1220	17	4	5	10
18½	14	10	6.66	75 to 125	990 to 1660	19¾	3	3½	12
20	14	10	6.66	75 to 122	990 to 1660	19¾	4	5	12
17	10	15	5.10	50 to 100	510 to 1020	14	3	3½	12
20	12	15	7.34	50 to 100	730 to 1460	17	4	5	12
20	15	15	11.47	50 to 100	1145 to 2290	21	....	....	.....
25	15	15	11.47	50 to 100	1145 to 2290	21	—	—	.....

peed of Piston.—A piston speed of 100 feet per minute is commonly umed as correct in practice, but for short-stroke pumps this gives too a speed of rotation, requiring too frequent a reversal of the valves. long stroke pumps, 2 feet and upward, this speed may be considerably eded, if valves and passages are of ample area.

mber of Strokes required to Attain a Piston Speed from 50 to 125 Feet per Minute for Pumps having Strokes from 3 to 18 Inches in Length.

per min.	Length of Stroke in Inches.									
	3	4	5	6	7	8	10	12	15	18
	Number of Strokes per Minute.									
50	200	150	120	100	86	75	60	50	40	33
55	220	165	132	110	94	82.5	66	55	44	37
60	240	180	144	120	103	90	72	60	48	40
65	260	195	156	130	111	97.5	78	65	52	43
70	280	210	168	140	120	105	84	70	56	47
75	300	225	180	150	128	112.5	90	75	60	50
80	320	240	192	160	137	120	96	80	64	53
85	340	255	204	170	146	127.5	102	85	68	57
90	360	270	216	180	154	135	108	90	72	60
95	380	285	228	190	163	142.5	114	95	76	63
0	400	300	240	200	171	150	120	100	80	67
5	420	315	252	210	180	157.5	126	105	84	70
0	440	330	264	220	188	165	132	110	88	73
5	460	345	276	230	197	172.5	138	115	92	77
0	480	360	288	240	206	180	144	120	96	80
5	500	375	300	250	214	187.5	150	125	100	83

Piston Speed of Pumping-engines. (John Birkinbine, Trans. . M. E., v. 459.)—In dealing with such a ponderous and unyielding subnce as water there are many difficulties to overcome in making a pump k with a high piston speed. The attainment of moderately high speed however, easily accomplished. Well-proportioned pumping-engines of e capacity, provided with ample water-ways and properly constructed es, are operated successfully against heavy pressures at a speed of 250 ft. minute, without "thug," concussion, or injury to the apparatus, and re is no doubt that the speed can be still further increased.

peed of Water through Valves.—If areas through valves and er passages are sufficient to give a velocity of 250 ft. per min. or less, y are ample. The water should be carefully guided and not too abruptly lected. (F. W. Dean, *Eng. News*, Aug. 10, 1893.)

Boiler-feed Pumps.—Practice has shown that 100 ft. of piston speed minute is the limit, if excessive wear and tear is to be avoided.

The velocity of water through the suction-pipe must not exceed 200 ft. r minute, else the resistance of the suction is too great.

The approximate size of suction-pipe, where the length does not exceed ft. and there are not more than two elbows, may be found as follows; 7/10 of the diameter of the cylinder multiplied by 1/100 of the piston speed feet. For duplex pumps of small size, a pipe one size larger is usually nployed. The velocity of flow in the discharge-pipe should not exceed 0 ft. per minute. The volume of discharge and length of pipe vary so eatly in different installations that where the water is to be forced more han 50 ft. the size of discharge-pipe should be calculated for the particular onditions, allowing no greater velocity than 500 ft. per minute. The size of ischarge-pipe is calculated in single-cylinder pumps from 250 to 400 ft. per inute. Greater velocity is permitted in the larger pipes.

In determining the proper size of pump for a steam-boiler, allowances must be made for a supply of water sufficient to cover all the demands of ngines, steam-heating, etc., up to the capacity of generator, and should not be calculated simply according to the requirements of the engine. In practice engines use all the way from 12 up to 50, or more, pounds of steam per H.P. per hour when being worked up to capacity. When an engine is overloaded or *underloaded more water per* H.P. will be required ... *operating at its rated capacity.* The average run of horiz...

boilers will evaporate from 2 to 3 lbs. of water per sq. ft. of heating-surf per hour, but may be driven up to 6 lbs. if the grate-surface is too larg the draught too great for economical working.

Pump-Valves.—A. F. Nagle (Trans. A. S. M. E., x. 521) gives a num of designs with dimensions of double-beat or Cornish valves used in la pumping-engines, with a discussion of the theory of their proportions. following is a summary of the proportions of the valves described.

SUMMARY OF VALVE PROPORTIONS.

Location of Engine.	Diam. of Valve in inches.	Weight in Water per square inch of Inside Un-balanced Area, in lbs.	Ratio of Seat-area to Inside Un-balanced Area.	Pressure upon Seat per sq. in., in lbs.	Action.
Providence high-service engine	12	1 lb. reduced to .66 lb.	16%	377 lbs.	Good
Providence Cornish-engine............	16	1.28	12	680	Good
St. Louis Water Wks.	16	1.86	67	250	Some no
Milwaukee " "	7	.40	88	120	Some noi high spe
Chicago " "	25	1.41	75	151	Noisy
" " "	15	1.31	85	140	"
" " " wood seats.........	15	1.16	94	132	"
Chicago Water Wks.	8	.96	75	151	"

Mr. Nagle says: There is one feature in which the Cornish valves necessarily defective, namely, the lift must always be quite large, unless g power is sacrificed to reduce it. It is undeniable that a small lift is pr able to a great one, and hence it naturally leads to the substitutio numerous small valves for one or several large ones. To what extrem duction of size this view might safely lead must be left to the judgme the engineer for the particular case in hand, but certainly, theoretically must adopt small valves. Mr. Corliss at one time carried the theor far as to make them only 1⅝ inches in diameter, but from 3 to 4 inch the more common practice now. A small valve presents proportionat larger surface of discharge with the same lift than a larger valve, so whatever the total area of valve-seat opening, its full contents can be charged with less lift through numerous small valves than with one l one.

Henry R. Worthington was the first to use numerous small rubber va in preference to the larger metal valves. These valves work well unde the conditions of a city pumping-engine. A volute spring is generally to limit the rise of the valve.

In the Leavitt high-duty sewerage-engine at Boston (*Am. Machinist*, 31, 1884), the valves are of rubber, ¾-inch thick, the opening in valve being 13⅛ × 4⅛ inches. The valves have iron face and back-plates, form their own hinges.

CENTRIFUGAL PUMPS.

Relation of Height of Lift to Velocity.—The height of depends only on the tangential velocity of the circumference, every tan tial velocity giving a constant height of lift—sometimes termed "head whether the pump is small or large. The quantity of water discharged proportion to the area of the discharging orifices at the circumference, proportion to the square of the diameter, when the breadth is kept the sa R. H. Buel (App. Cyc. Mech., ii, 606) gives the following:

Let Q represent the quantity of water, in cubic feet, to be pumped minute, h the height of suction in feet, h' the height of discharge in feet, d the diameter of suction-pipe, equal to the diameter of discharge-pipe

ccording to Fink, $d = 0.36 \sqrt{\frac{Q}{\sqrt{2g(h+h')}}}$, g being the accel-
to gravity.
tion takes place on one side of the wheel, the inside diameter of equal to $1.2d$, and the outside to $2.4d$. If the suction takes place s of the wheel, the inside diameter of the wheel is equal to $0.85d$, side to $1.7d$. Then the suction-pipe will have two branches, the n equal to half the area of d. The suction-pipe should be as short to prevent air from entering the pump. The tangential velocity r edge of wheel for the delivery Q is equal to $1.25 \sqrt{2g(h+h')}$ ond.
are six in number, constructed as follows: Divide the central , which incloses the outer edges of the two arms, into any num-l parts by dividing the radii, and divide the breadth of the wheel manner by drawing concentric circles. The intersections of the ii with the corresponding circles give points of the arm.
iments with Appold's pump, a velocity of circumference of 500 n. raised the water 1 ft. high, and maintained it at that level charging any; and double the velocity raised the water to four eight, as the centrifugal force was proportionate to the square city; consequently,

0 ft. per min. raised the water 1 ft. without discharge.
0 " " " " " 4 " " "
0 " " " " " 16 " " "
0 " " " " " 64 " " "

test height to which the water had been raised without discharge, riments with the 1-ft. pump, was 67.7 ft., with a velocity of 4153 ., being rather less than the calculated height, owing probably to th the greater pressure. A velocity of 1128 ft. per min. raised the ft. without any discharge, and the maximum effect from the loyed in raising to the same height 5½ ft. was obtained at the 1678 ft. per min., giving a discharge of 1400 gals. per min. from mp. The additional velocity required to effect a discharge of er min., through a 1-ft. pump working at a dead level without any ift, is 550 ft. per min. Consequently, adding this number in each velocity given above, at which no discharge takes place, the fol-ocities are obtained for the maximum effect to be produced in

1050 ft. per min., velocity for 1 ft. height of lift.
1550 " " " " 4 " " "
2550 " " " " 16 " " "
4550 " " " " 64 " " "

neral terms, the velocity in feet per minute for the circumference np to be driven, to raise the water to a certain height, is equal to $\sqrt{\text{height of lift in feet}}$.

ce Centrifugal Pumps, Class B—For Lifts from 15 to 35 ft.

Size of Pipes.		Economical Capacity, in gallons per min.	Total Capacity, in gallons per min.	Horse-power per Ft. Lift, for smaller quantity.
Suction.	Discharge.			
2 in.	1½ in.	20 to 50	150	.024
2½	2	60 to 80	300	.035
3½	3	80 to 160	650	.055
4½	4	160 to 350	1,250	.075
6	5	330 to 600	1,850	.175
6	6	500 to 900	2,600	.22
8	8	1,100 to 2,000	4,750	.45
10	10	1,600 to 3,000	7,500	.62
12	12	2,000 to 3,000	10,000	1.00
14	14	3,000 to 5,000	14,000	1.25
15	15	3,500 to 7,000	16,000	1.40
18	18	6,000 to 11,000	22,000	2.40

Table of Diameters and Width of Pulleys, Width of and Number of Revolutions per Minute Necess raise Minimum Quantity of Water to Different H with Different Sizes of Pumps of Class B.

Size.	Diameter of Pulley.	Width of Pump.	Width of Belt.	Minimum Quantity of Water.	Height in Feet and Revolutions per Minute.							
					6	8	10	12	16	20	25	30
Ins.	Ins.	Ins.	Ins.									
1½	5	5	3	40	465	515	560	605	680	745	820	[illegible]
2	5	5	4	60	425	475	515	560	625	680	750	810
3	7½	7	6	80	390	435	475	510	575	630	695	750
4	7½	7	7	160	365	405	445	475	535	590	645	700
5	12	11	8	330	320	355	390	415	470	520	570	610
6	14	11	9	500	285	315	345	370	415	460	500	540
8	16	12	10	1100	215	240	260	280	310	340	375	410
10	18	12	10	1600	170	190	210	225	250	275	300	[illegible]
12	22	14	12	2000	150	165	185	195	220	240	265	[illegible]
14	24	14	12	3000	135	150	165	175	195	215	240	[illegible]
15	28	15	14	3500	125	145	155	165	190	210	230	245
18	28	16	14	6000	110	120	130	135	160	175	190	[illegible]

Efficiencies of Centrifugal and Reciprocating Pu W. O. Webber (Trans. A. S. M. E., vii. 598) gives diagrams sho relative efficiencies of centrifugal and reciprocating pumps, from w following figures are taken for the different lifts stated:

Lift, feet:
2 5 10 15 20 25 30 35 40 50 60 80 100 120 160 200
Efficiency reciprocating pump:
.. .. .30 .45 .55 .61 .66 .68 .71 .75 .77 .82 .85 .87 .90 .9
Efficiency centrifugal pump:
.50 .56 .64 .68 .69 .68 .66 .62 .58 .50 .40 ... ...

The term efficiency here used indicates the value of W. H. P. ÷ or horse-power of the water raised divided by the indicated horse- the steam-engine, and does not therefore show the full efficiency of th but that of the combined pump and engine. It is, however, a ver way of showing the relative values of different kinds of pumping having their motive power forming a part of the plant.

The highest value of this term, given by Mr. Webber, is .9164 for 170 ft., and 3615 gals. per min. This was obtained in a test of the pumping-engine at Lawrence, Mass., July 24, 1879.

With reciprocating pumps, for higher lifts than 170 ft., the curve ciencies falls, and from 200 to 300 ft. lift the average value see .84. Below 170 ft. the curve also falls reversely and slowly, until at ft. its descent becomes more rapid, and at 85 ft. .727 appears recorded performance. There are not any very satisfactory recor this lift, but some figures are given for the yearly coal consum total number of gallons pumped by engines in Holland under a from which an efficiency of .44 has been deduced.

With centrifugal pumps, the lift at which the maximum efficien tained is approximately 17 ft. At lifts from 12 to 18 ft. some m large experience claim now to obtain from 65% to 70% of useful ef .613 appears to be the best done at a public test under 14.7 ft. head.

The drainage-pumps constructed some years ago for the Haarl were designed to lift 70 tons per min. 15 ft., and they weighed tons. Centrifugal pumps for the same work weigh only 5 tons. Th of a centrifugal pump and engine to lift 10,000 gals. per min. 35 f 6 tons.

The pumps placed by Gwynne at the Ferrara Marshes, Northern 1865, are, it is believed, capable of handling more water than oth pumping-engines in existence. The work performed by these pum *lifting of 2000 tons per min.*—over 600,000,000 gals. per 24 hours—or *lift of about 10 ft.* [illegible] of 12.5 ft.). (See Engineering, 1875.) The efficiency [illegible] pumps seems to increase as the es, approximately as follows: A 2″ pump (this designation meaning the size of discharge-outlet in inches of diameter), giving an effi- of 38%, a 3″ pump 45%, and a 4″ pump 52%, a 5″ pump 60%, and a 6″ 4% efficiency.

Tests of Centrifugal Pumps.

W. O. Webber, Trans. A. S. M. E., ix. 237.

ker.	Andrews.	Andrews.	Andrews.	Heald & Sisco.	Heald & Sisco.	Heald & Sisco.	Berlin. Schwartz-kopff.
....	No. 9.	No. 9.	No. 9.	No. 10.	No. 10.	No. 10.	No. 9.
ischarge.	9⅛″	9⅛″	9⅛″	10″	10″	10″	9¼″
uction ...	9¾″	9¾″	9¾″	12″	12″	12″	10.3″
isk......	26″	26″	26″	30.5″	30.5″	30.5″	20.5″
r minute.	191.9	195.5	200.5	188.3	202.7	213.7	500
er minute	1513.12	2023.82	2499.33	1673.37	2044.9	2371.67	1944.8
in feet....	12.25	12.62	13.08	12.33	12.58	13.0	16.46
H.P.......	4.69	6.47	8.28	5.22	6.51	7.81	....
eter H.P.	10.09	12.2	14.38	8.11	10.74	14.02	11
cy........	46.52	53.0	57.57	64.5	60.74	55.72	73.1

es of Centrifugal Pumps.—For forms of pump vanes, see by W. O. Webber, Trans. A. S. M. E., ix. 228, and discussion thereon fs. Thurston, Wood, and others.

Centrifugal Pump used as a Suction Dredge.—The ws centrifugal pump was used by Gen. Gillmore, U. S. A., in 1871, in ing the channel over the bar at the mouth of the St. John's River, a. The pump was a No. 9, with suction and discharge pipes each 9 diam. It was driven at 300 revolutions per minute by belt from an developing 26 useful horse-power.

ough 200 revolutions of the pump disk per minute will easily raise llons of clear water 12 ft. high, through a straight vertical 9-inch 00 revolutions were required to raise 2500 gallons of sand and water igh, through two inclined suction-pipes having two turns each, dis- d through a pipe having one turn.

proportion of sand that can be pumped depends greatly upon its e gravity and fineness. The calcareous and argillaceous sands flow freely than the silicious, and fine sands are less liable to choke the han those that are coarse. When working at high speed, 50% to 55% of an be raised through a straight vertical pipe, giving for every 10 cubic of material discharged 5 to 5½ cubic yards of compact sand. With ppliances used on the St. John's bar, the proportion of sand seldom led 45%, generally ranging from 30% to 35% when working under the favorable conditions.

umping 2500 gallons, or 12.6 cubic yards of sand and water per minute, would therefore be obtained from 3.7 to 4.3 cubic yards of sand. Dur- e early stages of the work, before the teeth under the drag had been rly arranged to aid the flow of sand into the pipes, the yield was con- ably below this average. (From catalogue of Jos. Edwards & Co., of the Andrews Pump, New York.)

DUTY TRIALS OF PUMPING-ENGINES.

committee of the A. S. M. E. (Trans., xii. 530) reported in 1891 on a ard method of conducting duty trials. Instead of the old unit of of foot-pounds of work per 100 lbs. of coal used, the committee recom- d a new unit, foot-pounds of work per million heat-units furnished by oiler. The variations in quality of coal make the old standard unfit as s of duty ratings. The new unit is the precise equivalent of 100 lbs. of in cases where each pound of coal imparts 10,000 heat-units to the r in the boiler, or where the evaporation is 10,000 ÷ 965.7 = 10.355 lbs. of r from and at 212° per pound of fuel. This evaporative result is readily ned from all grades of *Cumberland bituminous* coal, used in [illegible] n tubular *boilers, and, in many cases, from the best grade* [illegible] *oal.*

FEED-WATER.

16. Weight of water supplied to boiler by main feed-pump........ lbs.
17. Weight of water supplied to boiler from various other sources. lbs.
18. Total weight of feed-water supplied from all sources........... lbs.

PRESSURES.

19. Boiler pressure indicated by gauge.............................. lbs.
20. Pressure indicated by gauge on force main.......................... lbs.
21. Vacuum indicated by gauge on suction main...................... ins.
22. Pressure corresponding to vacuum given in preceding line...... lbs.
23. Vertical distance between the centres of the two gauges....... ins.
24. Pressure equivalent to distance between the two gauges........ lbs.

MISCELLANEOUS DATA.

25. Duration of trial.. hrs.
26. Total number of single strokes during trial........ ....
27. Percentage of moisture in steam supplied to engine, or number of degrees of superheating........... .... % or
28. Total leakage of pump during trial, determined from results of leakage test .... lbs.
29. Mean effective pressure, measured from diagrams taken from steam-cylinders .... M.E.P.

PRINCIPAL RESULTS.

30. Duty.. ft. lbs.
31. Percentage of leakage... %
32. Capacity... gals.
33. Percentage of total friction... %

ADDITIONAL RESULTS.

34. Number of double strokes of steam-piston per minute ....
35. Indicated horse-power developed by the various steam-cylinders I.H.P.
36. Feed-water consumed by the plant per hour..... .... lbs.
37. Feed-water consumed by the plant per indicated horse-power per hour, corrected for moisture in steam. .... lbs.
38. Number of heat units consumed per indicated horse-power per hour........ .... B.T.U.
39. Number of heat units consumed per indicated horse-power per minute........... .... B.T.U.
40. Steam accounted for by indicator at cut-off and release in the various steam-cylinders.......... .... lbs.
41. Proportion which steam accounted for by indicator bears to the feed-water consumption.... ....
42. Number of double strokes of pump per minute.. .. ..
43. Mean effective pressure, measured from pump diagrams ...... M.E.P.
44. Indicated horse-power exerted in pump-cylinders....... ...... I.H.P.
45. Work done (or duty) per 100 lbs. of coal .... ft. lbs.

SAMPLE DIAGRAM TAKEN FROM STEAM-CYLINDERS.

(Also, if possible, full measurement of the diagrams, embracing pressures at the initial point, cut-off, release, and compression; also back pressure, and the proportions of the stroke completed at the various points noted.)

SAMPLE DIAGRAM TAKEN FROM PUMP-CYLINDERS.

These are not necessary to the main object, but it is desirable to [illegible] them.

DATA AND RESULTS OF BOILER TEST.

(In accordance with the scheme recommended by the Boiler-test Committee of the Society.)

VACUUM PUMPS—AIR-LIFT PUMP.

The Pulsometer.—In the pulsometer the water is raised by [illegible] into the pump-chamber by the condensation of steam within it, and [illegible] forced into the delivery-pipe by the pressure of a new quantity of steam [illegible] the surface of the water. Two chambers are used which work alternately, one raising while the other is discharging.

Test of a Pulsometer.—A test of a pulsometer is described by De V. Wood in Trans. A. S. M. E., xiii. It had a 3½-inch suction-pipe, stood [illegible] high, and weighed [illegible] The steam-pipe [illegible] diameter. A throttle was placed [illegible]

o, and pressure gauges placed on both sides of the throttle, well and thermometer placed beyond the throttle. The wire throttling caused superheating.
of steam used were computed from the increase of the tem e water in passing through the pump.

m × loss of heat = lbs. of water sucked in × increase of temp.

eat in a pound of steam is the total heat in a pound of satu- found from "steam tables" for the given pressure, plus the eating, minus the temperature of the discharged water; or

$$\text{unds of steam} = \frac{\text{lbs. water} \times \text{increase of temp.}}{H - 0.48t - T.}$$

'or the four tests are given in the following table:

nd Results.	Number of Test.			
	1	2	3	4
inute	71	60	57	64
pipe before throttl'g	114	110	127	104.3
n pipe after throttl'g	19	30	43.8	26.1
fter throttling, deg. F.	270.4	277	309.0	270.1
of superheat'g, deg. F.	3.1	3.4	17.4	1.4
det'd from temp., lbs.	1617	931	1518	1019.9
d, lbs.	404,786	186.362	228,425	248,053
efore entering pump,	75.15	80.6	76.3	70.25
rise of	4.47	5.5	7.49	4.55
y gauge on lift, ft.	29.90	54.05	54.05	29.90
y gauge on suction	12.26	12.26	19.67	19.67
y gauge, total (H)	42.16	66.31	73.72	49.57
y measure, total (h)	32.8	57.80	66.6	41.60
ion of plant (h) ÷ (H)	0.777	0.877	0.911	0.839
pulsometer	0.012	0.0155	0.0126	0.0138
exclusive of boiler	0.0093	0.0136	0.0115	0.0116
if that of boiler be 0.7	0.0065	0.0095	0.0080	0.0081
aporates 10 lbs. water	10,511,400	13,391,000	11,059,000	12,036,300

tests having the highest lift (54.05 ft.), that was more efficient e smaller suction (12.26 ft.), and this was also the most efficient ests. But, on the other hand, the other two tests having the ft.), that was the more efficient which had the greater suction t no law in this regard was established. The pressures used, .1, follow the order of magnitude of the total heads, but are nal thereto. No attempt was made to determine what press- e the best efficiency for any particular head. The pressure used to a practical runner, and he judged that when the pump was larly and well, the pressure then existing was the proper one. that, in the first test, a pressure of 19 lbs. of steam should pro- r number of strokes and pump over 50% more water than 26.1 eing the same, as in the fourth experiment.
nery in discussion of Prof. Wood's paper says, referring to y himself and others at the Centennial Exhibition in 1876 (see e Judges, Group xx.), that a vacuum-pump tested by him in uty of 4.7 millions; one tested by J. F. Flagg, at the Cincinnati 1875, gave a maximum duty of 3.25 millions. Several vacuum am-pumps, compared later on the same basis, were reported duties of 10 to 11 millions, the steam-pumps doing no better uum-pumps. Injectors, when used for lifting water not re- heated, have an efficiency of 2 to 5 millions; vacuum-pumps ly between 3 and 10; small steam-pumps between 8 and 15; -pumps, between 15 and 30, and pumping-engines between 30 ons.
h record of test of a pulsometer is given in *Eng'g*, Nov. 24 Height of *suction 11.27 ft.*; total height of lift, 102.6 ft *of delivery-pipe, 118 ft.*; quantity delivered per hou s. *Weight of steam used* per H. P. per hour, 92.76 lb

done per pound of steam 21,345 foot-pounds, equal to a duty of foot-pounds pe 100 lbs. of coal, if 10 lbs of steam were gener pound of coal.

The Jet-pump.—This machine works by means of the tend stream or jet of fluid to drive or carry contiguous particles of flu with it. The water-jet pump, in its present form, was invented James Thomson, and first described in 1852. In some experime small scale as to the efficiency of the jet-pump, the greatest effici found to take place when the depth from which the water was draw suction-pipe was about nine tenths of the height from which the to form the jet; the flow up the suction-pipe being in that case a fifth of that of the jet, and the efficiency, consequently, 9/10 × 1/ This is but a low efficiency; but it is probable that it may be incr improvements in proportions of the machine. (Rankine, S. E.)

The Injector when used as a pump has a very low efficien Injectors, under Steam-boilers.)

Air-lift Pump.—The air-lift pump consists of a vertical w with its lower end submerged in a well, and a smaller pipe deliv into it at the bottom. The rising column in the pipe consists of air with water, the air being in bubbles of various sizes, and is therefo than a column of water of the same height; consequently the wat pipe is raised above the level of the surrounding water. This m raising water was proposed as early as 1797, by Loescher, of Frei was mentioned by Collon in lectures in Paris in 1876, but its first application probably was by Werner Siemens in Berlin in 1885. Pohle experimented on the principle in California in 1886, and U. S on apparatus involving it were granted to Pohle and Hill in the sa A paper describing tests of the air-lift pump made by Randall, Br Behr was read before the Technical Society of the Pacific Coast in

The diameter of the pump-column was 3 in., of the air-pipe 0.9 of the air-discharge nozzle 5/8 in. The air-pipe had four sharp ben length of 35 ft. plus the depth of submersion.

The water was pumped from a closed pipe-well (55 ft. deep and diameter). The efficiency of the pump was based on the least w retically required to compress the air and deliver it to the receiver efficiency of the compressor be taken at 70%, the efficiency of the p compressor together would be 70% of the efficiency found for t alone.

For a given submersion (h) and lift (H), the ratio of the two be within reasonable limits, (H) being not much greater than (h), the was greatest when the pressure in the receiver did not greatly ex head due to the submersion. The smaller the ratio $H \div h$, the hi the efficiency.

The pump, as erected, showed the following efficiencies:

For $H \div h =$	0.5	1.0	1.5	2.0
Efficiency =	50%	40%	30%	25%

The fact that there are absolutely no moving parts makes t especially fitted for handling dirty or gritty water, sewage, mi and acid or alkali solutions in chemical or metallurgical works.

In Newark, N. J., pumps of this type are at work having a total of 1,000,000 gallons daily, lifting water from three 8-in. artesian we Newark Chemical Works use an air-lift pump to raise sulphuric ac gravity. The Colorado Central Consolidated Mining Co., in one of at Georgetown, Colo., lifts water in one case 250 ft., using a series

For a full account of the theory of the pump, and details of above referred to, see *Eng'g News*, June 8, 1893.

THE HYDRAULIC RAM.

Efficiency.—The hydraulic ram is used where a considerab water with a moderate fall is available, to raise a small portion of to a height exceeding that of the fall. The following are rules Eytelwein as the results of his experiments (from Rankine):

Let Q be the whole supply of water in cubic feet per second, of lifted to the height h above the pond, and $Q - q$ runs to waste at H below the pond; L, the length of the supply-pipe, from the po waste-clack; D, its diameter in feet; then

$$D = \sqrt{(1.63Q)}; \quad L = H + h + \frac{h}{H} \times 2 \text{ feet};$$

Volume of air vessel = volume of feed pipe;

$$\frac{qh}{(Q-q)H} = 1.12 - 0.2\sqrt{\frac{h}{H}} \text{ when } \frac{h}{H} \text{ does not exceed 20.}$$

$$\left(1+\frac{h}{10H}\right) \text{ nearly, when } \frac{h}{H} \text{ does not exceed 12.}$$

gives $\frac{q(H+h)}{QH} = 1.42 - .28\sqrt{\frac{h}{H}}.$

five sixths of the values given by D'Aubuisson's formula [illegible]

ft to fall. ...	4	6	8	10	12	14	16	18	20	22	24	26
per cent....	72	61	52	44	37	31	25	19	14	9	4	0

arpenter (*Eng'g Mechanics*, 1894) reports the results of four constructed by Rumsey & Co., Seneca Falls. The ram was connection for 1¼-inch supply and ½-inch discharge. The ed was 1½ inches in diameter, about 50 feet long, with 3 elbows, equivalent to about 65 feet of straight pipe, so far as resistned. Each run was made with a different stroke for the waste the supply and delivery head being constant; the object of t was to find that stroke of clack-valve which would give the ncy.

ke, per cent.	100	80	60	46
okes per minute.	52	56	61	66
eet of water	5.67	5.77	5.58	5.65
feet of water	19.75	19.75	19.75	19.75
mped, pounds	297	296	301	297.5
pplied, pounds	1615	1567	1518	1455.5
cent	64.9	66	74.9	70

y, 74.9, the highest realized, was obtained when the clack-valve tance equal to 60% of its full stroke, the full travel being 15/16

of Water Delivered by the Hydraulic Ram. ad Works.)—From 80 to 100 feet conveyance, one seventh of pring can be discharged at an elevation five times as high as ply the ram; or, one fourteenth can be raised and discharged as high as the fall applied.

e conveyed by a ram 3000 feet, and elevated 200 feet. The om 25 to 50 feet long.

g table gives the capacity of several sizes of rams, the dimenipes to be used, and the size of the spring or brook to which ted:

ity of Water nished per y the Spring ook to which e Ram is dapted.	Caliber of Pipes.		Weight of Pipe (Lead), if Wrought Iron, then of Ordinary Weight.		
	Drive.	Discharge.	Drive-pipe for head or fall not over 10 ft.	Discharge-pipe for not over 50 ft. rise.	Discharge-pipe for over 50 ft. and not exceeding 100 ft. in height.
s. per min.	inch.	inch.	per foot.	per foot.	per foot.
¾ to 2	¾	⅜	2 lbs.	10 ozs.	1 lb.
1½ " 4	1	½	3 "	12 "	1 " 4 ozs.
" 7	1¼	½	5 "	12 "	1 " 4 ozs
" 14	2	¾	8 "	1 lb. 4 "	2 "
" 25	2½	1	13 "	2 "	3 "
" 40	2½	1¼	13 "	3 "	4 "
" 75	4	2	21 "	7 "	8 "

HYDRAULIC-PRESSURE TRANSMISSION.

Water under high pressure (700 to 2000 lbs. per square inch and u affords a very satisfactory method of transmitting power to a d especially for the movement of heavy loads at small velocities, as b and elevators. The system consists usually of one or more pumps of developing the required pressure; accumulators, which are vertic ders with heavily-weighted plungers passing through stuffing-boxe upper end, by which a quantity of water may be accumulated at t sure to which the plunger is weighted; the distributing-pipes; and the cranes, or other machinery to be operated.

The earliest important use of hydraulic pressure probably was Bramah hydraulic press, patented in 1796. Sir W. G. Armstrong is one of the pioneers in the adaptation of the hydraulic system to cra use of the accumulator by Armstrong led to the extended use of hy machinery. Recent developments and applications of the system are due to Ralph Tweddell, of London, and Sir Joseph Whitworth. Sir Bessemer, in his patent of May 13, 1856, No. 1292, first suggested the hydraulic pressure for compressing steel ingots while in the fluid sta

The Gross Amount of Energy of the water under pressure in the accumulator, measured in foot-pounds, is its volume in cubic its pressure in pounds per square foot. The horse-power of a given steadily flowing is H.P. $= \frac{144pQ}{550} = .2618pQ$, in which Q is the quantity in cubic feet per second and p the pressure in pounds per square inc

The loss of energy due to velocity of flow in the pipe is calculated lows (R. G. Blaine, *Eng'g*, May 22 and June 5, 1891):

According to D'Arcy, every pound of water loses $\frac{\lambda 4L}{D}$ times its energy, or energy due to its velocity, in passing along a straight pip in length and D feet diameter, where λ is a variable coefficient. Fo cast-iron pipes it may be taken as $\lambda = .005\left(1 + \frac{1}{12D}\right)$, or for diam *inches* $= d$.

$d =$	½	1	2	3	4	5	6	7	8	9
$\lambda =$	.015	.01	.0075	.00667	.00625	.006	.00583	.00571	.00563	.00556

The loss of energy per minute is $60 \times 62.36Q \times \frac{\lambda 4L}{D}\frac{v^2}{2g}$, and the power wasted in the pipe is $W = \frac{.6363\lambda L(\text{H.P.})^2}{p^2 D^5}$, in which λ varies w diameter as above. $p =$ pressure at entrance in pounds per square Values of $.6363\lambda$ for different diameters of pipe in inches are:

$d =$	½	1	2	3	4	5	6	7	8	9
	.00954	.00636	.00477	.00424	.00398	.00382	.00371	.00363	.00358	.00353

Efficiency of Hydraulic Apparatus.—The useful effect direct hydraulic plunger or ram is usually taken at 93%. The follo given as the efficiency of a ram with chain-and-pulley multiplyin properly proportioned and well lubricated:

Multiplying....	2 to 1	4 to 1	6 to 1	8 to 1	10 to 1	12 to 1	14 to 1
Efficiency %....	80	76	72	67	63	59	54

With large sheaves, small steel pins, and wire rope for multiplyi the efficiency has been found as high as 66% for a multiplication of 20

Henry Adams gives the following formula for effective pressure in and hoists:

$P =$ accumulator pressure in pounds per square inch;
$m =$ ratio of multiplying power;
$E =$ effective pressure in pounds per square inch, including all allow for friction;

$$E = P(.84 - .02m).$$

J. E. Tuit (*Eng'g*, June 15, 1888) describes some experiments on the tion of hydraulic jacks from 3¼ to 13⅜-inch diameter, fitted with leather packings. The friction loss varied from 5.6% to 18.8% accor the condition of the leather, the distribution of the load on the ra The friction increased considerably with eccentric loads. With bein ing a plunger, 14 inch diameter, showed a friction loss of from 11.4% the load being c[illegible] from 15.0% to 7.5% with eccentric load, ntage of loss d[illegible] both cases with increase of load.

ckness of Hydraulic Cylinders.—From a table used by Sir Armstrong we take the following, for cast-iron cylinders, for an in- ressure of 1000 lbs. per square inch:

of cylinder, inches..	2	4	6	8	10	12	16	20	24
ess, inches.........	0.832	1.146	1.552	1.875	2.222	2.578	3.19	3.69	4.11

any other pressure multiply by the ratio of that pressure to 1000. figures correspond nearly to the formula $t = 0.175d + 0.48$, in which ckness and d = diameter in inches, up to 16 inches diameter, but for es diameter the addition 0.48 is reduced to 0.19 and at 24 inches it ears. For formulæ for thick cylinders see page 287, *ante*.

iron should not be used for pressures exceeding 2000 lbs. per square For higher pressures steel castings or forged steel should be used. rking pressures of 750 lbs. per square inch the test pressure should lbs. per square inch, and for 1500 lbs. the test pressure should not be an 3500 lbs.

ed of Hoisting by Hydraulic Pressure.—The maximum ble speed for warehouse cranes is 6 feet per second; for platform 4 feet per second; for passenger and wagon hoists, heavy loads, 2 er second. The maximum speed under any circumstances should exceed 10 feet per second.

Speed of Water Through Valves should never be greater 00 feet per second.

ed of Water Through Pipes.—Experiments on water at 1600 essure per square inch flowing into a flanging-machine ram, 20-inch ter, through a ½-inch pipe contracted at one point to ¼-inch, gave a y of 114 feet per second in the pipe, and 456 feet at the reduced sec- Through a ½-inch pipe reduced to ⅜-inch at one point the velocity 3 feet per second in the pipe and 381 feet at the reduced section. In a pipe without contraction the velocity was 355 feet per second.

many of the above notes the author is indebted to Mr. John Platt, ting engineer, of New York.

gh-pressure Hydraulic Presses in Iron-works are de- d by R. M. Daelen, of Germany, in Trans. A. I. M. E. 1892. The fol- distinct arrangements used in different systems of high-pressure ulic work are discussed and illustrated:

team-pump, with fly-wheel and accumulator.
team pump, without fly-wheel and with accumulator.
team-pump, without fly-wheel and without accumulator.

hese three systems the valve-motion of the working press is operated high-pressure column. This is avoided in the following:

ingle-acting steam-intensifier without accumulator.
team-pump with fly-wheel, without accumulator and with pipe-circuit.
team-pump with fly-wheel, without accumulator and without pipe- t.

disadvantages of accumulators are thus stated: The weighted plungers formerly served in most cases as accumulators, cause violent shocks pipe-line when changes take place in the movement of the water, t in many places, in order to avoid bursting from this cause, the pipes ade exclusively of forged and bored steel. The seats and cones of the lic valves are cut by the water (at high speed), and in such cases only ost careful maintenance can prevent great losses of power.

draulic Power in London.—The general principle involved mping water into mains laid in the streets, from which service-pipes arried into the houses to work lifts or three-cylinder motors when ry power is required. In some cases a small Pelton wheel has been working under a pressure of over 700 lbs. on the square inch. Over 55 of hydraulic mains are at present laid (1892).

reservoir of power consists of capacious accumulators, loaded to a ure of 800 lbs. per square inch, thus producing the same effect as if supply-tanks were placed at 1700 feet above the street-level. The is taken from the Thames or from wells, and all sediment is removed from by filtration before it reaches the main engine-pumps.

re are over 1750 machines at work, and the supply is about 6,500,000 s per week.

essential that the water used should be clean. The storage-tank ex- over the whole boiler-house and coal-store. The tank is divided, and ain amount of mud is deposited here. It then passes throu ondenser of the engines, and it is turned into a set of filt r. The body of the filter is a cast-iron cylinder, contain

granular filtering material resting upon a false bottom; under th
tributing arrangement, affording passage for the air, and under t
bottom of the tank. The dirty water is supplied to the filters fr
head tank. After passing through the filters the clean effluent
into the clean-water tank, from which the pumping-engines
supply. The cleaning of the filters, which is done at intervals of
effected so thoroughly *in situ* that the filtering material never re
removed.

The engine-house contains six sets of triple-expansion en
cylinders are 15-inch, 22-inch, 36 inch × 24-inch. Each cylind
single plunger-pump with a 5-inch ram, secured directly to the
the connecting-rod being double to clear the pump. The boile
150 lbs. on the square inch. Each pump will deliver 300 gallons
minute under a pressure of 800 lbs. to the square inch, the engi
about 61 revolutions per minute. This is a high velocity, com
heavy pressure; but the valves work silently and without percep

The consumption of steam is 14.1 pounds per horse per hour.

The water delivered from the main pumps passes into the ac
The rams are 20 inches in diameter, and have a stroke of 23 fee
each loaded with 110 tons of slag, contained in a wrought-iron
box suspended from a cross-head on the top of the ram.

One of the accumulators is loaded a little more heavily than t
that they rise and fall successively; the more heavily loaded act
valve on the main steam-pipe. If the engines supply more w
wanted, the lighter of the two rams first rises as far as it can g
then ascends, and when it has nearly reached the top, shuts of
checks the supply of water automatically.

The mains in the public streets are so constructed and laid a
fectly trustworthy and free from leakage.

Every pipe and valve used throughout the system is tested to
square inch before being placed on the ground and again tested
pressure in the trenches to insure the perfect tightness of the
jointing material used is gutta-percha.

The average rate obtained by the company is about 3 shillin
sand gallons. The principal use of the power is for intermittent w
where direct pressure can be employed, as, for instance, passeng
cranes, presses, warehouse hoists, etc.

An important use of the hydraulic power is its application t
guishing of fire by means of Greathead's injector hydrant. B
these hydrants a continuous fire-engine is available.

Hydraulic Riveting-machines.—Hydraulic riveting
duced in England by Mr. R. H. Tweddell. Fixed riveters were first
1868. Portable riveting-machines were introduced in 1872.

The riveting of the large steel plates in the Forth Bridge was do
portable machines working with a pressure of 1000 lbs. per squa
exceptional cases 3 tons per inch was used. (Proc. Inst. M. E., M

An application of hydraulic pressure invented by Andrew Hi
Liverpool, dispenses with the necessity of accumulators. It co
three-throw pump driven by shafting or worked by steam, an
partially upon the work accumulated in a heavy fly-wheel. The
passage from the pumps and back to them is in constant circ
very feeble pressure, requiring a minimum of power to preserve
water ready for action at the desired moment, when by the use
current is stopped from going back to the pumps, and is thrown
piston of the tool to be set in motion. The water is now confine
driving-belt or steam-engine, supplemented by the momentum of
fly-wheel, is employed in closing up the rivet, or bending or forg
ject subjected to its operation.

Hydraulic Forging.—In the production of heavy forg
cast ingots of mild steel it is essential that the mass of metal
operated on as equally as possible throughout its entire thickn
employing a steam-hammer for this purpose it has been found th
ternal surface of the ingot absorbs a large proportion of the sudd
of the blow, and that a comparatively small effect only is produ
central portions of the ingot, owing to the resistance offered by
of the mass to the rapid motion of the falling hammer—a disadv
is entirely overcome by the slow, though powerful, compre
hydraulic forging-press, ... appears destined to superse
hammer for the produ... ...ssive steel forgings.

forging-press the force-pump and the large or main cylinder re in direct and constant communication. There are no inter es of any kind, nor has the pump any clack-valves, but i' its cylinder full of water direct into the cylinder of the press, the same water, as it were, back again on the return stroke. oth cylinders and the pipe connecting them are full, the large press rises and falls simultaneously with each stroke of the ng up a continuous oscillating motion, the ram, of course, shorter distance, owing to the larger capacity of the press ournal Iron and Steel Institute, 1891. See also illustrated article Mechanism," page 668.)

complete illustrated account of the development of the hy- ng-press, see a paper by R. H. Tweddell in Proc. Inst. C. E., vol.

lic Forging-press.—A 2000-ton forging-press erected at orges in Belgium is described in *Eng. and M. Jour.*, Nov. 25, 1893. is composed essentially of two parts—the press itself and the The compressor is formed of a vertical steam-cylinder and a linder. The piston-rod of the former forms the piston of the hydraulic piston discharges the water into the press proper. tion is made by a cylindrical balanced valve; as soon as the eleased the steam-piston falls automatically under the action of ring its descent the steam passes to the other face of the piston cylinder, and finally escapes from the upper end.

m enters under the piston of the compressor-cylinder the pis- d its rod forces the water into the press proper. The pressure on the piston of the latter is transmitted through a cross-head ng which is upon the anvil. To raise the cross-head two small steam-cylinders are used, their piston-rods being connected to ad; steam acts only on the pistons of these cylinders from below. on of steam to the cylinders, which stand on top of the press gulated by the same lever which directs the motions of the com- e movement given to the dies is sufficient for all the ordinary forging.

30 blows per minute has been attained. A double press on the , having two compressors and giving a maximum pressure of s been erected in the Krupp works, at Essen.

en Intensifier. (*Iron Age*, Aug. 1890.)—The object of the o increase the pressure obtained by the ordinary accumulator essary to operate powerful hydraulic machines requiring very es, without increasing the pressure carried in the accumulator eral hydraulic system.

Intensifier consists of one outer stationary cylinder and one er which moves in the outer cylinder and on a fixed or stationary ger. When operated in connection with the hydraulic bloom- ethod of working is as follows: The inner cylinder having been ater and connected through the hollow plunger with the hydrau- of the shear, water at the ordinary accumulator-pressure is ad- the outer cylinder, which being four times the sectional area of gives a pressure in the inner cylinder and shear cylinder con- with of four times the accumulator-pressure—that is, if the ac- ressure is 500 lbs. per square inch the pressure in the intensifier lbs. per square inch.

lic Engine driving an Air-compressor and a hammer. (*Iron Age*, May 12, 1892.)—The great hammer in Rome, is one of the largest in existence. Its falling weight 100 tons, and the foundation belonging to it consists of a block of 1000 tons. The stroke is 16 feet 4¾ inches; the diameter of 6 feet 3⅛ inches; diameter of piston-rod 13¾ inches; total height ner, 62 feet 4 inches. The power to work the hammer, as well as nes of 100 and 150 tons respectively, and other auxiliary appli- ging to it, is furnished by four air-compressors coupled together directly by water-pressure engines, by means of which the air is to 73.5 *pounds per square inch*. The cylinders of the water- *gines, which are provided with* a bronze lining, have a 13¾-inch *troke is 47¾ inches*, with a pressure of water on the piston *264.6 pounds per square inch*. The compressors are ... *diameter, and have* 47¾-inch stroke. Each of the... *power equal to 280* horse-power. The compress...

[illegible]

FUEL.

Theory of Combustion of Solid Fuel [illegible]

[illegible]

If the disengaged carbon is cooled below [illegible] fore coming in contact with oxygen, it [illegible] smoke, and when deposited on solid bodies, soot.

But if the disengaged carbon is maintained at [illegible] and supplied with oxygen sufficient for its [illegible]

eat of Combustion of Fuels. (Rankine.)—The follow-
ws the total heat of combustion with oxygen of one pound of substances named in it, in British thermal units, and also in evaporated from 212°. It also shows the weight of oxygen re-
mbine with each pound of the combustible and the weight of y in order to supply that oxygen. The quantities of heat are authority of MM. Favre and Silbermann.

ombustible.	Lbs. Oxygen per lb. Combustible.	Lb. Air (about).	Total British Heat-units.	Evaporative Power from 212° F., lbs.
as..................	8	36	62,032	64.2
erfectly burned so as arbonic oxide..........	1⅓	6	4,400	4.55
ectly burned so as to onic acid,............	2⅔	12	14,500	15.0
, 1 lb.......	3 3/7	15 3/7	21,344	22.1
id hydrocarbons, 1 lb.	....	......	from 21,700 to 19,000	from 22½ to 20
ide, as much as is made perfect combustion of rbon, viz., 2⅓ lbs......	1⅓	6	10,000	10.45

rfect combustion of carbon, making carbonic oxide, produces e third of the heat which is yielded by the complete combustion.
heat of combustion of any compound of hydrogen and carbon e sum of the quantities of heat which the constituents would pro-
ately by their combustion. (Marsh-gas is an exception.)
ting the total heat of combustion of compounds containing oxy-
ll as hydrogen and carbon, the following principle is to be When hydrogen and oxygen exist in a compound in the proper to form water (that is, by weight one part of hydrogen to eight , these constituents have no effect on the total heat of combus-
drogen exists in a greater proportion, only the surplus of hydro-
hat which is required by the oxygen is to be taken into account.
wing is a general formula (Dulong's) for the total heat of combus-
compound of carbon, hydrogen, and oxygen:
and O be the fractions of one pound of the compound, which pectively of carbon, hydrogen, and oxygen, the remainder being sh, and other impurities. Let h be the total heat of combustion nd of the compound in British thermal units. Then

$$h = 14{,}500\left\{C + 4.28\left(H - \frac{O}{8}\right)\right\}.$$

wing table shows the composition of those compounds which are nce, either as furnishing oxygen for combustion, as entering into ition, or as being produced by the combustion of fuel:

Names.	Symbol of Chemical Composition.	Proportions of Element by Weight.	Chemical Equivalent by Weight.	Proportions of Elements by Volume.
....	......	N 77 + O 23	100	N 79 + O 21
....	H_2O	H 2 + O 16	18	H 2 + O
....	NH_3	H 3 + N 14	17	H 3 + N
xide......................	CO	C 12 + O 16	28	C + O
cid.......................	CO_2	C 12 + O 32	44	C + O 2
....	CH_2	C 12 + H 2	[illegible]	[illegible] H 2
fire-damp................	CH_4	C 12 + H 4	[illegible]	[illegible] H 4
cid	SO_2	S 32 + O 32	[illegible]	[illegible]
hydrogen..................	SH_2	S 32 + H 2	[illegible]	[illegible]
arbon.....................	S_2C	S 64 + C 12	[illegible]	[illegible]

Since each lb. of C requires $2\frac{2}{3}$ lbs. of O to burn it to CO_2, and air contains 23% of O, by weight, $2\frac{2}{3} \div 0.23$ or 11.6 lbs. of air are required to burn 1 lb. of C.

Analyses of Gases of Combustion.—The following are selected from a large number of analyses of gases from locomotive boilers, to show the range of composition under different circumstances (P. H. Dudley, Trans. A. I. M. E., iv. 250):

Test.	CO_2	CO	O	N	
1	13.8	2.5	2.5	81.6	No smoke visible.
2	11.5	....	6	82.5	Old fire, escaping gas white, engine working hard.
3	8.5	...	8	83	Fresh fire, much black gas, " " "
4	2.3	....	17.2	80.5	Old fire, damper closed, engine standing still.
5	5.7	....	14.7	79.6	" " smoke white, engine working hard.
6	8.4	1.2	8.4	82	New fire, engine not working hard.
7	12	1	4.4	82.6	Smoke black, engine not working hard.
8	3.4	...	16.8	76.8	" dark, blower on, engine standing still.
9	6	....	13.5	81.5	" white, engine working hard.

In analyses on the Cleveland and Pittsburgh road, in every instance when the smoke was the blackest, there was found the greatest percentage of unconsumed oxygen in the product, showing that something besides the mere presence for oxygen is required to effect the combustion of the volatile carbon of fuels.

J. C. Hoadley (Trans. A. S. M. E., vi. 749) found as the mean of a great number of analyses of flue gases from a boiler using anthracite coal:

CO_2, 13.10; CO, 0.30; O, 11.94; N, 74.66.

The loss of heat due to burning C to CO instead of to CO_2 was 2.18%. The surplus oxygen averaged 113.3% of the O required for the C of the fuel, the average for different weeks ranging from 88.6% to 137%.

Analyses made to determine the CO produced by excessively rapid firing gave results from 2.54% to 4.81% CO and 5.12 to 8.01% CO_2; the ratio of C in the CO to total carbon burned being from 43.80% to 48.55%, and the number of pounds of air supplied to the furnace per pound of coal being from 33.2 to 19.3 lbs. The loss due to burning C to CO was from 27.84% to 30.86 of the full power of the coal.

Temperature of the Fire. (Rankine, S. E., p. 283.)—By temperature of the fire is meant the temperature of the products of combustion at the instant that the combustion is complete. The elevation of that temperature above the temperature at which the air and the fuel are supplied to the furnace may be computed by dividing the total heat of combustion of one lb. of fuel by the weight and by the mean specific heat of the whole products of combustion, and of the air employed for their dilution under constant pressure. The specific heat under constant pressure of these products is about as follows:

Carbonic-acid gas, 0.217; steam, 0.475; nitrogen (probably), 0.245; air, 0.238; ashes, probably about 0.200. Using these data, the following results are obtained for pure carbon and for olefiant gas burned, respectively, first, in just sufficient air, theoretically, for their combustion, and, second, when an equal amount of air is supplied in addition for dilution.

Fuel.	Products undiluted.		Products diluted.	
	Carbon.	Olefiant Gas.	Carbon.	Olefiant Gas.
Total heat of combustion, per lb...	14,500	21,300	14,500	21,300
Wt. of products of combustion, lbs.	13	16.43	25	31.86
Their mean specific heat........	0.237	0.257	0.238	0.248
Specific heat × weight...........	3.08	4.22	5.94	7.9
Elevation of temperature, F.......	4580°	5050°	2440°	2710°

[The above calculations are made on the assumption that the specific heats of the gases are constant, but they probably increase with the increase of temperature (see Specific Heat), in which case the temperature would be less than those above given. The temperature would be further he heat rendered latent by the conversion into steam of any t in the fuel.]

Temperature in Combustion of Gases. (*Eng'g*, d April 2, 1886.)—It is found that the temperatures obtained nt fall short of those obtained by calculation. Three theo- een given to account for this: 1. The cooling effect of the containing vessel; 2. The retardation of the evolution of heat ssociation; 3. The increase of the specific heat of the gases at mperatures. The calculated temperatures are obtainable only tion that the gases shall combine instantaneously and simulta- ughout their whole mass. This condition is practically impos- riments. The gases formed at the beginning of an explosion maining combustible gases and tend to retard or check the of the remainder.

CLASSIFICATION OF SOLID FUELS.

ssifies solid fuels as follows (*Eng'g and M'g Jour.*, July, 1874):

Name of Fuel.	Ratio $\frac{O}{H}$ or $\frac{O+N}{H}$*.	Proportion of Coke or Charcoal yielded by the Dry Pure Fuel.
se........................	8	0.28 @ 0.30
ose and encasing matter)....	7	.30 @ .35
sil fuel	6 @ 5	.35 @ .40
brown coal.................	5	.40 @ .50
coals	4 @ 1	.50 @ .90
....	1 @ 0.75	.90 @ .92

inous coals he divides into five classes as below:

Type.	Elementary Composition. C.	H.	O.	Ratio $\frac{O}{H}$ or $\frac{O+N}{H}$*.	Proportion of Coke yielded by Distillation.	Nature and Appearance of Coke.
ing dry	75@80	5.5@4.5	19.5@15	4@3	0.50@.60	Pulverulent.
ing fat g coals, oals,	80@85	5.8@5	14.2@10	3@2	.60@.68	Melted, but friable.
at coals, ksmiths'	84@89	5 @4.5	11 @5.5	2@1	.68@.74	Melted; somewhat compact.
ning [illegible]t g coals. coals,	88@91	5.5@4.[illegible]	6.5@5.5	1	.74@.82	Melted; very compact.
anthra- ,	90@93	4.5@4	5.5@3	1	.82@.90	Pulverulent.

itrogen rarely exceeds 1 per cent of the weight of the fuel.
[illegible]luding bituminous lignites, which resemble petroleums.

ives the following: The extreme differences in the chemical and properties of different kinds of coal are very great. The f free carbon ranges from 30 to 93 per cent; that of hydrocar- ous kinds from 5 to 58 per cent; that of water, or oxygen and the proportions which form water, from an inappreciably r to 27 per cent; that of ash, from 1½ to 26 per cent.
ns varieties of coal may be divided into principal cl[illegible] thracite coal; 2, semi-bituminous coal; 3, bitumin[illegible] or cannel coal; 5, lignite or brown coal.

Diminution of H and O in Series from Wood to Ant

(Groves and Thorp's Chemical Technology, vol. i., Fuels, p.

Substance.	Carbon.	Hydrogen.
Woody fibre.	52.65	5.25
Peat from Vulcaire	59.57	5.96
Lignite from Cologne	66.04	5.27
Earthy brown coal	73.18	5.98
Coal from Belestat, secondary	75.06	5.84
Coal from Rive de Gier	89.29	5.05
Anthracite, Mayenne, transition formation	91.58	3.96

Progressive Change from Wood to Graphit

(J. S. Newberry in Johnson's Cyclopedia.)

	Wood.	Loss.	Lignite.	Loss.	Bituminous coal.	Loss.	Anthracite.	Loss
Carbon	49.1	18.65	30.45	12.35	18.10	3.57	14.53	1.42
Hydrogen	6.3	3.25	3.05	1.85	1.20	0.93	0.27	0.14
Oxygen	44.6	24.40	20.20	18.13	2.07	1.32	0.65	0.65
	100.0	46.30	53.70	32.33	21.37	5.82	15.45	2.21

Classification of Coals, as Anthracite, Bituminou Prof. Persifer Frazer (Trans. A. I. M. E., vi. 430) proposes a c tion of coals according to their "fuel ratio," that is, the ratio the f bon bears to the volatile hydrocarbon.

In arranging coals under this classification, the accidental impuri as sulphur, earthy matter, and moisture, are disregarded, and the stituents alone are considered.

	Carbon Ratio.	Fixed Carbon.	Vola Hydro
I. Hard dry anthracite.	100 to 12	100. to 92.31%	0. t
II. Semi-anthracite	12 to 8	92.31 to 88.89	7.69 t
III. Semi-bituminous.	8 to 5	88.89 to 83.33	11.11 t
IV. Bituminous	5 to 0	83.33 to 0.	16.67 t

It appears to the author that the above classification does not d line at the proper point between the semi-bituminous and the bit coals, viz., at a ratio of C ÷ V.H.C. = 5, or fixed carbon 83.33%, vol drocarbon 16.67%, since it would throw many of the steam coals of C and Somerset counties, Penn., and the Cumberland, Md., and Poc Va., coals, which are practically of one class, and properly t semi-bituminous coals, into the bituminous class. The dividing tween the semi-anthracite and semi-bituminous coals, C ÷ V.H would place several coals known as semi-anthracite in the semi-bit class. The following is proposed by the author as a better classific

	Carbon Ratio.	Fixed Carbon.	Vol.
I. Hard dry anthracite..	100 to 12	100 to 92.31%	0 t
II. Semi-anthracite	12 to 7	92.31 to 87.5	7.69 t
III. Semi-bituminous	7 to 3	87.5 to 75	12.5 t
IV. Bituminous	3 to 0	75 to 0	25 t

Rhode Island Graphitic Anthracite.—A peculiar gr found at Cranston, near Providence, R. I. It resembles both graph anthracite coal, and has about the following composition (A. E. Hun A. I. M. E., xvii., 678): Graphitic carbon, 78%; volatile matter, 2.6 15.06%; phosphorus, .045%. It burns with extreme difficulty.

ANALYSES OF COALS.

Composition of Pennsylvania Anthracites. (Tra M. E., xiv., 706.)—Samples weighing 100 to 200 lbs. were collected fr of 100 to 200 tons as shipped to market, and reduced by proper me labora[illegible] Thirty-three samples were analyzed by McCre ing re[illegible]ws. They show the mean character of the coal of t impo[illegible]s in the Northern field in the vicinity of Wilke the [illegible] (Lehigh) field in the vicinity of Hazleton, a field in the vicinity of Shenandoah, and in the Southern field between Chunk and Tamaqua.

Bed.	Name of Field.	Water.	Volatile Matter.	Fixed Carbon.	Ash.	Sulphur.	Vol. Matter. Per cent of total combustible.	Ratio, C ÷ V.H.C.
on...	E. Middle	3.71	3.08	86.40	6.22	.58	3.44	28.07
oth..	E. Middle	4.12	3.08	86.38	5.92	.49	3.45	27.99
ose ..	W. Middle	3.54	3.72	81.59	10.65	.50	4.36	21.93
oth .	W. Middle	3.16	3.72	81.14	11.08	.90	4.38	21.83
ose F	Southern	3.01	4.13	87.98	4.38	.50	4.48	21.32
Mtn..	W. Middle	3.04	3.95	82.66	9.88	.46	4.56	20.93
Foot	W. Middle	3.41	3.98	80.87	11.23	.51	4.69	20.32
oth .	Southern	3.09	4.28	83.81	8.18	.64	4.85	19.62
oth .	Northern	3.42	4.38	83.27	8.20	.73	5.00	19.00
l Bed	Loyalsock	1.30	8.10	83.34	6.23	1.03	8.86	10.29

above analyses were made of coals of all sizes (mixed). When coal is ed into sizes for shipment the purity of the different sizes as regards aries greatly. Samples from one mine gave results as follows:

Name of Coal.	Screened Through inches.	Screened Over inches.	Analyses. Fixed Carbon.	Analyses. Ash.
g	2.5	1.75	88.49	5.66
ove	1.75	1.25	83.67	10.17
estnut......	1.25	.75	80.72	12.67
a..........	.75	.50	79.05	14.66
ckwheat...	.50	.25	76.92	16.62

Bernice Basin, Pa., Coals.

	Water.	Vol. H.C.	Fixed C.	Ash.	Sulphur.
e Basin, Fullivan and oming Cos.; range of 8..	0.96 to 1.97	3.56 to 8.56	82.52 to 89.39	3.27 to 9.34	0.24 to 1.04

coal is on the dividing-line between the anthracites and semi-anthra- and is similar to the coal of the Lykens Valley district.
e recent analyses (Trans. A. I. M. E., xiv. 721) give:

	Water.	Vol. H.C.	Fixed Carb.	Ash.	Sulphur.
ng seam.......	0 65	9.40	83.69	5.34	0.91
below seam....	3.67	15.42	71.34	8.97	0.59

rst is a semi-anthracite, the second a semi-bituminous.

ace Occupied by Anthracite Coal. (J. C. I. W., vol. iii.)—The contents of 2240 lbs. of hard Lehigh coal is a little over 36 feet; an ge Schuylkill W. A., 37 to 38 feet; Shamokin, 38 to 39 feet; Lorberry, y 41.

ording to measurements made with Wilkesbarre anthracite coal from yoming Valley, it requires 32.2 cu. ft. of lump, 33.9 cu. ft. broken, u. ft. egg, 34.8 cu. ft. of stove, 35.7 cu. ft. of chestnut, and 36.7 cu. ft. a, to make one ton of coal of 2240 lbs.; while it requires 28.8 cu. ft. of 30.3 cu. ft. of broken, 30.8 cu. ft. of egg, 31.1 cu. ft. of stove, 31.9 cu. chestnut, and 32.8 cu. ft. of pea, to make one ton of 2000 lbs.

mposition of Anthracite and Semi-bituminous Coals. s. A. I. M. E., vi. 430.)—Hard dry anthracites, 16 analyses by Rogers, a range from 94.10 to 82.47 fixed carbon, 1.40 to 9.53 volatile matter, 50 to 8.00 ash, water, and impurities. Of the fuel constituents alone, xed carbon ranges from 98.53 to 89.63, and the volatile matter from 1.47 37, the corresponding carbon ratios, or C ÷ Vol. H.C. being from 67.02 4.

i-anthracites.—12 analyses by Rogers show a range of from 90.23 t fixed carbon, 7.07 to 13.75 volatile matter, and 2.20 ... ash mpurities. *Excluding the ash, etc., the range of f ... 42, and the volatile combustible* 7.27 to 15.58, the c... *eing from 12.75 to 5.41.*

Semi-bituminous Coals.—10 analyses of Penna. and Maryland coals fixed carbon 68.41 to 84.80, volatile matter 11.2 to 17.28, and ash, water, impurities 4 to 13.99. The percentage of the fuel constituents is fixed ca 79.84 to 88.80, volatile combustible 11.20 to 20.16, and the carbon ratio 11. 3.96.

American Semi-bituminous and Bituminous Coals

(Selected chiefly from various papers in Trans. A. I. M. E.)

	Moisture.	Vol. Hydro-carbon.	Fixed Carbon	Ash.
Penna. Semi-bituminous ;				
Broad Top, extremes of 5	.79	13.84	78.46	6.00
	.78	17.38	76.14	4.81
Somerset Co., extremes of 5	1.27	14.33	77.77	6.63
	1.89	18.51	65.90	10.62
Blair Co., average of 5	1.07	26.72	60.77	9.45
Cambria Co., average of 7, lower bed, B.	0.74	21.21	68.94	7.51
Cambria Co., 1, upper bed, C.	1.14	17.18	73.42	6.58
Cambria Co., South Fork, 1	....	15.51	78.60	5.84
Centre Co., 1	0.60	22.60	68.71	5.40
Clearfield Co., average of 9, upper bed, C.	0.70	23.94	69.28	4.62
Clearfield Co., average of 8, lower bed, D.	0.81	21.10	74.08	3.86
Clearfield Co., range of 17 anal.	0.41 to 1.94	20.09 to 25.19	66.69 to 74.02	2.65 to 7.65
Bituminous :				
Jefferson Co., average of 26	1.21	32.53	60.99	3.76
Clarion Co., average of 7	1.97	38.60	54.15	4.10
Armstrong Co., 1	1.18	42.55	49.69	4.58
Connellsville Coal	1.26	30.10	59.61	8.23
Coke from Conn'ville (Standard)	.49	0.01	87.46	11.32
Youghiogheny Coal	1.03	36.49	59.05	2.61
Pittsburgh, Ocean Mine	.28	39.09	57.33	3.30

The percentage of volatile matter in the Kittaning lower bed B and Freeport lower bed D increases with great uniformity from east to west; th

		Volatile Matter.	Fixed Carbon
Clearfield Co, bed	D	20.09 to 25.19	68.73 to 74.7
" " "	B	22.56 to 26.13	64.37 to 69.6
Clarion Co., "	B	35.70 to 42.55	47.51 to 55.4
" "	D	37.15 to 40.80	51.39 to 56.3

Connellsville Coal and Coke. (Trans. A. I. M. E., xiii. 33 The Connellsville coal-field, in the southwestern part of Pennsylvania strip about 3 miles wide and 60 miles in length. The mine workings confined to the Pittsburgh seam, which here has its best development size, and its quality best adapted to coke-making. It generally aff from 7 to 8 feet of coal.

The following analyses by T. T. Morrell show about its range of compo tion :

	Moisture.	Vol. Mat.	Fixed C.	Ash.	Sulphur.	Phosph
Herold Mine	1.26	28.83	60.79	8.44	.67	.0[illegible]
Kintz Mine	0.79	31.91	56.46	9.52	1.32	[illegible]

In comparing the composition of coals across the Appalachian field, in the *western section* of Pennsylvania, it will be noted that the Connellsv *variety occupies* a peculiar position between the rather dry semi-bitum *coals eastward* of it and the fat bituminous coals flanking it on the we *beneath* the Connellsville or Pittsburgh coal-bed occurs an inter to 600 feet of "barren measures," separating it from th coal measures of Western Pennsylvania. The followi imilarity in composition in the coals of these upper and res in the same geographical belt or basin.

om the Upper Coal-measures (Penna.) in a Westward Order.

	Moisture.	Vol. Mat.	Fixed Carb.	Ash.	Sulphur.
....	1.35	3.45	89.06	5.81	0.30
.....	0.89	15.52	74.28	9.29	0.71
....	1.66	22.35	68.77	5.96	1.24
.....	....	31.38	60.30	7.24	1.09
....	1.02	33.50	61.34	3.28	0.86
.....	1.41	37.66	54.44	5.86	0.64

m the Lower Coal-measures in a Westward Order.

	Moisture.	Vol. Mat.	Fixed Carb.	Ash.	Sulphur.
....	1.35	3.45	89.06	5.81	0 30
....	0.77	18.18	73.34	6.69	1.02
. ..	1.40	27.23	61.84	6.93	2.60
....	1.18	16.54	74.46	5.96	1.86
....	0.92	24.36	62.22	7.69	4.92
....	0.96	38.20	52.03	5.14	3.66

ia and Ohio Bituminous Coals. Variation of Coals of the same Beds in different Dis-

0 analyses in the reports of the Pennsylvania Geological wing are selected. They are divided into different groups. analysis in each group is given, ash and other impurities and the percentage in 100 of combustible matter being

	No. of Analyses	Fixed Carbon	Vol. H. C.	Carbon Ratio.
l-bed, upper bench... ...	5			
ship, Greene Co.....		59.72	40.28	1.48
ship, Washington Co....		53.22	46.78	1.13
l-bed, lower bench.......	9			
ip, Greene Co.....		60.69	39.31	1.54
, Washington Co........		54.31	45.69	1.19
ed..	3			
Greene Co...............		64.39	35.61	1.80
eek, Greene Co.		60.35	39.65	1.52
ed:				
Washington Co...........		60.87	39.13	1.65
		59.11	40.89	1.20
" "	5	63.54	36.46	1.74
		50.97	49.03	1.04
eene Cc..................	3	61.80	38.20	1.61
		54.33	45.67	1.19
ashington Co., average .		66.44	33.56	1.98
reene Co.......... ...	1	57.83	42.17	1.37
mi-bituminous (showing l. mat. to the eastward).	8	79.73	20.27	3 93
		75.47	24.53	3.07
....	7			
eorgetown...............		40.68	59.32	0.68
Georgetown..............		62.57	37.43	1.66
OHIO.				
ed in Ohio:				
hio.....................		61.45	38.55	1.59
hio....		63.46	36.54	1.73
		66 14	33.86	1.95
io....................		63.46	36.54	1.73
		64.93	35.07	1 85
....		60.92	39.08	
		62.33	37 .67	

Analyses of Southern and Western Coal

	Moisture.	Vol. Mat.	Fixed C.
OHIO.			
Hocking Valley.	5.00	32.80	53.15
	7.40	29.20	60.45
MARYLAND.			
Cumberland..................	95	19.13	72.70
	1.23	15.47	73.51
VIRGINIA.			
South of James River, 23 analyses, range	from 0.67	27.28	46.70
	to 2.46	38.60	67.83
Average of 23............	1.48	32.24	58.89
North of James River, eastern outcrop,	0.40	18.60	71.00
	1.79	23.96	59.98
Carbonite or Natural Coke....	1.57	9.64	79.93
	1.56	14.26	81.61
Western outcrop, 11 analyses, range	from	21.33	54.97
	to ..	30.50	70.80
Average of 11............		26.06	63.75
Pocahontas Flat-top* (Castner & Curran's Circular)	0.52	23.90	74.20
	0.62	18.48	75.22
WEST VIRGINIA (New River.)			
Quinnimont,† 3 analyses	from 0.76	17.57	75.89
	to 0.94	18.19	79.40
Nuttalburgh †.................	0.84	29.59	69.00
	1.35	25.35	70.67
VIRGINIA and KENTUCKY.			
Big Stone Gap Field,‡ 9 analyses, range	from 0.80	31.44	54.80
	to 2.01	36.27	63.50
KENTUCKY.			
Pulaski Co., 3 analyses, range	from 1.26	35.15	60.85
	to 1.32	39.44	52.48
Muhlenberg Co., 4 analyses, range	from 3.60	30.60	58.80
	to 7.06	38.70	53.70
Kentucky Cannel Coals,§ 5 analyses, range	from	40.20‖	59.80 coke
	to	63.30‖	33.70 coke
TENNESSEE.			
Scott Co., Range of several.¶..	from 70	32.33	46.61
	to 1.83	41.29	61.66
Roane Co., Rockwood.	1.75	26.62	60.11
Hamilton Co., Melville........	2.74	26.50	67.08
Marion Co., Etna	94	23.72	63.94
Sewanee Co., Tracy City......	1.60	29.30	61.00
Kelly Co., Whiteside..........	1.30	21.80	74.20
GEORGIA.			
Dade Co......................	1.20	23.05	60.50
ALABAMA.			
Warren Field:			
Jefferson Co., Birmingham..	3.01	42.76	48.30
" " Black Creek ..	.12	26.11	71.64
Tuscaloosa Co............	1.59	38.33	54.64
Cahaba Field, Bibb Co ..... Helena Vein.	2.00	32.90	53.98
Coke Vein....	1.78	30.60	66.58

* Analyses of Pocahontas Coal by John Pattinson, F.C.S., 189

	C.	H.	O.	N.	S.	Ash.	Water.	C
Lumps...	86.51	4.44	4.95	0.66	0.61	1.54	1.29	7
Small ...	83.13	4.29	5.33	0.66	0.56	4.63	1.40	7

Calorific value, by Thomson's Calorimeter: Lumps = 15.4 lb *evaporated* from and at 212°; small = 14.7 lbs.

† *These* coals are coked in beehive ovens, and yield from 63% to

‡ *This field* covers about 120 square miles in Virginia, and about *miles in Kentucky.*

§ *The principal* use of the cannel coals is for enriching

‖ Volatile matter including moisture.

¶ ...es from Morgan, Rhea, Anderson, and

ALABAMA COALS. (W. B. Phillips, *Eng. & M. J.*, June 3, 1893.)

of ...	Location.	Proximate. Vol. and Combust. Matter.	Proximate. Fixed Carbon.	Ultimate. Carbon.	Ultimate. Hydrogen.	Ultimate. Oxygen.	Ultimate. Nitrogen.	Ultimate. Sulphur.	Ultimate. Ash.	Ultimate. Moisture.
orth	Helena	34.30	60.50	73.23	7.98	11.92	1.07	0.60	3.50	1.70
....	Pratt mines..	33.45	63.20	75.82	10.52	7.51	1.73	1.07	2.00	1.35
ood	Brookwood..	27.80	58.70	72.47	10.38	1.60	0.40	1.65	11.90	1.60
ock.	Blocton	34.80	60.60	72.75	8.61	11.12	1.48	1.44	2.65	1.95
ood	"	35.65	57.30	70.82	10.19	9.95	1.31	0.68	5.25	1.80
....	Pratt mines..	31.55	64.95	75.05	9.91	8.95	1.62	0.97	2.35	1.15
....	Brookwood..	30.50	66.30	73.96	10.50	9.57	1.62	1.15	2.20	1.00
....	Blue Creek ..	25.80	69.90	72.68	10.77	9.83	1.39	1.03	2.80	1.50
....	Coalburg	32.55	65.57	74.59	10.58	9.48	1.31	1.32	1.90	0.82
....	....	30.15	52.90	60.37	10.70	9.00	1.26	1.72	16.30	0.65

	Moisture.	Vol. Mat.	Fixed C.	Ash.	Sulphur.
TEXAS.					
dine	3.54	30.84	50.69	14.93	......
Field, Vein I...........	1.91	20.04	62.71	15.35	......
" " II...........	1.37	16.42	68.18	13.02	......
" " III...........	0.84	29.35	50.18	19.63	......
" " IV...	0.45	21.6	45.75	29.1	3.15
INDIANA.					
oal, average.*.	2.10	37.35	57.95	2.60	......
" Lafayette	13.05	32.34	48.78	5.81	......
" Sand Creek†..... ...	4.50	91.00		4.50	.....
ILLINOIS.‡					
le......................	8.22	39.40	43.95	8.43	.....
or	7.20	38.88	45.30	8.60	.....
le......................	11.00	32.55	53.00	3.65	.....
....	5.78	48.70	45.37	6.15	......
n...	8.45	34.99	44.50	12.06	......
y	10.80	27.32	44.78	17.10	......
ndale....................	6.36	26.40	59.84	7.40	......
oin......................	8.86	23.54	60.60	7.00	......
rbon.............	6.12	24.68	66.50	2.70	......
on...............	6.27	57.11	26.30	10.32	.....

iana Block Coal (J. S. Alexander, Trans. A. I. M. E., iv. 100).—The block coal of the Brazil (Indiana) district differs in chemical com- n but little from the coking coals of Western Pennsylvania. The al difference, however, is quite marked; the latter has a cuboid struc- ade up of bituminous particles lying against each other, so that under tion of heat fusion throughout the mass readily takes place, while coal is formed of alternate layers of rich bituminous matter and a al-like substance, which is not only very slow of combustion, but so s the transmission of heat that agglutination is prevented, and the urns away layer by layer, retaining its form until consumed.

alysis by E. T. Cox: C, 72.94; H, 4.50; O, 11.77; N, 1.79; ash, 4.50; ure, 4.50.

Illinois coals are extremely variable in character. The above anal- *e given in D. L. Barnes's paper on* "American Locomotive Practice," *S. C. E. 1893*, except the last, the Staunton coal, which is by Hunt *(Trans. A. S. M. E., v. 266)*. The Staunton coal is remar- *rcentage of volatile* matter, but it is excelled in this

Nixon's Navigation Welsh Coal is remarkably pure, and tains not more than 3 to 4 per cent of ashes, giving 88 per cent of har lustrous coke. The quantity of fixed carbon it contains would clas among the dry coals, but on account of its coke and its intensity of bustion it belongs to the class of fat, or long-flaming coals.

Chemical analysis gave the following results: Carbon, 90.27; hydr 4.39; sulphur, .69; nitrogen, .49; oxygen (difference), 4.16.

The analysis showed the following composition of the volatile parts bon, 22.53; hydrogen, 34.96; O + N + S, 42.51.

The heat of combustion was found to be, as a result of several ex ments, 8864 calories for the unit of weight. Calculated according composition, the heat of combustion would be 8805 calories = 15,849 thermal units per pound.

This coal is generally used in trial-trips of steam-vessels in Great Br

Sampling Coal for Analysis.—J. P. Kimball, Trans. A. I. xii. 317, says: The unsuitable sampling of a coal-seam, or the imp preparation of the sample in the laboratory, often gives rise to errors terminations of the ash so wide in range as to vitiate the analysis practical purposes; every other single determination, excepting mo showing its relative part of the error. The determination of sulphu ash are especially liable to error, as they are intimately associated slates.

Wm. Forsyth, in his paper on The Heating Value of Western Coals *News*, Jan. 17, 1895), says: This trouble in getting a fairly average sam anthracite coal has compelled the Reading R. R. Co., in getting their sa to take as much as 300 lbs. for one sample, drawn direct from the chut it stands ready for shipment.

The directions for collecting samples of coal for analysis at the C., B. laboratory are as follows:

Two samples should be taken, one marked "average," the other "sel Each sample should contain about 10 lbs., made up of lumps about th of an orange taken from different parts of the dump or car, and so sel that they shall represent as nearly as possible, first, the average lot; sec the best coal.

An example of the difference between an "average" and a "sel sample, taken from Mr. Forsyth's paper, is the following of an Illinois

	Moisture.	Vol. Mat.	Fixed Carbon.	Ash.
Average	1.36	27.69	35.41	35.54
Select............	1.90	34.70	48.23	15.17

The theoretical evaporative power of the former was 9.13 lbs. of w from and at 212° per lb. of coal, and that of the latter 11.44 lbs.

Relative Value of Fine Sizes of Anthracite.—For bur on a grate coal-dust is commercially valueless, the finest commercial thracites being sold at the following rates per ton at the mines, acco to a recent address by Mr. Eckley B. Coxe (1893):

Size.	Range of Size.	Price at Mines
Chestnut....................	1¼ to ⅞ inch	$2.75
Pea	⅞ to 9/16	1.25
Buckwheat..................	9/16 to ⅜	0.75
Rice..............	⅜ to 3/16	0.25
Barley..................	3/16 to 2/32	0.10

But when coal is reduced to an impalpable dust, a method of burni becomes possible to which even the finest of these sizes is wholly dapted; the coal may be blown in as dust, mixed with its proper propo of air, and no grate at all is then required.

Pressed Fuel. (E. F. Loiseau, Trans. A. I. M. E., viii. 314.)—Pre fuel has been made from anthracite dust by mixing the dust with ten cent of its bulk of dry pitch, which is prepared by separating from tar temperature of 572° F. the volatile matter it contains. The mixture is heated by steam to 212°, at which temperature the pitch acquires its menting properties, and is passed between two rollers, on the periph which are milled out a series of semi-oval cavities. The lumps of the ture, about the size of an egg, drop out under the rollers on an endless which carries them to a screen in eight minutes, which time is suffic cool the lumps, and they are then ready for delivery.

The enterprise of making the pressed fuel above described was not mercially successful, on account of the low price of other coal. In Fra *[illegible] briquettes*" are regularly made of coal-dust (bituminous semi[illegible])

VE VALUE OF STEAM COALS.

a coal may be determined, with more or less approx-
three different methods.
lysis ; 2d. by combustion in a coal calorimeter ; 3d,
m-boiler. The first two methods give what may be
eating value, the third gives the practical value.
first two methods depends on the precision of the
calorimetry adopted, and upon the care and skill of
sults of the third method are subject to numerous
nd error, and may be taken as approximately true
conditions under which the test is made. Analysis
with considerable accuracy the heating value which
the conditions of perfect combustion and complete
produced. A boiler test gives the actual result under
ess imperfect combustion, and of numerous and va-
give the highest practical heating value, if the condi-
aft, extent of heating surface, method of firing, etc.,
or the particular coal tested, and it may give results
if these conditions are adverse or unsuitable to the

tests being so extremely variable, their use for the
g the relative steaming values of different coals has
conclusions. A notable instance is found in the
n's tests, made in 1844, the only extensive series of
ever made. He reported the steaming value of the
ion Co.'s coal to be far the lowest of all the anthra-
easily explained by an examination of the conditions
the test, which were entirely unsuited to that coal.
ult for Pittsburgh coal which is far beneath that now
practice, his low result being chiefly due to the use

Proposed Apparatus for Determining the Heating
als (Trans. A. I. M. E., xiv. 727) the author described
ratus designed to test fuel on a large scale, avoiding
oiler test. It consists of a fire-brick furnace enclosed
two cylindrical shells containing a great number of
unded by cooling water and through which the gases
hile being cooled. No steam is generated in the ap-
assed through it and allowed to escape at a tempera-
e product of the weight of the water passed through
crease in temperature is the measure of the heating

difference of opinion concerning the value of chemi-
s of approximating the heating power of coal. It
-Kestner and Meunier-Dollfus, in their extensive series
pe in 1868, that the heating power as determined by
greater than that given to chemical analysis accord-

n Paris by M. Mahler, however, show a much closer
and calorimetric tests. A brief description of these
the French, may be found in an article by the author
y, vol. i. page 97.
expressed by the formula,

$$\text{itish Thermal Units} = 14{,}500C + 62{,}500\left(H - \frac{O}{8}\right),^*$$

re respectively the percentage of carbon, hydrogen,
led by 100. A study of M. Mahler's calorimetric tests
um difference between the results of these tests and
power by Dulong's law in any single case is only a
esults of 31 tests show that Dulong's formula gives an
hermal units less than the calorimetric tests, the
value being over 14,000 thermal units, a difference of

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ng's formula with Berthelot's figure for the heating
tish thermal units,

$$r = 14{,}650C + 62{,}025\left(H - \frac{(O + N) - 1}{8}\right).$$

Mahler's calorimetric apparatus consists of a strong steel ves "bomb" immersed in water, proper precaution being taken to preven ation. One gram of the coal to be tested is placed in a platinum boat this bomb, oxygen gas is introduced under a pressure of 20 to 25 atmosp and the coal ignited explosively by an electric spark. Combustion plete and instantaneous, the heat is radiated into the surrounding weighing 2200 grams, and its quantity is determined by the rise in tem ture of this water, due corrections being made for the heat capacity apparatus itself. The accuracy of the apparatus is remarkable, du tests giving results varying only about 2 parts in 1000.

The close agreement of the results of calorimetric tests when pr conducted, and of the heating power calculated from chemical analys dicates that either the chemical or the calorimetric method may cepted as correct enough for all practical purposes for determining th heating power of coal. The results obtained by either method m taken as a standard by which the results of a boiler test are to be pared, and the difference between the total heating power, and the re the boiler test is a measure of the inefficiency of the boiler under th ditions of any particular test.

In practice with good anthracite coal, in a steam-boiler properly p tioned, and with all conditions favorable, it is possible to obtain steam 80% of the total heat of combustion of the coal. This result was obtained in the tests at the Centennial Exhibition in 1876, in five dif boilers. An efficiency of 70% to 75% may easily be obtained in regular tice. With bituminous coals it is difficult to obtain as close an appro the theoretical maximum of economy, for the reason that some of th tile combustible portion of the coal escapes unburned, the difficulty in ing rapidly as the content of volatile matter increases beyond 3% most coals of the Western States it is with difficulty that as much as 65% of the theoretical efficiency can be obtained without the use of ga ducers.

The chemical analysis heretofore referred to is the ultimate analys the percentage of carbon, hydrogen, and oxygen of the dry coal. It is f however, from a study of Mahler's tests that the proximate analysis, gives fixed carbon, volatile matter, moisture, and ash, may be relied giving a measure of the heating value with a limit of error of only abo After deducting the moisture and ash, and calculating the fixed carbo percentage of the coal dry and free from ash, the author has constructe following table:

APPROXIMATE HEATING VALUE OF COALS.

Percentage F. C. in Coal Dry and Free from Ash.	Heating Value B.T.U. per lb. Comb'le.	Equiv. Water Evap. from and at 212° per lb. Combustible.	Percentage F. C. in Coal Dry and Free from Ash.	Heating Value B.T.U. per lb. Comb'le.	Equiv. W Evap. f and at per l Combust
100	14500	15.00	68	15480	16.[illegible]
97	14760	15.28	63	15120	15.[illegible]
94	15120	15.65	60	14580	15.[illegible]
90	15480	16.03	57	14040	14.[illegible]
87	15660	16.21	54	13320	13.[illegible]
80	15840	16.40	51	12600	13.[illegible]
72	15660	16.21	50	12240	12.[illegible]

Below 50% the law of decrease of heating-power shown in the table a ently does not hold, as some cannel coals and lignites show much h heating-power than would be predicted from their chemical constitutio

The use of this table may be shown as follows:

Given a coal containing moisture 2%, ash 8%, fixed carbon 61%, and vol matter 29%, what is its probable heating value? Deducting moisture ash we find the fixed carbon is 61/90 or 68% of the total of fixed carbo *volatile matter*. One pound of the coal dry and free from ash would, b *table, have* a heating value of 15,480 thermal units, but as the ash and m *ure, having* no heating value, are 10% of the total weight of the coal would *have* 90% of the table value, or 13,932 thermal units. This by 965, the latent heat of steam at 212° gives an equivalent evapora lb. of coal [illegible] ıe that can be obtained in practice from this coal would efficiency of the boiler, and this largely upon the difficulty ning its volatile combustible matter in the boiler furnace. cy of 65% could be obtained, then the evaporation per lb. of 212° would be $14.42 \times .65 = 9.37$ lbs.

ıthracite coal, in which the combustible portion is, say, 97% 3% volatile matter, the highest result that can be expected th all conditions favorable is 12.2 lbs. of water evaporated ɔer lb. of combustible, which is 80% of 15.28 lbs. the theower. With the best semi-bituminous coals, such as Cumıontas, in which the fixed carbon is 80% of the total comor 76% of the theoretical 16.4 lbs., may be obtained. For vith a fixed carbon ratio of 68%, 11 lbs., or 69% of the theos about the best practically obtainable with the best boilers. Ohio coals, with a fixed carbon ratio of 60%, 10 lbs., or 66% 15.09 lbs., has been obtained, under favorable conditions, arch over the furnace. With coals mined west of Ohio, ratios, the boiler efficiency is not apt to be as high as 60%. res a table of probable maximum boiler-test results from fixed carbon ratios may be constructed as follows:

o..............	97	80	68	60	54	50
t 212° per lb. combustible, maximum in boiler-tests:						
	12.2	12.5	11	10	8.3	7.0
per cent.......	80	76	69	66	60	55
diation, imperfect combustion, etc:						
	20	24	31	34	40	45

ɔetween the loss of 20% with anthracite and the greater her coals is chiefly due to imperfect combustion of the the more highly volatile coals sending up the chimney the of smoke and unburned hydrocarbon gases. It is a measure of the boiler furnace and of the inefficiency of heatingy the deposition of soot, the latter being primarily caused ion of the ordinary furnace and its unsuitability to the f bituminous coal. If in a boiler-test with an ordinary furs are obtained than those in the above table, it is an indicaıble conditions, such as bad firing, wrong proportions of lraft, and the like, which are remediable. Higher results ɔnly with gas-producers, or other styles of furnace esper smokeless combustion.

rnace Adapted for Different Coals. (From the ı "The Evaporative Power of Bituminous Coals," Trans. 57.)—Almost any kind of a furnace will be found well ıg anthracite coals and semi-bituminous coals containing volatile matter. Probably the best furnace for burning contain between 20% and 40% volatile matter, including the Welsh, Nova Scotia, and the Pittsburgh and Monongahela lain grate-bar furnace with a fire-brick arch thrown over of keeping the combustion-chamber thoroughly hot. The oals containing over 40% volatile matter will be a furnace e-brick with a large combustion-chamber, and some speintroducing very hot air to the gases distilled from the ly, a separate gas-producer and combustion-chamber, with ing both air and gas before they unite in the combustionıaracter of furnace to be especially avoid d in burning all containing over 20% of volatile matter is the ordinary furboiler is set directly above the grate bars, and in which the of the boiler are directly exposed to radiation from the The question of admitting air above the grate is still union *Engineer* recently said: "All our experience, extending goes to show that when the production of smoke is predevices for admitting air, either there is an increase in the u-l or a diminution in the production of steam. * * * The nter yet devised is a good fireman."

draught Furnaces.—Recent experiments show that *coal considerable* saving may be made by causing the *wards from the freshly-fired* coal through the hot coal *lar good results are also* obtained by the upward draught *coal under the bed of* hot coal instead of on top. (See

Calorimetric Tests of American Coals.—From a [illegible] tests of American and foreign coals, made with an oxygen [illegible] Geo. H. Barrus (Trans. A. S. M. E., vol. xiv. 816), the following [illegible] showing the range of variation:

	Percentage of Ash.	Total Heat of Combustion. B. T. U.
Semi-bituminous.		
George's Cr'k, Cumberl'd, Md., 10 tests	6.1	14,2[illegible]
	8.6	12,8[illegible]
Pocahontas, Va., 5 tests	3.2	14,0[illegible]
	6.2	13,6[illegible]
New River, Va., 6 tests	3.5	13,9[illegible]
	5.7	13,8[illegible]
Elk Garden, Va., 1 test	7.8	13,180
Welsh, 1 test	7.7	13,3[illegible]
Bituminous.		
Youghiogheny, Pa., lump	5.9	12,9[illegible]
" " slack	10.2	11,664
Frontenac, Kansas	17.7	10,506
Cape Breton, (Caledonia)	8.7	12,429
Lancashire, Eng	6.8	12,122
Anthracite, 11 tests	10.5	11,5[illegible]
	9.1	13,189

Evaporative Power of Bituminous Co[illegible]

(Tests with Babcock & Wilcox Boilers, Trans. A. S. M. E.

Name of Coal.	Duration of Test.	Grate Surface, sq. ft.	Heating Surface, sq. ft.	Percentage of Refuse.	Coal burned per sq. ft. of Grate, pounds,	Water evaporated per sq. ft. of Heating Surface per hour, pounds,	Water per pound Coal from and at 212°, lbs.
1. Welsh	13⅛ hrs	40	1079	7.5	6.3	2.07	11.53
2. Anthracite scr's 1/5 Powelton, Pa., Semi-bit. 4/5,	10¼ h	60	3126	8.8	17.6	4.32	11.32
3. Pittsbg'h fine slack	4 hrs	33.7	1679	12.3	21.9	4.47	8.12
" 3d Pool lump	10 "	43.5	2760	4.8	27.5	4.76	10.47
4. Castle Shannon, nr Pittsb'gh, ⅜ nut, ⅝ lump,	42¼ h	69.1	4784	10.5	27.9	4.13	10.60
5. Ill. "run of mine"	6 days.	....	1196	....	....	1.41	9.49
" Ind. block, "very good"	3 d'ys	....	1196	....	....	2.95	9.47
6. Jackson, O., nut	8 hrs.	48	3358	9.6	32.1	4.11	8.93
" Staunton, Ill., nut	8 "	60	3358	17.7	25.1	2.27	5.09
7. Renton screenings.	5 h 50 m	21.2	1564	13.8	31.5	2.95	6.88
" Wellington scr'gs	6 h 30 m	21.2	1561	18.3	27	2.93	7.89
" Black Diam. scr'gs	5 h 58 m	21.2	1564	19.3	36.4	3.11	6.29
" Seattle screenings.	6 h 24 m	21.2	1564	13.4	31.3	2.91	6.86
" Wellington lump	6 h 19 m	21.2	1564	13.8	28.2	3.52	9.02
" Cardiff lump	6 h 47 m	21.2	1564	11.7	26.7	3.69	10.07
" " [illegible]	[illegible]3 m	21.2	1564	19.1	25.6	3.35	9.62
[illegible]outh P[illegible]	[illegible] m	21.2	1564	13.9	28.9	3.53	8.96
[illegible]eattle [illegible]	[illegible] m	21.2	1564	9.5	34.1	3.57	7.98

: 1. London, England; 2. Peacedale, R. I.; 3. Cincinnati, O.; h, Pa.; 5. Chicago, Ill.; 6. Springfield, O.; 7. San Francisco,

ove tests the furnace was supplied with a fire-brick arch for radiation of heat from the coal directly to the boiler.

ng of Coal. (I. P. Kimball, Trans. A. I. M. E., viii. 204.)—effect of the weathering of coal, while sometimes increasing eight, is to diminish the quantity of carbon and disposable to increase the quantity of oxygen and of indisposable hy-e a reduction in the calorific value.

pyrites in coal tends to produce rapid oxidation and mechan-tion of the mass, with development of heat, loss of coking ontaneous ignition.

preciable results of the weathering of anthracite within the of exposure of stocked coal are confined to the oxidation of pyrites. In coking coals, however, weathering reduces and s the coking power, while the pyrites are converted from the hide into comparatively innocuous sulphates.

nd that at a temperature of 158° to 180° Fahr., three coals lost ys an average of 3.6% of calorific power. (See also paper by , Trans. A. I. M. E., iv. 55.)

COKE.

olid material left after evaporating the volatile ingredients of means of partial combustion in furnaces called coke ovens, on in the retorts of gas-works.

n ovens is preferred to gas coke as fuel. It is of a dark-gray htly metallic lustre, porous, brittle, and hard.

on of coke yielded by a given weight of coal is very different inds of coal, ranging from 0.9 to 0.35.

porous texture, it readily attracts and retains water from the nd sometimes, if it is kept without proper shelter, from 0.15 to s weight consists of moisture.

Analyses of Coke.

eport of John R. Procter, Kentucky Geological Survey.)

Where Made.	Fixed Carbon	Ash.	Sulphur.
Pa. (Average of 3 samples)........	88.96	9.74	0.810
Tenn. " " 4 "	80.51	16.34	1.595
Ala. " " 4 "	87.29	10.54	1.195
a. " " 3 "	92.53	5.74	0.597
. Va. " " 8 "	92.38	7.21	0.562
. Ky. " " 7 "	93.23	5.69	0.749

eriments in Coking. CONNELLSVILLE REGION.
(John Fulton, *Amer. Mfr.*, Feb. 10, 1893.)

Coal Charged.	Ash made.	Fine Coke made.	Market Coke made	Total Coke made.	Per cent of Yield.				Per Cent Lost.
					Ash.	Fine Coke.	Market Coke	Total Coke.	
lb.	lb.	lb.	lb.	lb.					
2,420	99	385	7,518	7,903	00.80	3 10	60.53	63.63	35.57
1,090	90	359	6,580	6,939	00.81	3.24	59.33	62.57	36.62
9,120	77	272	5,418	5.690	00.84	2.98	59.41	62.39	36 77
9,020	74	349	5,334	5,683	00.82	3 87	59.13	63.00	36.18
1,650	340	1365	24,850	26,215	00.82	3.28	59.66	62.94	36.24

s show, in a general average, that Connellsville coal [illegible] dern beehive oven will yield 66.17% of marketable [illegible] *or braize, and 0.82% of ash.*

Mining Jour., March 25, 1893.)

Recovery of By-products in Coke Manufac
many considerable progress has been made in the recovery
The Hoffman-Otto oven has been most largely used, its p
being that it is connected with regenerators. In 1884 4
system were running, and in 1892 the number had increased

A Hoffman-Otto oven in Westphalia takes a charge of 6¼
and converts it into coke in 48 hours. The product of an
1025 tons in the Ruhr district, 1170 tons in Silesia, and 960 ton
trict. The yield from dry coal is 75% to 77% of coke, 2.5% to 3
to 1.2% of sulphate of ammonia in the Ruhr district; 65% to
4.5% of tar, and 1% to 1.25% of sulphate of ammonia in the Upp
and 68% to 72% of coke, 4% to 4.3% of tar and 1.8% to 1.9% of sulp
in the Saar district. A group of 60 Hoffman ovens, therefore
the following:

District.	Coke, tons.	Ta ton
Ruhr	51,200	186
Upper Silesia	48,000	300
Saar	40,500	240

An oven which has been introduced lately into German
with the recovery of by-products is the Semet-Solvay, whi
than the Hoffman-Otto, and for this reason 73% to 77% of
mixed with 23% to 27% of coal low in volatile matter, and
coke. Mixtures of this kind yield a larger percentage of c
other hand, the amount of gas is lessened, and therefore the
ammonia is not so great.

In the manufacture of coke from soft coal in retort ovens
those constructed so as to save the by-products formed in
ations, the coke has the disadvantage of being more por
more easily crushed cell-walls than when the same coal i
ordinary beehive-oven.

References: F. W. Luerman, Verein Deutscher Eisenhu
Iron Age, March 31, 1892; *Amer. Mfr.*, April 28, 1893. An
of articles on the manufacture of coke, by John Fulton, of
is published in the *Colliery Engineer*, beginning in January,

Making Hard Coke.—J. J. Fronheiser and C. S. Pri
bria Iron Co., Johnstown, Pa., have made an improvement
facture by which coke of any degree of hardness may be tu
accomplished by first grinding the coal to a coarse powde
with a hydrate of lime (air or water slacked caustic li

n the coke is very light, 38, 36, and 33 lbs. are regarded as a bushel. from 42 to 50 lbs. are given as the weight of a bushel ; in this case ould be quite heavy.

cts of the Distillation of Coal.—S. P. Sadler's Handbook al Organic Chemistry gives a diagram showing over 50 chemical hat are derived from distillation of coal. The first derivatives are as-liquor, coal-tar, and coke. From the gas-liquor are derived nd sulphate, chloride and carbonate of ammonia. The coal-tar into oils lighter than water or crude naphtha, oils heavier than erwise dead oil or tar, commonly called creosote,—and pitch. wo former are derived a variety of chemical products.

e coal-tar there comes an almost endless chain of known combina- e greatest industry based upon their use is the manufacture of he enormous extent to which this has grown can be judged from at there are over 600 different coal-tar colors in use, and many more et are too expensive for this purpose. Many medicinal prepara- e from the series, pitch for paving purposes, and chemicals for rapher, the rubber manufacturers and tanners, as well as for timber and cloths.

position of the hydrocarbons in a soft coal is uncertain and quite but the ultimate analysis of the average coal shows that it ap- quite nearly to the composition of CH_4 (marsh-gas). (W. H. Trans. A. I. M. E., xx. 625.)

WOOD AS FUEL.

hen newly felled, contains a proportion of moisture which varies n in different kinds and in different specimens, ranging between %, and being on an average about 40%. After 8 or 12 months' ordi- ng in the air the proportion of moisture is from 20 to 25%. This dryness, or almost perfect dryness if required, can be produced days' drying in an oven supplied with air at about 240° F. When ke is used as the fuel for that oven, 1 lb. of fuel suffices to expel s. of moisture from the wood. This is the result of experiments e scale by Mr. J. R. Napier. If air-dried wood were used as e oven, from 2 to 2½ lbs. of wood would probably be required to he same effect.

cific gravity of different kinds of wood ranges from 0.3 to 1.2.

y dry wood contains about 50% of carbon, the remainder consisting tirely of oxygen and hydrogen in the proportions which form e coniferous family contain a small quantity of turpentine, which ocarbon. The proportion of ash in wood is from 1% to 5%. The of combustion of all kinds of wood, when dry, is almost ex- same, and is that due to the 50% of carbon.

ve is from Rankine; but according to the table by S. P. Sharpless . I. W., iv. 36, the ash varies from 0.03% to 1.20% in American woods. uel value, instead of being the same for all woods, ranges from white oak) to 5546 calories (for long-leaf pine) = 6600 to 9883 British nits for dry wood, the fuel value of 0.50 lbs. carbon being 7272

ng Value of Wood.—The following table is given in several eference, authority and quality of coal referred to not stated. ght of one cord of different woods (thoroughly air-dried) is about :

r hard maple....	4500 lbs.	equal to 1800 lbs. coal.	(Others give 2000.)
....	3850 "	" 1540 " "	(" 1715.)
l and black oak..	3250 "	" 1300 " "	(" 1450.)
estnut, and elm..	2350 "	" 940 " "	(" 1050.)
ge pine..........	2000 "	" 800 " "	(" 925.)

g to the figures in the last column, it is said :

e above it is safe to assume that 2¼ lbs. of dry wood are equal to ge quality of soft coal and that the full value of the same weight t woods is very nearly the same—that is, a pound of hickory is more for fuel than a pound of pine, assuming both to be dry. It nt that the wood be dry, as each 10% of water or moisture in wood t about 12% from its value as fuel.

an average wood of the analysis C 51%, H 6.5%, O 42.0%, ash 0.5%, dry, its fuel *value per pound, according* to Dulong's formula, V =

$\left[14{,}500\,C + 62{,}000\left(H - \frac{O}{8}\right)\right]$, is 8170 British thermal units. If the w
ordinarily dried in air, contains 25% of moisture, then the heating va
pound of such wood is three quarters of 8170 = 6127 heat-units.
heat required to heat and evaporate the ¼ lb. of water from the atmo
temperature, and to heat the steam made from this water to the te
ture of the chimney gases, say 150 heat-units per pound to heat the
212°, 966 units to evaporate it at that temperature, and 100 heat
raise the temperature of the steam to 420° F., or 1216 in all = 304 fo
which subtracted from the 6127, leaves 5824 heat units as the net fu
of the wood per pound, or about 0.4 that of a pound of carbon.

Composition of Wood.

(Analysis of Woods, by M. Eugene Chevandier.)

Woods.	Composition.			
	Carbon.	Hydrogen.	Oxygen.	Nitrogen.
Beech	49.36%	6.01%	42.69%	0.91%
Oak	49.64	5.92	41.16	1.29
Birch	50.20	6.20	41.62	1.15
Poplar	49.37	6.21	41.60	0.96
Willow	49.96	5.96	39.56	0.96
Average	49.70%	6.06%	41.30%	1.05%

The following table, prepared by M. Violette, shows the propo
water expelled from wood at gradually increasing temperatures:

Temperature.	Water Expelled from 100 Parts of W		
	Oak.	Ash.	Elm.
257° Fahr	15.26	14.78	15.32
302° Fahr	17.93	16.19	17.02
347° Fahr	32.13	21.22	36.91
392° Fahr	35.80	27.51	33.38
437° Fahr	44.31	33.38	40.56

The wood operated upon had been kept in store during two years.
wood which has been strongly dried by means of artificial heat is
posed to the atmosphere, it reabsorbs about as much water as it c
in its air-dried state.

A cord of wood = 4 × 4 × 8 = 128 cu. ft. About 56% solid wood
interstitial spaces. (Marcus Bull, Phila., 1829. J. C. I. W., vol. i. p.
B. E. Fernow gives the per cent of solid wood in a cord as determi
cially in Prussia (J. C. I. W., vol. iii. p. 20):

Timber cords, 74.07% = 80 cu. ft. per cord;
Firewood cords (over 6″ diam.), 69.44% = 75 cu. ft. per cord;
"Billet" cords (over 3″ diam.), 55.55% = 60 cu. ft. per cord;
"Brush" woods less than 3″ diam., 18.52%; Roots, 37.00%.

CHARCOAL.

Charcoal is made by evaporating the volatile constituents of w
peat, either by a partial combustion of a conical heap of the materia
charred, covered with a layer of earth, or by the combustion of a
portion of fuel in a furnace, in which are placed retorts containing
terial to be charged.

According to Peclet, 100 parts by weight of wood when charred in
yield from 17 to 22 parts by weight of charcoal, and when charr
from 28 to 30 parts.

... ordinary condition of the wood ...
... 100 consist of moisture. Of the re
... one half, or 37½% of the gross w
... n an average nearly half of the

uring the partial combustion in a heap, and about one quarter stillation in a retort.

parts by weight of wood in a retort, 12½ parts of wood must the furnace. Hence in this process the whole expenditure of uce from 28 to 30 parts of charcoal is 112½ parts; so that if the arcoal obtained is compared with the whole weight of wood amount is from 25% to 27%; and the proportion lost is on an ÷ 37½ = 0.3, nearly.

to Peclet, good wood charcoal contains about 0.07 of its weight proportion of ash in peat charcoal is very variable, and is es- n average at about 0.18. (Rankine.)

mation concerning charcoal may be found in the Journal of the n Workers' Assn., vols. i. to vi. From this source the following een taken:

f Charcoal from a Cord of Wood.—From 45 to 50 e cord in the kiln, and from 30 to 35 in the meiler. Prof. Egles- s. A. I. M. E., viii. 395, says the yield from kilns in the Lake region is often from 50 to 60 bushels for hard wood and 50 for he average is about 50 bushels.

ent yield per cord depends largely upon whether the cord is a 128 cu. ft. or not.

months' test of a kiln at Goodrich, Tenn., Dr. H. M. Pierce found ollows: Dimensions of kiln—inside diameter of base, 28 ft. 8 in.; ing of arch, 26 ft. 8 in.; height of walls, 8 ft.; rise of arch, 5 ft.; cords. Highest yield of charcoal per cord of wood (measured) s, lowest 50.14 bushels, average 53.65 bushels.

arges 12, length of each turn or period from one charging to days. (J. C. I. W., vol. vi. p. 26.)

from Different Methods of Charcoal-making.

g Methods.	Character of Wood used.	Yield. In Volume per cent.	Yield. In Weight per cent.	Bushels of Charcoal per Cord of Wood.	Weight in Lbs. per Bushel of Charcoal.
a's experiments	Birch dried at 230 F.......	....	35.9	......	......
retorts, fuel ex-	Air dry, av. good yellow pine weighing abt. 28 lbs. per cu. ft.	77.0	28.3	63.4	15.7
retorts, fuel in-		65.8	24.2	54.2	15.7
vens, av. results	Good dry fir and pine, mixed.	81.0	27.7	66.7	13.3
vens, av. results	Poor wood, mixed fir and pine	70.0	25 8	62.0	13.3
meilers excep-	Fir and white-pine wood, mixed. Av. 25 lbs. per cu. ft.	72.2	24 7	59.5	13.3
eilers, av. results		52.5	18 3	43.9	13.3
kilns, av. results	Av. good yellow pine weighing abt. 25 lbs. per cu. ft.	54.7	22.0	45.0	17 5
meilers, av. re- .... -		42.9	17.1	35 0	17.5

mption of Charcoal in Blast-furnaces per Ton of n; average consumption according to census of 1880, 1.14 tons per ton of pig. The consumption at the best furnaces is much average. As low as 0 853 ton, is recorded of the Morgan furnace; e, 0.858; Elk Rapids, 0.884. (1892.)

ption of Water and of Gases by Charcoal.—Svedlius, l-book for charcoal-burners, prepared for the Swedish Govern- s: Fresh charcoal, *also reheated charcoal*, contains scarcely but *when cooled it absorbs it very rapidly* ... that after *r hours, it may contain 4% to 8% of water.* ... lapse of a *the moisture of* charcoal may not increase ... and may *t at 10% to 15%, or an average of 12%.* ... charred *coal ought, then, to contain about 84* ... parts *ash, and 1 part hydrogen.*

M. Saussure, operating with blocks of fine boxwood charco[illegible] burnt, found that by simply placing such blocks in contact wi[illegible] gases they absorbed them in the following proportion:

	Volumes.		
Ammonia	90.00	Carbonic oxide	[illegible]
Hydrochloric acid gas	85.00	Oxygen	[illegible]
Sulphurous acid	65.00	Nitrogen	[illegible]
Sulphuretted hydrogen	55.00	Carburetted hydrogen	[illegible]
Nitrous oxide (laughing-gas)	40.00	Hydrogen	[illegible]
Carbonic acid	35.00		

It is this enormous absorptive power that renders of so muc[illegible] comparatively slight sprinkling of charcoal over dead animal ma[illegible] preventive of the escape of odors arising from decomposition.

In a box or case containing one cubic foot of charcoal may [illegible] without mechanical compression a little over nine cubic feet o[illegible] representing a mechanical pressure of one hundred and twenty-[illegible] to the square inch. From the store thus preserved the oxyg[illegible] drawn by a small hand-pump.

Composition of Charcoal Produced at Various Te[illegible] tures. (By M. Violette.)

	Temperature of Carbonization.		Composition of the Solid Produc[illegible]			
			Carbon.	Hydrogen.	Oxygen.	Nitrogen and Loss.
	Cent.	Fahr.	Per cent.	Per cent.	Per cent.	Per cent.
1	150°	302°	47.51	6.12	46.29	0.08
2	200	392	51.82	3.99	43.98	0.23
3	250	482	65.59	4.81	28.97	0.63
4	300	592	73.24	4.25	21.96	0.57
5	350	662	76.64	4.14	18.44	0.61
6	432	810	81.64	4.96	15.24	1.61
7	1023	1873	81.97	2.30	14.15	1.60

The wood experimented on was that of black alder, or alder b[illegible] which furnishes a charcoal suitable for gunpowder. It was p[illegible] dried at 150 deg. C. = 302 deg. F.

MISCELLANEOUS SOLID FUELS.

Dust Fuel—Dust Explosions.—Dust when mixed in air b[illegible] such extreme rapidity as in some cases to cause explosions. Exp[illegible] flour-mills have been attributed to ignition of the dust in confined [illegible] Experiments in England in 1876 on the effect of coal-dust in carrying [illegible] mines showed that in a dusty passage the flame from a blown-out [illegible] travel 50 yards. Prof. F. A. Abel (Trans. A. I. M. E., xiii. 260) says t[illegible] dust in mines much promotes and extends explosions, and that it m[illegible] ily be brought into operation as a fiercely burning agent which w[illegible] flame rapidly as far as its mixture with air extends, and will opera[illegible] explosive agent though the medium of a very small proportion of f[illegible] in the air of the mine. The explosive violence of the combustion o[illegible] largely due to the instantaneous heating and consequent expansio[illegible] air. (See also paper on "Coal Dust as an Explosive Agent," by [illegible] Raymond, Trans. A. I. M. E. 1894.) Experiments made in German[illegible] show that pulverized fuel may be burned without smoke, and w[illegible] economy. The fuel, instead of being introduced into the fire-bo[illegible] ordinary [illegible]er, is first reduced to a powder by pulverisers of [illegible] stru[illegible] place of the ordinary boiler fire-box there is a co[illegible] cha[illegible] m of a closed furnace lined with fire-brick and wi[illegible] similar in construction to those used in ob[illegible] [illegible] throws a constant stream of the fuel into t[illegible] [illegible] ated that it scatters the powder through[illegible] e fire-box. When this powder is once ignited, and it is very by first raising the lining to a high temperature by an open bustion continues in an intense and regular manner under the current of air which carries it in. (*Mfrs. Record*, April, 1893.) fuel was used in the Crompton rotary puddling-furnace at rsenal, England, in 1873. (Jour. I. & S. I., i. 1873, p. 91.)

Turf, as usually dried in the air, contains from 25% to 30% of must be allowed for in estimating its heat of combustion. This g been evaporated, the analysis of M. Regnault gives, in 100 ectly dry peat of the best quality: C 58%, H 6%, O 31%, Ash 5%. amples of peat the quantity of ash is greater, amounting to 7% es to 11%.

c gravity of peat in its ordinary state is about 0.4 or 0.5. It can ed by machinery to a much greater density. (Rankine.)

m-engine, i. 61) gives as the average composition of dried Irish H 6%, O 30%, N 1.25%, Ash 4%.

Dulong's formula to this analysis, we obtain for the heating value dry peat 10,260 heat-units per pound, and for air-dried peat con- f moisture, after making allowance for evaporating the water, its per pound.

t as Fuel.—The heating power of sawdust is naturally the und as that of the wood from which it is derived, but if allowed is more like spent tan (which see below). The conditions neces- ning sawdust are that plenty of room should be given it in the l sufficient air supplied on the surface of the mass. The same havings, refuse lumber, etc. Sawdust is frequently burned in c., by being blown into the furnace by a fan-blast.

nanure has been successfully used as fuel by the Cable Rail- Chicago. It was mixed with soft coal and burned in an ordinary ided with a fire-brick arch.

n Bark as Fuel.—Tan, or oak bark, after having been used sses of tanning, is burned as fuel. The spent tan consists of the ion of the bark. According to M. Peclet, five parts of oak bark r parts of dry tan; and the heating power of perfectly dry tan, 5% of ash, is 6100 English units; whilst that of tan in an ordinary ness, containing 30% of water, is only 4284 English units. The ater evaporated from and at 212° by one pound of tan, equiva- heating powers, is, for perfectly dry tan, 5.46 lbs., for tan with 3.84 lbs. Experiments by Prof. R. H. Thurston (Jour. Frank. gave with the Crockett furnace, the wet tan containing 59% of aporation from and at 212° F. of 4.24 lbs. of water per pound an, and with the Thompson furnace an evaporation of 3.19 lbs. f wet tan containing 55% of water. The Thompson furnace con- fire-brick ovens, each 9 feet × 4 feet 4 inches, containing 234 of grate in all, for three boilers with a total heating surface of feet, a ratio of heating to grate surface of 9 to 1. The tan was holes in the top. The Crockett furnace was an ordinary fire- e, 6 × 4 feet, built in front of the boiler, instead of under it, the ting surface to grate being 14.6 to 1. According to Prof. Thurs- itions of success in burning wet fuel are the surrounding of the pletely with heated surfaces and with burning fuel that it may ried, and then so arranging the apparatus that thorough com- be secured, and that the rapidity of combustion be precisely never exceed the rapidity of desiccation. Where this rapidity on is exceeded the dry portion is consumed completely, leaving d mass of fuel which refuses to take fire.

s Fuel. (*Eng'g Mechanics*, Feb., 1893, p. 55.)—Experiments in ed that winter-wheat straw, dried at 230° F., had the following : C, 46.1; H, 5.6; N, 0.42; O, 43.7; Ash, 4.1. Heating value in mal units: dry straw, 6290; with 6% water, 5770; with 10% water, straws of other grains the heating value of dry straw ranged r buckwheat to 6750 for flax.

E., vol. 1, p. 62) gives the *mean composition of wheat and barley 36; H. 5; O. 38; O, 0.50;* Ash, 4.75; water, 15.75, th *than 1%. The heating value of straw of this compo g's formula, and deducting the heat lost in ev heat units. Clark erroneously gives it as* 8144 h

s Fuel in Sugar Manufacture.—Baga *sugar-cane, after the juice has been* extracte generated for every pound of carbon consumed, the 66% of generating 297,834 heat units as against 345,200, or a units in favor of the 72% bagasse.

"Assuming the temperature of the waste gases to be surrounding atmosphere and water in the bagasse at 8 tity of air necessary for the combustion of one pound the lost heat will be as follows: In the waste gases, heat 450° F., and in vaporizing the moisture, etc., the 66% b 112,546 heat units, and 116,150 for the 72% bagasse.

"Subtracting these quantities from the above, we find will produce 185,288 available heat units, or nearly 38 bagasse, which gives 299,050 units. Accordingly, one ton at 66% mill extraction will produce 680 lbs. bagasse, equal able heat units, while the same cane at 72% extraction w bagasse, equal to 167,468,000 units.

"A similar calculation for the case of Louisiana cane co fibre, and 16% total solids in the juice, assuming 75% mill that bagasse from one ton of cane contains 157,395,64 which 56,146,500 have to be deducted.

"This would make such bagasse worth on an average per ton of cane ground. Under fairly good conditions, orate 7½ lbs. water, while the best boiler plants evaporate the bagasse from 1 ton of cane at 75% mill extraction sho 689 lbs. to 919 lbs. of water. The juice extracted from su der these conditions contain 1260 lbs. of water. If we water added during the process of manufacture is 10% juice made, the total water handled is 1410 lbs. From th in this case, the commercial massecuite would be about 1 the original mill juice, or say 225 lbs. Said mill juice equals 1650 lbs. liquor handled; and 1650 lbs., minus 225 lb the quantity of water to be evaporated during the proces To effect a 7½-lb. evaporation requires 190 lbs. of coal, an lb. evaporation.

"To reduce 1650 lbs. of juice to syrup of, say, 27° Baumé oration of 1770 lbs of water, leaving 480 lbs. of syrup. complished in the open air, it will require about 156 lbs. boiler evaporation, and 117 at 10 lbs. evaporation.

PETROLEUM.

oducts of the Distillation of Crude Petroleum.

American petroleum of sp. gr. 0.800 may be split up by fractional ion as follows (Robinson's Gas and Petroleum Engines):

of tion F.	Distillate.	Percentages.	Specific Gravity.	Flashing Point. Deg. F.
	Rhigolene.			
140°	Chymogene.	traces.	.590 to .625	....
158°	Gasolene (petroleum spirit)...	1.5	.636 to .657	....
248°	Benzine, naphtha C, benzolene.	10.	.680 to .700	14
	Benzine, naphtha B	2.5	.714 to .718	....
	" " A..........	2.	.725 to .737	32
	Polishing oils.	....	....	....
nd ls.	Kerosene (lamp-oil)............	50.	.802 to .820	100 to 122
	Lubricating oil...............	15.	.850 to .915	230
.....	Paraffine wax................	2.	....	....
.....	Residue and Loss..............	16.	....	....

na Petroleum, produced at Lima, Ohio, is of a dark green color, uid, and marks 48° Baumé at 15° C. (sp. gr., 0.792).

distillation in fifty parts, each part representing 2% by volume, gave llowing results:

Sp. Gr.	Per cent.	Sp. Gr.	Per cent.	Sp. Gr.	Per cent.	Sp. Gr.	Per cent.	Sp. Gr.	Per cent.	Sp. Gr.
0.680	18	0.720	34	0.764	50	0.802	68	0.820	88	0.815
.683	20	.728	36	.768	52 to 58	.806	70	.825	90	.815
.685	22	.730	38	.772			72	.830		
.690	24	.735	40	.778			73	.830	92 to 100	Residuum
.694	26	.740	42	.782	60	.800	76	.810		
.698	28	.742	44	.788	62	.804	78	.820		
.700	30	.746	46	.792	64	.808	82	.818		
.706	32	.760	48	.800	66	.812	86	.816		

RETURNS.

per cent naphtha, 70° Baumé.	6 per cent paraffine oil.
" burning oil.	10 " residuum.

distillation started at 28° C., this being due to the large amount of ha present, and when 60% was reached, at a temperature of 310° C., ydrocarbons remaining in the retort were dissociated, then gases ed, lighter distillates were obtained, and, as usual in such cases, the rature decreased from 310° C. down gradually to 200° C., until 75% of s obtained, and from this point the temperature remained constant the end of the distillation. Therefore these hydrocarbons in *statu ndi* absorbed much heat. (*Jour. Am. Chem. Soc.*)

lue of Petroleum as Fuel.—Thos. Urquhart, of Russia (Proc. I. E., Jan. 1889), gives the following table of the theoretical evaporaower of petroleum in comparison with that of coal, as determined by s. Favre & Silbermann:

Fuel.	Specific Gravity at 32° F., Water = 1.000.	Chem. Comp.			Heating-power, British Thermal Units.	Theoret. Evap., lbs. Water per lb. Fuel, from and at 212° F.
		C.	H.	O.		
	S. G.	p. c.	p. c.	p. c.	Units.	lbs.
. heavy crude oil....	0.886	84.9	13.7	1.4	20,736	21.48
sian light crude oil..	*0.884*	*86.3*	*13.6*	0.1	22,027	22.79
heavy " "..	*0.938*	*86.6*	*12.3*	1.1	20,138	20.85
um refuse..........	*0.928*	*87.1*	*11.7*	1.2	19,832	[illegible]
nglish Coal, Mean amples............	*1.380*	*80.0*	5.0	8.0	14,112	[illegible]

difference in the cost of handling the coal, ashes, and oil.
In 1892 there were reported to the Engineers' Club of P
comparative figures, from tests undertaken to ascertain t
of coal, petroleum, and gas.

	I
1 lb. anthracite coal evaporated	
1 lb. bituminous coal	
1 lb. fuel oil, 36° gravity	
1 cubic foot gas, 20 C. P.	

The gas used was that obtained in the distillation of p
about the same fuel-value as natural or coal-gas of equal c
Taking the efficiency of bituminous coal as a basis, the c
petroleum is more than 60% greater than that of coal; wher
petroleum exceeds coal only about 45%—the one containing
and the other 21,000.

Crude Petroleum vs. Indiana Block Co raising at the South Chicago Steel Works Trans. A. I. M. E., xvii, 807.)—With coal, 14 tubular boiler quired 25 men to operate them; with fuel oil, 6 men were r of 19 men at $2 per day, or $38 per day.

For one week's work 2731 barrels of oil were used, again required for the same work, showing 3.22 barrels of oil to b ton of coal. With oil at 60 cents per barrel and coal at $2.1 ative cost of oil to coal is as $1.93 to $2.15. No evapo made.

Petroleum as a Metallurgical Fuel.—C. E. Fe M. E., xvii, 809) reports a series of trials with oil as fuel in open-hearth steel-furnaces, and in raising steam with resu In a run of six weeks the consumption of oil, partly refi and some of the naphtha being removed), in heating 14-inch furnaces was about 6½ gallons per ton of blooms. 2. In m open-hearth furnace 48 gallons of oil were used per ton o six weeks' trial with Lima oil from 47 to 54 gallons of oil w ton of ingots. 4. In a six months' trial with Siemens heat consumption of Lima oil was 6 gallons per ton of ingots.

roducer, and by utilizing the sensible heat of the gas in ace. It ought to be possible to oxidize one out of every with oxygen derived from water-vapor. The thermic ration are as follows:

	Heat-units.
CO (3 lbs. gasified with air and 1 lb. with water)	17,600
ich furnish 1.33 lbs. of oxygen to combine with 1 orb by dissociation........	10,333
f 9.333 lbs. CO, 0.167 lb. H, and 13.39 lbs. N, heated	3,748
n and loss	3,519
	17,600

is blown into a producer with the air is almost all conivided water before entering the fuel, and consequently er in these calculations.

ter liberates .167 lb. of hydrogen, which is delivered to in combustion the same heat that it absorbs in the pron. According to this calculation, therefore, 60% of the nbustion is theoretically recovered by the dissociation of all the sensible heat of the gas be counted, with radiaor items, as loss, yet the gas must carry 4 × 14,500 – 3 heat-units, or 87% of the calorific energy of the carbon. s a loss in conversion of 13%, without crediting the gas it, or charging it with the heat required for generating , or taking into account the loss due to oxidizing some $_2$. In good producer-practice the proportion of CO_2 in rom 4% to 7% of the C burned to CO_2, but the extra heat should be largely recovered in the dissociation of more erefore does not represent as much loss as it would indir of energy, this gas has the advantage of carrying 4.46 han would be present if the fourth pound of coal had ir; and in practical working the use of steam reduces ering in the producer.

as.—In anthracite coal there is a volatile combustible y from 1.5% to over 7%. The amount of energy derived own in the following theoretical gasification made with mposition: Carbon, 85%; vol. HC, 5%; ash, 10%; 80 lbs. carburned to CO; 5 lbs. carbon burned to CO_2; three fourths ygen derived from air, and one fourth from water.

	Products.		
	Pounds.	Cubic Feet.	Anal. by Vol.
........CO	186.66	2529.24	33.4
........CO_2	18.33	157.64	2.0
stilled)	5.00	116.60	1.6
required, of which O liberate........H	3.75	712.50	9.4
associatied with N	301.05	4064.17	53.6
	514.79	7580.15	100.0

ove gas obtained from 100 lbs. anthracite:

66 lbs. CO....	807,304	heat-units.
00 " CH_4..........	117,500	"
75 " H	232,500	"
	1,157,304	"
y in gas per lb.......	2,248	"
" 100 lbs. of coal..	1,349,500	"
of the conversion	86%.	

nd H exceeds the results obtained in practice. The senas will *probably account for this discrepancy*, and, there*assume the possibility of delivering at least* 8?% of the *acite.*

Gas.—*A theoretical gasification of 100 lbs.* ... *and 32% of volatile combustible (which* ... *coal), is made in the following table* ... *urned to CO and 5 lbs. to* CO_2; one fourt...

derived from steam and three fourths from air; the heat val volatile combustible is taken at 20,000 heat-units to the pound. In ing volumetric proportions all the volatile hydrocarbons, fixed a condensing, are classed as marsh-gas, since it is only by some su tive assumption that even an approximate idea of the volumetric tion can be formed. The energy, however, is calculated from wei

Process.	Products. Pounds.	Products. Cubic Feet.	Ana
50 lbs. C burned to..................CO	116.66	1580.7	
5 lbs. C burned to.......... ......CO_2	18.33	157.6	
32 lbs. vol. HC (distilled)...............	32.00	746.2	
80 lbs. O are required, of which 20 lbs., derived from H_2O, liberate.......H	2.5	475.0	
60 lbs. O, derived from air, are associated with......................N	200.70	2709.4	
	370.19	5668.9	

Energy in 116.66 lbs. CO..	504,554	heat-units.
" " 32.00 lbs. vol. HC....	640,000	"
" " 2.50 lbs. H.........	155,000	"
	1,299,554	"
Energy in coal.....	1,437,500	"
Per cent of energy delivered in gas............		90.0
Heat-units in 1 lb. of gas........................		3,484

Water-gas.—Water-gas is made in an intermittent process, by up the fuel-bed of the producer to a high state of incandescence some cases utilizing the resulting gas, which is a lean producer-g shutting off the air and forcing steam through the fuel, which di the water into its elements of oxygen and hydrogen, the former co with the carbon of the coal, and the latter being liberated.

This gas can never play a very important part in the industrial fiel to the large loss of energy entailed in its production, yet there ar and special purposes where it is desirable, even at a great excess in unit of heat over producer-gas; for instance, in small high-temperat naces, where much regeneration is impracticable, or where the "bl gas can be used for other purposes instead of being wasted.

The reactions and energy required in the production of 1000 feet o gas, composed, theoretically, of equal volumes of CO and H, are as

500 cubic feet of H weigh..................................	2.63
500 cubic feet of CO weigh..................................	36.89
Total weight of 1000 cubic feet........	39.52

Now, as CO is composed of 12 parts C to 16 of O, the weight of C lbs. is 15.81 lbs. and of O 21.08 lbs. When this oxygen is derived fro it liberates, as above, 2.635 lbs. of hydrogen. The heat developed sorbed in these reactions (roughly, as we will not take into account ergy required to elevate the coal from the temperature of the atm to say 1800°) is as follows:

	He
2.635 lbs. H absorb in dissociation from water 2.625 × 62,000..	= 1
15.81 lbs. C burned to CO develops 15.81 × 4400................	=
Excess of heat-absorption over heat-development	=

If this excess could be made up from C burnt to CO_2 without loss ation, we would only have to burn an additional 4.83 lbs. C to sup heat, and we could then make 1000 feet of water-gas from 20.64 lbs bon (equal 24 lbs. of 85% coal). This would be the perfection of gas as the gas would contain really the same energy as the coal; but in require in practice more than double this amount of coal, and do no more than 50% of the energy of the fuel in the gas, because the su heat is obtained in an indirect way and with imperfect combustion. this, it is not often that the sum of the CO and H exceed 90%, the bal ing CO_2 and N. But water-gas should be made with much less lo ergy by burning the "blow-up" (producer) gas in brick regene stored-up heat of which can be returned to the producer by the blowing-up.

The following table shows what may be considered avera

e weight and energy of 1000 cubic feet, of the four types of
eating and illuminating purposes:

	Natural Gas.	Coal-gas.	Water-gas.	Producer-gas.	
				Anthra.	Bitu.
.. ..	0.50	6.0	45.0	27.0	27.0
....	2.18	46.0	45.0	12.0	12.0
....	92.6	40.0	2.0	1.2	2.5
....	0.31	4.0	......	......	0.4
....	0.26	0.5	4.0	2.5	2.5
....	3.61	1.5	2.0	57.0	56.2
....	0.34	0.5	0.5	0.3	0.3
....	....	1.5	1.5	....	......
ubic feet..........	%45.6	32.0	45.6	65.6	65.9
00 cubic feet......	1,100,000	735,000	322,000	137,455	156,917

Natural Gas in Ohio and Indiana.

(*Eng. and M. J.*, April 21, 1894.)

	Ohio.			Indiana.			
	Fos-toria.	Findlay	St. Mary's.	Muncie.	Ander-son.	Koko-mo.	Mar-ion.
....	1.89	1.64	1.94	2.35	1.86	1.42	1.20
....	92.84	93.35	93.85	92.67	93.07	94.16	93.57
....	.20	.35	.20	.25	.47	.30	.15
de..	.55	.41	.44	.45	.73	.55	.60
....	.20	.25	.23	.25	.26	.29	.30
....	.35	.39	.35	.35	.42	.30	.55
....	3.82	3.41	2.98	3.53	3.02	2.80	3.42
ide	.15	.20	.21	.15	.15	.18	.20

ly 30,000 cubic feet of gas have the heating power of one

roducer-gas from One Ton of Coal.

(W. H. Blauvelt, Trans. A. I. M. E., xviii. 614.)

l.	Per Cent.	Cubic Feet.	Lbs.	Equal to—
..	25.3	33,213.84	2451.20	1050.51 lbs. C + 1400.7 lbs. O.
..	9.2	12,077.76	63.56	63.56 " H.
..	3.1	4,069.68	174.66	174.66 " CH_4.
..	0.8	1,050.24	77.78	77.78 " C_2H_4.
..	3.4	4,463.52	519.02	141.54 " C + 377.44 lbs. O.
s.	58.2	76,404.96	5659.63	7350.17 " Air.
	100.0	131,280.00	8945.85	

on this basis, the 131,280 ft. of gas from the ton of coal con-
B.T.U., or 155 B.T.U. per cubic ft., or 2270 B.T.U. per lb.
ion of the coal from which this gas was made was as follows:
olatile matter, 36.22%; fixed carbon, 57.98%; sulphur, 0.70%;
ton contains 1159.6 lbs. carbon and 724.4 lbs. volatile com-
ergy of which is 31,302,200 B.T.U. Hence, in the processes of
purification *there was a loss of* 35.2% of the energy of the

on of the hydrocarbons in a soft coal is uncertain and quite
ultimate analysis of the average coal shows that
arly to the composition of CH_4 (marsh-gas).
phasizes the following points as highly importa
tice:

First. That a large percentage of the energy of the coal is lost w[illegible] gas is made in the ordinary low producer and cooled to the tempera[illegible] the air before being used. To prevent these sources of loss, the p[illegible] should be placed so as to lose as little as possible of the sensible hea[illegible] gas, and prevent condensation of the hydrocarbon vapors. A high f[illegible] should be carried, keeping the producer cool on top, thereby prevent[illegible] breaking-down of the hydrocarbons and the deposit of soot, as well a[illegible] ing the carbonic acid low.

Second. That a producer should be blown with as much steam mi[illegible] the air as will maintain incandescence. This reduces the percen[illegible] nitrogen and increases the hydrogen, thereby greatly enriching t[illegible] The temperature of the producer is kept down, diminishing the loss [illegible] by radiation through the walls, and in a large measure preventing c[illegible]

The Combustion of Producer-gas. (H. H. Campbell, [illegible] A. I. M. E., xix, 128.)—The combustion of the components of ordinar[illegible] ducer-gas may be represented by the following formulæ:

$$C_2H_4 + 6O = 2CO_2 + 2H_2O; \quad 2H + O = H_2O;$$
$$CH_4 + 4O = CO_2 + 2H_2O; \quad CO + O = CO_2.$$

AVERAGE COMPOSITION BY VOLUME OF PRODUCER-GAS: A, MADE WITH [illegible] GRATES, NO STEAM IN BLAST; B, OPEN GRATES, STEAM-JET IN BLA[illegible] SAMPLES OF EACH.

	CO_2.	O.	C_2H_4.	CO.	H.	CH_4.
A min	3.6	0.4	0.2	20.0	5.3	3.0
A max........	5.6	0.4	0.4	24.8	8.5	5.2
A average...	4.84	0.4	0.34	22.1	6.8	3.74
B min........	4.6	0.4	0.2	20.8	6.9	2.2
B max	6.0	0.8	0.4	24.0	9.8	3.4
B average...	5.3	0.54	0.36	22.74	8.37	2.56

The coal used contained carbon 82%, hydrogen 4.7%.

The following are analyses of products of combustion:

	CO_2.	O.	CO.	CH_4.	H.	N.
Minimum........	15.2	0.2	trace.	trace.	trace.	[illegible]
Maximum	17.2	1.6	2.0	0.6	2.0	[illegible]
Average	16.3	0.8	0.4	0.1	0.2	[illegible]

Use of Steam in Producers and in Boiler-furnaces. [illegible] W. Raymond, Trans. A. I. M. E., xx. 635.)—No possible use of stea[illegible] cause a gain of heat. If steam be introduced into a bed of incand[illegible] carbon it is decomposed into hydrogen and oxygen.

The heat absorbed by the reduction of one pound of steam to hydro[illegible] much greater in amount than the heat generated by the union o[illegible] oxygen thus set free with carbon, forming either carbonic oxide or car[illegible] acid. Consequently, the effect of steam alone upon a bed of incande[illegible] fuel is to chill it. In every water-gas apparatus, designed to produ[illegible] means of the decomposition of steam a fuel-gas relatively free from [illegible] gen, the loss of heat in the producer must be compensated by some re[illegible] ing device.

This loss may be recovered if the hydrogen of the steam is subseque[illegible] burned, to form steam again. Such a combustion of the hydrogen is [illegible] templated, in the case of fuel-gas, as secured in the subsequent use of [illegible] gas. Assuming the oxidation of H to be complete, the use of stea[illegible] cause neither gain nor loss of heat, but a simple transference, the [illegible] absorbed by steam decomposition being restored by hydrogen combu[illegible] In practice, it may be doubted whether this restoration is ever comp[illegible] But it is certain that an excess of steam would defeat the reaction [illegible] gether, and that there must be a certain proportion of steam, which [illegible] mits the realization of important advantages, without too great a net l[illegible] heat.

The advantage to be secured (in boiler furnaces using small [illegible] anthracite) consists principally in the transfer of heat from the lower [illegible] of the fire, where it is not wanted, to the upper side, where it is wa[illegible] *The decomposition of the steam below cools the fuel and the grate[illegible] whereas a blast of air alone would produce, at that point, intense com[illegible] tion (for*[illegible] first CO_2), to the injury of the grate, the fusion of [illegible] *the f*[illegible]

T[illegible] steam most economical is not easily determine[illegible] te[illegible] eam itself, the nature of the fuel mixture, [illegible] r[illegible] ary air-supply, introduced into the [illegible] he fire-bed, are factors affecting the problem. (See Trans. ., xx. 625.)

nalyses by Volume and by Weight.—To convert an an- mixed gas by volume into analysis by weight: Multiply the per- f each constituent gas by the density of that gas (see p. 166). Divide luct by the sum of the products to obtain the percentages by weight.

uel for Small Furnaces.—E. P. Reichhelm (*Am. Mach.*, 1895) discusses the use of gaseous fuel for forge fires, for drop- n annealing-ovens and furnaces for melting brass and copper, for lening, muffle-furnaces, and kilns. Under ordinary conditions, in naces he estimates that the loss by draught, radiation, and the f space not occupied by work is, with coal, 80%, with petroleum 70%, gas above the grade of producer-gas 25%. He gives the following omparative cost of fuels, as used in these furnaces:

Kind of Gas.	No. of Heat-units in 1,000 cu. ft. used.	No. of Heat-units in Furnaces after Deducting 25% Loss.	Average Cost per 1,000 Ft.	Cost of 1,000,000 Heat-units Obtained in Furnaces.
gas	1,000,000	750,000	....	....
, 20 candle-power	675,000	506,250	$1.25	$2.46
ted water-gas.	646,000	484,500	1.00	2.06
gas, 20 candle-power	690,000	517,500	.90	1.73
as from coke.	313,000	234,750	.40	1.70
as from bituminous coal	377,000	282,750	.45	1.59
as and producer-gas mixed	185,000	138,750	.20	1.44
r-gas	150,000	112,500	.15	1.33
-gas, fuel 2½ gals. per 1000 ft.	306,365	229,774	.15	.65
per ton, per 1,000,000 heat-units utilized				.73
etroleum, 3 cts. per gal., per 1,000,000 heat-units.				.73

eichhelm gives the following figures from practice in melting brass l and with naphtha converted into gas: 1800 lbs. of metal require of coal, at $4.65 per ton, equal to $2.51, or, say, 15 cents per 100 lbs. report: 2500 lbs. of metal require 47 gals. of naphtha, at 6 cents per al to $2.82, or, say, 11¼ cents per 100 lbs.

ILLUMINATING-GAS.

-gas is made by distilling bituminous coal in retorts. The retort ly a long horizontal semi-cylindrical or ◠ shaped chamber, holding 60 to 300 lbs. of coal. The retorts are set in "benches" of from heated by one fire, which is generally of coke. The vapors distilled e coal are converted into a fixed gas by passing through the retort, is heated almost to whiteness.

as passes out of the retort through an "ascension-pipe" into a long tal pipe called the hydraulic main, where it deposits a portion of it contains; thence it goes into a condenser, a series of iron tubes ded by cold water, where it is freed from condensable vapors, as ia-water, then into a washer, where it is exposed to jets of water, to a scrubber, a large chamber partially filled with trays made of r iron, containing coke, fragments of brick or paving-stones, which t with a spray of water. By the washer and scrubber the gas is freed the last portion of tar and ammonia and from some of the sulphur unds. The gas is then finally *purified* from sulphur compounds by g it *through lime or oxide of iron*. The gas is drawn from the hy- *main and forced through the washer*, scrubber, etc., by an exhauster *ump.*

d of coal used is generally caking bituminous, but as u *cient in gases of high illuminating* power, there is add *cannel coal or other enricher.*

wing table, abridged from one in Johnson's Cyclop *candle power, etc.*, of some gas-coals and enrichers

Gas-coals, etc.	Vol. Matter.	Fixed Carb.	Ash.	Gas per ton of 2240 lbs. in cu. ft.	Cand.-pow'r of Gas.	Coke per ton of 2240 lbs.	
						lbs.	[illegible]
Pittsburgh, Pa	36.76	51.93	7.07	....	......	....	
Westmoreland, Pa	36.00	58.00	6.00	10,642	16.62	1544	[illegible]
Sterling, O................	37.50	56.90	5.60	10,528	18.81	1480	[illegible]
Despard, W. Va.........	40.00	53.30	6.70	10,765	20.41	1540	[illegible]
Darlington, O............	43.00	40.00	17.00	9,800	34.98	1320	[illegible]
Petonia, W. Va	46.00	41.00	13.00	13,200	42.79	1380	[illegible]
Grahamite, W. Va.......	53.50	44.50	2.00	15,000	28.70	1056	[illegible]

The products of the distillation of 100 lbs. of average gas-coal are follows. They vary according to the quality of coal and the tempe distillation.

Coke, 64 to 65 lbs.; tar, 6.5 to 7.5 lbs.; ammonia liquor, 10 to 12 fied gas, 15 to 12 lbs.; impurities and loss, 4.5% to 3.5%.

The composition of the gas by volume ranges about as follow gen, 38% to 48%; carbonic oxide, 2% to 14%; marsh-gas (Methane, CH 31%; heavy hydrocarbons (C_nH_{2n}, ethylene, propylene, benzole va 7.5% to 4.5%; nitrogen, 1% to 3%.

In the burning of the gas the nitrogen is inert; the hydrogen and oxide give heat but no light. The luminosity of the flame is due to composition by heat of the heavy hydrocarbons into lighter hydr and carbon, the latter being separated in a state of extreme sub By the heat of the flame this separated carbon is heated to inten ness, and the illuminating effect of the flame is due to the light of cence of the particles of carbon.

The attainment of the highest degree of luminosity of the flame upon the proper adjustment of the proportion of the heavy hydro (with due regard to their individual character) to the nature of the mixed therewith.

Investigations of Percy F. Frankland show that mixtures of ethyl hydrogen cease to have any luminous effect when the proportion lene does not exceed 10% of the whole. Mixtures of ethylene and oxide cease to have any luminous effect when the proportion of the does not exceed 20%, while all mixtures of ethylene and marsh-gas ha or less luminous effect. The luminosity of a mixture of 10% ethylene marsh-gas being equal to about 18 candles, and that of one of 20% and 80% marsh-gas about 25 candles. The illuminating effect of ma alone, when burned in an argand burner, is by no means inconsider

For further description, see the Treatises on Gas by King, Richar Hughes; also Appleton's Cyc. Mech., vol. I, p. 900.

Water-gas.—Water-gas is obtained by passing steam through coal, coke, or charcoal heated to redness or beyond. The steam is posed, its hydrogen being liberated and its oxygen burning the ca the fuel, producing carbonic-oxide gas. The chemical reaction is C $= CO + 2H$, or $2C + 2H_2O = C + CO_2 + 4H$, followed by a splitting the CO_2, making $2CO + 4H$. By weight the normal gas $CO + 2H$ posed of $\underset{12}{C} + \underset{16}{O} + \underset{2}{H}$ = 28 parts CO and 2 parts H, or 93.33% CO and by volume it is composed of equal parts of carbonic oxide and hy Water-gas produced as above described has great heating-power illuminating-power. It may, however, be used for lighting by caus heat to whiteness some solid substance, as is done in the Welsbach descent light.

An illuminating-gas is made from water-gas by adding to it hydro gases or vapors, which are usually obtained from petroleum or som *products*. A history of the development of modern illuminating wa *processes, together with a description of the most recent forms of* ap *given by* Alex. C. Humphreys, in a paper on "Water-gas in the ...," read ... the Mechanical Section of the British Asso ... in 1889. After describing many ... manufacture of water-gas may be en the process of T. S. C. Lowe was introduced. All the later ful processes are the modifications of Lowe's, the essential hich were "an apparatus consisting of a generator and superally fired; the superheater being heated by the secondary rom the generator, the heat so stored up in the loose brick of er being used, in the second part of the process, in the fixing permanent of the hydrocarbon gases; the second part of the sting in the passing of steam through the generator fire, and of oil or hydrocarbon at some point between the fire of the d the loose filling of the superheater."

gas process thus has two periods: first the "blow," during blown through the bed coal in the generator, and the partially us products are completely burned in the superheater, giving ortion of their heat to the fire-brick work contained in it, and to a chimney; second, the "run" during which the air blast e opening to the chimney closed, and steam is blown through ent bed of fuel. The resulting water-gas passing into the carmber in the base of the superheater is there charged with hypors, or spray (such as naphtha and other distillates or crude s through the superheater, where the hydrocarbon vapors beted into fixed illuminating gases. From the superheater the ses are passed, as in the coal-gas process, through washers, c., to the gas-holder. In this case, however, there is no amremoved.

gravity of water-gas increases with the increase of the heavy s which give it illuminating power. The following figures, taken t authorities, are given by F. H. Shelton in a paper on Waterore the Ohio Gas Light Association, in 1894:

r ...	19.5	20.	22.5	24.	25.4	26.3	28.3	29.6	.30 to 31.9
= 1)..	.571	.630	.589	.60 to .67	.64	.602	.70	.65	.65 to .71

yses of Water-gas and Coal-gas Compared.

ing analyses are taken from a report of Dr. Gideon E. Moore er Water-gas, 1885:

	Composition by Volume.			Composition by Weight.		
	Water-gas.		Coal-gas. Heidelberg.	Water-gas.		Coal-gas.
	Worcester.	Lake.		Worcester.	Lake.	
....	2.64	3.85	2.15	0.04402	0.06175	0.04559
id	0.14	0.30	3.01	0.00365	0.00753	0.09992
....	0.06	0.01	0.65	0.00114	0.00018	0.01569
....	11.29	12.80	2.55	0.18759	0.20454	0.05389
....	0.00	0.00	1.21	....	....	0.03834
or.....	1.53	2.63	1.33	0.07077	0.11700	0.07825
ide....	28.26	23.58	8.88	0.46934	0.37664	0.18758
....	18.88	20.95	34.02	0.17928	0.19133	0.41087
....	37.20	35.88	46.20	0.04421	0.04103	0.06987
	100.00	100.00	100.00	1.00000	1.00000	1.00000
heory.	0.5825	0.6057	0.4580	....	....	....
actice.	0.5915	0.6018	....	....	....	....
m 1 cu.				....	....	....
liquid.	650.1	688.7	642.0	....	....	....
vapor.	597.0	646.6	577.0	....	....	....
p..	5311.2°F.	5281.1°F.	5202.9°F.	....	....	....
ower.	*22.06*	*26.31*	....	....	....	....

values (B. T. U.) of the gases are calculate using the multipliers given below (comp

J. Thomsen), and multiplying the result by the weight of 1 cu. ft at 62° F., and atmospheric pressure.

The flame temperatures (theoretical) are calculated on the as complete combustion of the gases in air, without excess of air.

The candle-power was determined by photometric tests, using of ½-in. water-column, a candle consumption of 120 grains of per hour, and a meter rate of 5 cu. ft. per hour, the result bein for a temperature of 62° F. and a barometric pressure of 30 in. that the candle-power may be regulated at the pleasure of th charge of the apparatus, the range of candle-power being fro candles, according to the manipulation employed.

Calorific Equivalents of Constituents of Illum gas.

	Heat-units from 1 lb. Water Liquid.	Water Vapor.		Heat-unit Water Liquid.
Ethylene.........	21,524.4	20,134.8	Carbonic oxide..	4,395.6
Propylene........	21,222.0	19,834.2	Marsh-gas........	24,021.0
Benzole vapor....	18,954.0	17,847.0	Hydrogen........	61,524.0

Efficiency of a Water-gas Plant.—The practical effic illuminating water-gas setting is discussed in a paper by A. ((Proc. Am. Gaslight Assn., 1890), from which the following is abri

The results refer to 1000 cu. ft. of unpurified carburetted gas, 60° F. The total anthracite charged per 1000 cu. ft. of gas was 33 and unconsumed coal removed 9.9 lbs., leaving total combustible 23.5 lbs., which is taken to have a fuel-value of 14500 B. T. U. per a total of 340,750 heat-units.

		Composition by Volume.	Weight per 100 cu. ft.	Composition by Weight.
I. Carburetted Water-gas.	$CO_2 + H_2S$..	3.8	.465842	.09647
	C_nH_{2n}	14.6	1.139968	.23607
	CO........ ...	28.0	2.1868	.45285
	CH_4	17.0	.75854	.15710
	H............	35.6	.1991464	.04124
	N............	1.0	.078596	.01627
		100.0	4.8288924	1.00000
II. Uncarburetted gas.	CO_2	3.5	.429065	.1019
	CO...........	43.4	3.389540	.8051
	H............	51.8	.289821	.0688
	N............	1.3	.102175	.0242
		100.0	4.210601	1.0000
III. Blast products escaping from superheater.	CO_2..........	17.4	2.133066	.2464
	O.............	3.2	.2856096	.0329
	N	79.4	6.2405224	.7207
		100.0	8.6591980	1.0000
IV. Generator blast-gases.	CO_2..........	9.7	1.189123	.1436
	CO...........	17.8	1.390180	.1680
	N.	72.5	5.698210	.6884
		100.0	8.277513	1.0000

The heat energy absorbed by the apparatus is 23.5 × 14,500 = 3 units = A. Its disposition is as follows:

... rgy of the CO produced;

... y absorbed in the decomposition of the steam;

... nce between the sensible heat of the escaping ... f the entering oil;

... ried off by the escaping blast products;

... by radiation from the shells;

e heat carried away from the shells by convection (air-currents);
e heat rendered latent in the gasification of the oil;
sensible heat in the ash and unconsumed coal recovered from the tor.

heat equation is $A = B + C + D + E + F + G + H + I$; A being

A comparison of the CO in Tables I and II show that $\frac{280}{434}$, or 64.5% volume of carburetted gas is pure water-gas, distributed thus: CO_2, O, 28.0%; H, 33.4%; N, 0.8%; = 64.5%. 1 lb. of CO at 60° F. = 13.531 cu. per 1000 cu. ft. of gas = 280 ÷ 13.531 = 20.694 lbs. Energy of the CO 94 × 4395.6 = 91,043 heat-units, = B. 1 lb. of H at 60° F. = 189.2 cu. per M of gas = 334 ÷ 189.2 = 1.7653 lbs. Energy of the H per lb. ding to Thomsen, considering the steam generated by its combustion condensed to water at 75° F.) = 61,524 B. T. U. In Mr. Glasgow's ex-ents the steam entered the generator at 331° F.; the heat required to the product of combustion of 1 lb. of H, viz., 8.98 lbs. H_2O, from water to steam at 331° must therefore be deducted from Thomsen's figure, or − (8.98 × 1140.2) = 51,285 B. T. U. per lb. of H. Energy of the H, then, 53 × 51,285 = 90,533 heat-units, = C. The heat lost due to the sensible n the illuminating-gases, their temperature being 1450° F., and that of ntering oil 235° F., is 48.29 (weight) × .45786 sp. heat × 1215 (rise of tem-ure) = 26,864 heat-units = D.

e specific heat of the entering oil is approximately that of the issuing

heat carried off in 1000 cu. ft. of the escaping blast products is 86.592 ht) × .23645 (sp. heat) × 1474° (rise of temp.) = 30,180 heat-units: the erature of the escaping blast gases being 1550° F., and that of the ing air 76° F. But the amount of the blast gases, by registra-of an anemometer, checked by a calculation from the analyses of the gases, was 2457 cubic feet for every 1000 cubic feet of carburetted gas e. Hence the heat carried off per M. of carburetted gas is 30,180 × = 74,152 heat-units = E.

periments made by a radiometer covering four square feet of the shell e apparatus gave figures for the amount of heat lost by radiation 454 heat-units = F, and by convection = 15,696 heat-units = G.

e heat rendered latent by the gasefication of the oil was found by taking difference between all the heat fed into the carburetter and super-er and the total heat dissipated therefrom to be 12,841 heat-units = H. sensible heat in the ash and unconsumed coal is 9.9 lbs. × 1500° × .25 ht.) = 3712 heat-units = I.

e sum of all the items $B + C + D + E + F + G + H + I$ = 327,295 heat-s, which substracted from the heat energy of the combustible consumed, 50 heat-units, leaves 13,455 heat-units, or 4 per cent, unaccounted for.

the total heat energy of the coal consumed, or 340,750 heat-units, the gy wasted is the sum of items D, E, F, G, and I, amounting to 132,878 -units, or 39 per cent; the remainder, or 207,872 heat-units, or 61 per being utilized. The efficiency of the apparatus as a heat machine is efore 61 per cent.

e gallons, or 35 lbs. of crude petroleum were fed into the carburetter 1000 cu. ft. of gas made; deducting 5 lbs. of tar recovered, leaves 30 lbs. ,000 = 600,000 heat-units as the net heating value of the petroleum used. ng this to the heating value of the coal, 340,750 B. T. U., gives 940,750 -units, of which there is found as heat energy in the carburetted gas, as e table below, 764,050 heat units, or 81 per cent, which is the commer-efficiency of the apparatus, i.e., the ratio of the energy contained in nished product to the total energy of the coal and oil consumed.

e heating power per M. cu. ft. of the carburetted gas is		The heating power per M. of the uncarburetted gas is
38.0	CO_2	35.0
* 146.0 × .117220 × 21222.0 = 363200	CO	434.0 × .078100 × 4395.6 = 148[illegible]
280.0 × .078100 × 4395.6 = 96120	H	518.0 × .005594 × 61524.0 = 178[illegible]
170.0 × .044620 × 24021.0 = 182210	N	13.0
356.0 × .005594 × 61524.0 = 122520		
10.0		1000.0 [illegible]
00.0 764050		

heating value of the illuminants C_nH_{2n} is assur

The candle-power of the gas is 31, or 6.2 candle-power per gall[on] used. The calculated specific gravity is .6355, air being 1.

For description of the operation of a modern carburetted w[ater-gas] plant, see paper by J. Stelfox, *Eng'g*, July 20, 1894, p. 89.

Space required for a Water-gas Plant.—Mr. Shelton, [from] 15 modern plants of the form requiring the most floor-space, fig[ures the] average floor-space required per 1000 cubic feet of daily capacity as [follows]:

Water-gas Plants of Capacity in 24 hours of	Require an Area of Floor-s[pace for] each 1000 cu. ft. of abo[ve]
100,000 cubic feet	4 square feet.
200,000 " "	3.5 " "
400,000 " "	2.75 " "
600,000 " "	2 to 2.5 sq. ft.
7 to 10 million cubic feet	1.25 to 1.5 sq. ft.

These figures include scrubbing and condensing rooms, but not b[oiler and] engine rooms. In coal-gas plants of the most modern and compact fo[rm,] with 16 benches of 9 retorts each, with a capacity of 1,500,000 cubic [feet in] 24 hours, will require 4.8 sq. ft. of space per 1000 cu. ft. of gas, and [with] benches of 6 retorts each, with 300,000 cu. ft. capacity per 24 hours [will re]quire 6 sq. ft. of space per 1000 cu. ft. The storage-room required [for] gas-making materials is: for coal-gas, 1 cubic foot of room for e[very ...] cubic feet of gas made; for water-gas made from coke, 1 cubic foot for every 373 cu. ft. of gas made; and for water-gas made from ant[hracite,] 1 cu. ft. of room for every 645 cu. ft. of gas made.

The comparison is still more in favor of water-gas if the case is co[nsidered] of a water-gas plant added as an auxiliary to an existing coal-ga[s plant;] for, instead of requiring further space for storage of coke, part [of the space] already required for storage of coke produced and not at once sol[d may be] cut off, by reason of the water-gas plant creating a constant dem[and for] more or less of the coke so produced.

Mr. Shelton gives a calculation showing that a water-gas of .6[...] would require gas-mains eight per cent greater in diameter than [for the same] quantity coal-gas of .425 sp. gr. if the same pressure is maintain[ed at the] holder. The same quantity may be carried in pipes of the same [diameter] if the pressure is increased in proportion to the specific gravity. [With the] same pressure the increase of candle-power about balances the de[creased] flow. With five feet of coal-gas, giving, say, eighteen candle-pow[er, each] foot equals 3.6 candle-power; with water-gas of 23 candle-power [each] foot equals 4.6 candle-power, and 4 cubic feet gives 18.4 candle-[power, or] more than is given by 5 cubic feet of coal-gas. Water-gas may [be made] from oven-coke or gas-house coke as well as from anthracite coal, [and a water-]gas plant may be conveniently run in connection with a coal-gas [plant, the] surplus retort coke of the latter being used as the fuel of the form[er.]

In coal-gas making it is impracticable to enrich the gas to ov[er ...] candle-power without causing too great a tendency to smoke, but [water-gas] of as high as thirty candle-power is quite common. A mixture of [coal-gas] and water-gas of a higher C.P. than 20 can be advantageously dis[tributed.]

Fuel-value of Illuminating-gas.—E. G. Love (School [of Mines] Qtly, January, 1892) describes F. W. Hartley's calorimeter for de[termining] the calorific power of gases, and gives results obtained in tests o[f car]buretted water-gas made by the municipal branch of the Consoli[dated Gas Co.] of New York. The tests were made from time to time during th[e ...] years, and the figures give the heat-units per cubic foot at 60° [and 30] inches pressure: 715, 692, 725, 732, 691, 738, 735, 708, 734, 730, 731, 727. [Average] 721 heat units. Similar tests of mixtures of coal- and water-gases [made by] other branches of the same company give 694, 715, 684, 692, 727, 6[..., ...] 686 heat-units per foot, or an average of 694.7. The average of [all the] tests was 710.5 heat-units, and this we may fairly take as represe[nting the] calorific power of the illuminating gas of New York. One thousa[nd feet of] this gas, costing $1.25, would therefore yield 710,500 heat-units, wh[ich would] be equivalent to 568,400 heat-units for $1.00.

The common coal-gas of London, with an illuminating power o[f ...] *candles*, has a calorific power of about 668 units per foot, and cost[s ...] [...] cents per thousand.

[The] product obtained by decomposing steam by incandescen[t ...] [...] in the Motay process, consists of about 40% of CO, and s mixture would have a heating-power of about 300 units per cubic foot, f sold at 50 cents per 1000 cubic feet would furnish 600,000 units for $1.00, mpared with 568,400 units for $1.00 from illuminating gas at $1.25 per 1000 feet. This illuminating-gas if sold at $1.15 per thousand would therebe a more economical heating agent than the fuel-gas mentioned, at 50 s per thousand, and be much more advantageous than the latter, in that main, service, and meter could be used to furnish gas for both lighting heating.

arge number of fuel-gases tested by Mr. Love gave from 184 to 470 heat- per foot, with an average of 309 units.

king the cost of heat from illuminating-gas at the lowest figure given r. Love, viz., $1.00 for 600,000 heat-units, it is a very expensive fuel, equal al at $40 per ton of 2000 lbs., the coal having a calorific power of only 0 heat-units per pound, or about 83% of that of pure carbon:

$$600{,}000 : (12{,}000 \times 2000) :: \$1 : \$40.$$

FLOW OF GAS IN PIPES.

e rate of flow of gases of different densities, the diameter of pipes reed, etc., are given in King's Treatise on Coal Gas, vol. ii. 374, as follows:

d = diameter of pipe in inches,
Q = quantity of gas in cu. ft. per hour,
l = length of pipe in yards,
h = pressure in inches of water,
s = specific gravity of gas, air being 1,

$$d = \sqrt[5]{\frac{Q^2sl}{(1350)^2h}},$$

$$h = \frac{Q^2sl}{(1350)^2d^5},$$

$$Q = 1350d^2\sqrt{\frac{dh}{sl}} = 1350\sqrt{\frac{d^5h}{sl}}.$$

Molesworth gives $Q = 1000\sqrt{\dfrac{d^5h}{sl}}$.

J. P. Gill, *Am. Gas-light Jour.* 1894, gives $Q = 1291\sqrt{\dfrac{d^5h}{s(l+d)}}$.

This formula is said to be based on experimental data, and to make allowce for obstructions by tar, water, and other bodies tending to check the w of gas through the pipe.

A set of tables in Appleton's Cyc. Mech. for flow of gas in 2, 6, and 12 in. pes is calculated on the supposition that the quantity delivered varies the square of the diameter instead of as $d^2 \times \sqrt{d}$, or $\sqrt{d^5}$.

These tables give a flow in large pipes much less than that calculated by e formulæ above given, as is shown by the following example. Length of pe 100 yds., specific gravity of gas .042, pressure 1-in. water-column.

	2-in. Pipe.	6-in. Pipe.	12-in. Pipe.
$Q = 1350\sqrt{\frac{d^5h}{sl}}$	1178	18,368	103,912
$Q = 1000\sqrt{\frac{d^5h}{sl}}$	873	13,606	76,972
$Q = 1291\sqrt{\frac{d^5h}{s(l+d)}}$	1116	16,327	93,845
Table in App. Cyc	1290	11,657	46,628

An experiment made by Mr. Clegg, in London, with a 4-in. pipe, 6 miles g, pressure 3 in. of water, specific gravity of gas .398, gave a discharge o the atmosphere of 852 cu. ft. per hour, after a correction of 33 cu. ft. made for leakage.

bstituting this value, 852 cu. ft., for Q in the formula $Q = C\sqrt{d^5h}$ ... nd C, the coefficient, = 997, which corresponds nearly with the ... by Molesworth.

Services for Lamps. (Molesworth.)

Lamps.	Ft. from Main.	Require Pipe-bore.	Lamps.	Ft. from Main.	Re[quire] Pipe[-bore.]
2	40	⅜ in.	15	130	[illegible]
4	40	½ in.	20	150	[illegible]
6	50	⅝ in.	25	180	[illegible]
10	100	¾ in.	30	200	[illegible]

(In cold climates no service less than ¾ in. should be used.)

Maximum Supply of Gas through Pipes in cu. [ft. per] Hour, Specific Gravity being taken at .45, calc[ulated] from the Formula $Q = 1000\sqrt{d^5h \div sl}$. (Molesworth.)

LENGTH OF PIPE = 10 YARDS.

Diameter of Pipe in Inches.	Pressure by the Water-gauge in Inches								
	.1	.2	.3	.4	.5	.6	.7	.8	.9
⅜	13	18	22	26	29	31	34	36	[illegible]
½	26	37	46	53	59	64	70	74	[illegible]
¾	73	103	126	145	162	187	192	205	[illegible]
1	149	211	258	298	333	365	394	422	[illegible]
1¼	260	368	451	521	582	638	689	737	[illegible]
1½	411	581	711	821	918	1006	1082	1162	[illegible]
2	843	1192	1460	1686	1886	2066	2231	2385	[illegible]

LENGTH OF PIPE = 100 YARDS.

	Pressure by the Water-gauge in Inches.									
	.1	.2	.3	.4	.5	.75	1.0	1.25	1.5	2
½	8	12	14	17	19	23	26	29	32	[illegible]
¾	23	32	42	46	51	63	73	81	89	[illegible]
1	47	67	82	94	105	129	149	167	183	[illegible]
1¼	82	116	143	165	184	225	260	291	319	[illegible]
1½	130	184	225	260	290	356	411	459	503	[illegible]
2	267	377	462	533	596	730	843	943	1033	[illegible]
2½	466	659	807	932	1042	1276	1473	1647	1804	[illegible]
3	735	1039	1270	1470	1643	2012	2323	2598	2846	[illegible]
3½	1080	1528	1871	2161	2416	2958	3416	3820	4184	[illegible]
4	1508	2133	2613	3017	3373	4131	4770	5333	5842	[illegible]

LENGTH OF PIPE = 1000 YARDS.

	Pressure by the Water-gauge in Inches.					
	.5	.75	1.0	1.5	2.0	2.5
1	33	41	47	58	67	75
1½	92	113	130	159	184	206
2	189	231	267	327	377	422
2½	329	403	466	571	659	737
3	520	636	735	900	1039	1162
4	1067	1306	1508	1847	2133	2385
5	[illegible]	[illegible]	2635	3227	3727	4167
6	[illegible]	[illegible]	4157	5091	5879	[illegible]

LENGTH OF PIPE = 5000 YARDS.

ter pe s.	Pressure by the Water-gauge in Inches.				
	1.0	1.5	2.0	2.5	3.0
	119	146	169	189	207
	329	402	465	520	569
	675	826	955	1067	1168
	1179	1443	1667	1863	2041
	1859	2277	2629	2939	3220
	2733	3347	3865	4321	4734
	3816	4674	5397	6034	6610
	5123	6274	7245	8100	8873
	6667	8165	9428	10541	11547
	10516	12880	14872	16628	18215

. C. Humphreys says his experience goes to show that these tables small a flow, but it is difficult to accurately check the tables, on account of the extra friction introduced by rough pipes, bends, etc. For one rule is to allow 1/42 of an inch pressure for each right-angle bend. e there is apt to be trouble from frost it is well to use no service of meter than ¾ in., no matter how short it may be. In extremely cold s this is now often increased to 1 in., even for a single lamp. The best e in the U. S. now condemns any service less than ¾ in.

STEAM.

Temperature of Steam in contact with water depends upon essure under which it is generated. At the ordinary atmospheric re (14.7 lbs. per sq. in.) its temperature is 212° F. As the pressure is ed, as by the steam being generated in a closed vessel, its tempera- nd that of the water in its presence, increases.

urated Steam is steam of the temperature due to its pressure— erheated.

erheated Steam is steam heated to a temperature above that due ressure.

Steam is steam which contains no moisture. It may be either ed or superheated.

t Steam is steam containing intermingled moisture, mist, or spray. the same temperature as dry saturated steam of the same pressure. r introduced into the presence of superheated steam will flash into until the temperature of the steam is reduced to that due its pres- Water in the presence of saturated steam has the same temperature steam. Should cold water be introduced, lowering the temperature whole mass, some of the steam will be condensed, reducing the press- d temperature of the remainder, until an equilibrium is established.

nperature and Pressure of Saturated Steam.—The re- between the temperature and the pressure of steam, according to ult's experiments, is expressed by the formula (Buchanan's, as given

k) $t = \dfrac{2938.16}{6.1993544 - \log p} - 371.85$, in which p is the pressure in pounds are inch and t the temperature of the steam in Fahrenheit degrees. ies with accuracy between 120° F. and 446° F., corresponding to pres- f from 1.68 lbs. to 445 lbs. per square inch. (For other formulæ see and Peabody's Thermodynamics.)

al Heat of Saturated Steam (above 32° F.).—According to lt's experiments, the formula for total heat of steam is $H = 1091.7 +$ $32°$), in which t is temperature Fahr., and H the heat-units. (Ran- d many others; Clark gives 1091.16 instead of 1091.7.)

ent Heat of Steam.—The formula for latent heat of steam, as by Rankine *and others, is* $L = 1091.7 - .695(t - 32°)$. Clausius n *Fahrenheit units, as given by Clark*, is $L = 1092.6 - .708t$

Work.)—The following formulas are sufficiently accurate within the given ranges of pressure (Clark, S. E.):

From 14.7 lbs. to 50 lbs. total pressure per square inch...
From 50 lbs. to 200 lbs. total pressure per square inch....

Heat required to Generate 1 lb. of Steam fr

Sensible heat, to raise the water from 32° to 212° = ...
Latent heat, 1, of the formation of steam at 212° = ...
2, of expansion against the atmospheri
pressure, 2116.4 lbs. per sq. ft. × 26.36 cu. f
= 55,786 foot-pounds ÷ 778 = ...

Total heat above 32° F........

The Heat Unit, or British Thermal Unit.—
the heat-unit used in this work is that of Rankine, accepte
writers, viz., the quantity of heat required to raise the ten
of water 1° F. at or near its temperature of maximum
Peabody's definition, the heat required to raise a pound
to 62° F. is not generally accepted. (See Thurston, T
xiii. 351.)

Specific Heat of Saturated Steam.—The spec
rated steam is .305, that of water being 1; or it is 1.281, i
The expression .305 for specific heat is taken in a compou
to changes both of volume and of pressure which takes
tion of temperature of saturated steam. (Clark, S. E.)

This statement by Clark is not strictly accurate. Whe
of saturated steam is elevated, water being present and t
ing saturated, water is evaporated. To raise the tempe
water 1° F. requires 1 thermal unit, and to evaporate it at
require 0.695 *less* thermal unit, the latent heat of saturat
ing 0.695 B.T.U. for each increase of temperature of 1°
the specific heat of water and its saturated vapor combine

me of water, the volume of water being measured at the temperature . The relative volume is found by multiplying the volume in cu. ft. of b. of steam by the weight of a cu. ft. of water at 39° F., or 62.425 lbs.

seous Steam.—When saturated steam is superheated, or surred with heat, it advances from the condition of saturation into that of ity. The gaseous state is only arrived at by considerably elevating the erature, supposing the pressure remains the same. Steam thus suffily superheated is known as gaseous steam or steam gas.

tal Heat of Gaseous Steam.—Regnault found that the total of gaseous steam increased, like that of saturated steam, uniformly the temperature, and at the rate of .475 thermal units per pound for degree of temperature, under a constant pressure.

e general formula for the total heat of gaseous steam produced from nd of water at 32° F. is $H = 1074.6 + .475t$. [This formula is for vapor rated at 32°. It is not true if generated at 212°, or at any other temperathan 32°. (Prof. Wood.)]

he Specific Heat of Gaseous Steam is .475, under constant sure, as found by Regnault. It is identical with the coefficient of ine of total heat for each degree of temperature. [This is at atmospheric sure and 212° temperature. He found it not true for any other pressure. ry indicates that it would be less at higher temperatures. (Prof. Wood.)]

he Specific Density of Gaseous Steam is .622, that of air being hat is to say, the weight of a cubic foot of gaseous steam is about five hs of that of a cubic foot of air, of the same pressure and temperature. e density or weight of a cubic foot of gaseous steam is expressible by ame formula as that of air, except that the multiplier or coefficient is n proportion to the less specific density. Thus,

$$D' = \frac{2.7074p \times .622}{t + 461} = \frac{1.684p}{t + 461},$$

hich D' is the weight of a cubic foot of gaseous steam, p the total presper square inch, and t the temperature Fahrenheit.

perheated Steam.—The above remarks concerning gaseous steam aken from Clark's Steam-engine. Wood gives for the total heat (above of superheated steam $H = 1091.7 + 0.48(t - 32°)$.

e following is abridged from Peabody (Therm., p. 115, etc.).

hen far removed from the temperature of saturation, superheated steam ws the laws of perfect gases very nearly, but near the temperature of ration the departure from those laws is too great to allow of calculations hem for engineering purposes.

e specific heat at constant pressure, C_p, from the mean of three experits by Regnault, is 0.4805.

alues of the ratio of C_p to specific heat at constant volume:

Pressure p, pounds per square inch..	5	50	100	200	300
Ratio $C_p \div C_v = k =$	1.335	1.332	1.330	1.324	1.316

euner takes k as a constant = 1.333.

SPECIFIC HEAT AT CONSTANT VOLUME, SUPERHEATED STEAM.

Pressure, pounds per square inch.....	5	50	100	200	300
Specific heat C_v........................	0.351	.348	.346	.344	.341

It is quite as reasonable to assume that C_v is a constant as to suppose that is constant, as has been assumed. If we take C_v to be constant, then C_p ll appear as a variable.

If p = pressure in lbs. per sq. ft., v = volume in cubic feet, and T = mperature in degrees Fahrenheit + 460.7, then $pv = 93.5T - 971p^{\frac{1}{4}}$.

Total heat of superheated steam, $H = 0.4805(T - 10.38p^{\frac{1}{4}}) + 857.2$.

The Rationalization of Regnault's Experiments on eam. (J. McFarlane Gray, Proc. Inst. M. E., July, 1889.)—The formulæ structed by Regnault are strictly empirical, and were based entirely on experiments. They are therefore not valid beyond the range of temperes and pressures observed.

Gray has made a most elaborate calculation, based not on experiments fundamental principles of thermodynamics, from which h e for the pressure and total heat of steam, and presents t lated therefrom which show substantial agreement with Regnault's He gives the following examples of steam-pressures calculated for te tures beyond the range of Regnault's experiments.

Temperature.		Pounds per sq. in.	Temperature.		Poun
C.	Fahr.		C.	Fahr	
230	446	406.9	340	644	[illegible]
240	464	488.9	360	680	[illegible]
250	482	579.9	380	716	[illegible]
260	500	691.6	400	752	[illegible]
280	536	940.0	415	779	[illegible]
300	572	1261.8	427	800.6	[illegible]
320	608	1661.9			

These pressures are higher than those obtained by Regnault's f which gives for 415° C. only 4067.1 lbs. per square inch.

Table of the Properties of Saturated Steam.—In t of properties of saturated steam on the following pages the figure perature, total heat, and latent heat are taken, up to 210 lbs. absol sure, from the tables in Porter's Steam-engine Indicator, which tal been widely accepted as standard by American engineers. The fi total heat, given in the original as from 0° F., have been changed above 32° F. The figures for weight per cubic foot and for cubic pound have been taken from Dwelshauvers-Dery's table, Trans. A. vol. xi., as being probably more accurate than those of Porter. Th for relative volume are from Buel's table, in Dubois's translation bach, vol. ii. They agree quite closely with the relative volumes c from weights as given by Dery. From 211 to 219 lbs. the figures for ature, total heat, and latent heat are from Dery's table; and from 2 lbs. all the figures are from Buel's table. The figures have not bee out to as many decimal places as they are in most of the tables giv different authorities; but any figure beyond the fourth significant unnecessary in practice, and beyond the limit of error of the obs and of the formulæ from which the figures were derived.

Weight of 1 Cubic Foot of Steam in Decimals of a I Comparison of Different Authorities.

Absolute Pressure, lbs. per sq. in.	Weight of 1 cubic foot according to—					Absolute Pressure, lbs. per sq. in.	Weight of 1 cubi according to-		
	Porter.	Clark	Buel.	Dery.	Peabody.		Porter.	Clark	Buel.
1	.0030	.003	.00303	.00299	.00299	120	.27428	.2738	.2735
14.7	.03797	.0380	.03793	......	.0376	140	.31386	.3162	.3163
20	.0511	.0507	.0507	.0507	.0502	160	.35209	.3590	.3589
40	.0994	.0974	.0972	.0972	.0964	180	.38895	.4009	.4012
60	.1457	.1425	.1424	.1422	.1409	200	.42496	.4431	.4433
80	.19015	.1869	.1866	.1862	.1843	220	.....	.4842	.4832
100	.23302	.2307	.2303	.2296	.2271	240	.....	.5248	.5270

There are considerable differences between the figures of weigh ume of steam as given by different authorities. Porter's figures on the experiments of Fairbairn and Tate. The figures given by authorities are derived from theoretical formulæ which are believ more reliable results than the experiments. The figures for tem total heat, and latent heat as given by different authorities show agreement, all being derived from Regnault's experiments. See *Tables of Satu*[illegible]*Steam*; also Jacobus, Trans. A. S. M. E., vol.

Properties of Saturated Steam.

cury.	Absolute Pressure, lbs. per square inch.	Temperature Fahrenheit.	Total Heat above 32° F. In the Water h Heat-units.	Total Heat above 32° F. In the Steam H Heat-units.	Latent Heat L. $= H - h$. Heat-units.	Relative Volume. Vol. of Water at 39° F. = 1.	Volume. Cu. ft. in 1 lb. of Steam.	Weight of 1 cu. ft. Steam, lb.
74	.089	32	0	1091.7	1091.7	208080	3333.3	.00030
67	.122	40	8.	1094.1	1086.1	154330	2472.2	.00040
.56	.176	50	18.	1097.2	1079.2	107630	1724.1	.00058
.40	.254	60	28.01	1100.2	1072.2	76370	1223.4	.00082
.19	.359	70	38.02	1103.3	1065.3	54660	875.61	.00115
.90	.502	80	48.04	1106.3	1058.3	39690	635.80	.00158
.51	.692	90	58.06	1109.4	1051.3	29290	469.20	.00213
.00	.943	100	68.08	1112.4	1044.4	21830	349.70	.00286
.88	1	102.1	70.09	1113.1	1043.0	20623	334.23	.00299
.85	2	126.3	94.44	1120.5	1026.0	10730	173.23	.00577
.83	3	141.6	109.9	1125.1	1015.3	7325	117.98	.00848
.78	4	153.1	121.4	1128.6	1007.2	5588	89.80	.01112
.74	5	162.3	130.7	1131.4	1000.7	4530	72.50	.01373
.70	6	170.1	138.6	1133.8	995.2	3816	61.10	.01631
.67	7	176.9	145.4	1135.9	990.5	3302	53.00	.01887
.63	8	182.9	151.5	1137.7	986.2	2912	46.60	.02140
.60	9	188.3	156.9	1139.4	982.4	2607	41.82	.02391
.56	10	193.2	161.9	1140.9	979.0	2361	37.80	.02641
.52	11	197.8	166.5	1142.3	975.8	2159	34.61	.02889
.49	12	202.0	170.7	1143.5	972.8	1990	31.90	.03136
.45	13	205.9	174.7	1144.7	970.0	1846	29.58	.03381
.41	14	209.6	178.4	1145.9	967.4	1721	27.59	.03625
ge sure per in.	14.7	212	180.9	1146.6	965.7	1646	26.36	.03794
04	15	213.0	181.9	1146.9	965.0	1614	25.87	.03868
	16	216.3	185.3	1147.9	962.7	1519	24.33	.04110
	17	219.4	188.4	1148.9	960.5	1434	22.98	.04352
	18	222.4	191.4	1149 8	958.3	1359	21.78	.04592
	19	225.2	194.3	1150.6	956.3	1292	20.70	.04831
	20	227.9	197.0	1151.5	954.4	1231	19.72	.05070
	21	230.5	199.7	1152.2	952.6	1176	18.84	.05308
	22	233.0	202.2	1153.0	950.8	1126	18.03	.05545
	23	235.4	204.7	.7	949.1	1080	17.30	.05782
	24	237.8	207.0	1154.5	947.4	1038	16.62	.06018
	25	240.0	209.3	1155.1	945.8	998.4	15.99	.06253
	26	242.2	211.5	.8	944.3	962.3	15.42	.06487
	27	244.3	213.7	1156.4	942.8	928.8	14.88	.06721
	28	246.3	215.7	1157.1	941.3	897.6	14.38	.06955
	29	248.3	217.8	.7	939.9	868.5	13.91	.07188
	30	250.2	219.7	1158.3	938.9	841.3	13.48	.07420
	31	252.1	221.6	.8	937.2	815 8	13.07	.07652
	32	254.0	223.5	1159.4	935.9	791.8	12.68	.07884
	33	255.7	225.3	.9	934.6	769.2	12.32	.08115
	34	257.5	227.1	1160.5	933.4	748.0	11.98	.08346
	35	259.2	228.8	1161.0	932.2	727.9	11.[illegible]	[illegible]
	36	260.8	230.5	1161.5	931.0	708.8	11[illegible]	[illegible]
	37	262.5	232.1	1162.0	929.8	690.8	1[illegible]	[illegible]

Properties of Saturated Steam.

Gauge Pressure, lbs. per sq. in.	Absolute Pressure, lbs. per square inch.	Temperature Fahrenheit.	Total Heat above 32° F.		Latent Heat L. $= H - h$. Heat-units.	Relative Volume. Vol. of Water at 39° F. = 1.	Volume. Cu. ft. in 1 lb. of Steam	Weight of 1 cu. ft. Steam, lb.
			In the Water h Heat-units.	In the Steam H Heat-units.				
23.3	38	264.0	233.8	1162.5	928.7	673.7	10.79	.09264
24.3	39	265.6	235.4	.9	927.6	657.5	10.53	.09493
25.3	40	267.1	236.9	1163.4	926.5	642.0	10.28	.09721
26.3	41	268.6	238.5	.9	925.4	627.3	10.05	.09949
27.3	42	270.1	240.0	1164.3	924.4	613.3	9.83	.1018
28.3	43	271.5	241.4	.7	923.3	599.9	9.61	.1040
29.3	44	272.9	242.9	1165.2	922.3	587.0	9.41	.1063
30.3	45	274.3	244.3	.6	921.3	574.7	9.21	.1086
31.3	46	275.7	245.7	1166.0	920.4	563.0	9.02	.1108
32.3	47	277.0	247.0	.4	919.4	551.7	8.84	.1131
33.3	48	278.3	248.4	.8	918.5	540.9	8.67	.1153
34.3	49	279.6	249.7	1167.2	917.5	530.5	8.50	.1176
35.3	50	280.9	251.0	.6	916.6	520.5	8.34	.1198
36.3	51	282.1	252.2	1168.0	915.7	510.9	8.19	.1221
37.3	52	283.3	253.5	.4	914.9	501.7	8.04	.1243
38.3	53	284.5	254.7	.7	914.0	492.8	7.90	.1266
39.3	54	285.7	256.0	1169.1	913.1	484.2	7.76	.1288
40.3	55	286.9	257.2	.4	912.3	475.9	7.63	.1311
41.3	56	288.1	258.3	.8	911.5	467.9	7.50	.1333
42.3	57	289.1	259.5	1170.1	910.6	460.2	7.38	.1355
43.3	58	290.3	260.7	.5	909.8	452.7	7.26	.1377
44.3	59	291.4	261.8	.8	909.0	445.5	7.14	.1400
45.3	60	292.5	262.9	1171.2	908.2	438.5	7.03	.1422
46.3	61	293.6	264.0	.5	907.5	431.7	6.92	.1444
47.3	62	294.7	265.1	.8	906.7	425.2	6.82	.1466
48.3	63	295.7	266.2	1172.1	905.9	418.8	6.72	.1488
49.3	64	296.8	267.2	.4	905.2	412.6	6.62	.1511
50.3	65	297.8	268.3	.8	904.5	406.6	6.53	.1533
51.3	66	298.8	269.3	1173.1	903.7	400.8	6.43	.1555
52.3	67	299.8	270.4	.4	903.0	395.2	6.34	.1577
53.3	68	300.8	271.4	.7	902.3	389.8	6.25	.1599
54.3	69	301.8	272.4	1174.0	901.6	384.5	6.17	.1621
55.3	70	302.7	273.4	.3	900.9	379.3	6.09	.1643
56.3	71	303.7	274.4	.6	900.2	374.3	6.01	.1665
57.3	72	304.6	275.3	.8	899.5	369.4	5.93	.1687
58.3	73	305.6	276.3	1175.1	898.9	364.6	5.85	.1709
59.3	74	306.5	277.2	.4	898.2	360.0	5.78	.1731
60.3	75	307.4	278.2	.7	897.5	355.5	5.71	.1753
61.3	76	308.3	279.1	1176.0	896.9	351.1	5.63	.1775
62.3	77	309.2	280.0	.2	896.2	346.8	5.57	.1797
63.3	78	310.1	280.9	.5	895.6	342.6	5.50	.1819
64.3	79	310.9	281.8	1176.8	895.0	338.5	5.43	.1840
65.3	80	311.8	282.7	1177.0	894.3	334.5	5.37	.1862
66.3	81	312.7	283.6	.3	893.7	330.6	5.31	.1884
67.3	82	313.5	284.5	.6	893.1	326.8	5.25	.1906
68.3	83	314.4	285.3	.8	892.5	323.1	5.18	.1928
69.3	84	315.2	286.2	1178.1	891.9	319.5	5.13	.1950
70.3	85	316.0	287.0	.3	891.3	315.9	5.07	[illegible]

Properties of Saturated Steam.

Absolute Pressure, lbs per square inch.	Temperature Fahrenheit.	Total Heat above 32° F. In the Water h Heat-units.	Total Heat above 32° F. In the Steam H Heat-units.	Latent Heat L. $= H - h$. Heat-units.	Relative Volume. Vol. of Water at 39° F. = 1.	Volume. Cu. ft. in 1 lb. of Steam	Weight of 1 cu. ft. Steam, lb.
86	316.8	287.9	1178.6	890.7	312.5	5.02	.1993
87	317.7	288.7	.8	890.1	309.1	4.96	.2015
88	318.5	289.5	1179.1	889.5	305.8	4.91	.2036
89	319.3	290.4	.3	888.9	302.5	4.86	.2058
90	320.0	291.2	.6	888.4	299.4	4.81	.2080
91	320.8	292.0	.8	887.8	296.3	4.76	.2102
92	321.6	292.8	1180.0	887.2	293.2	4.71	.2123
93	322.4	293.6	.3	886.7	290.2	4.66	.2145
94	323.1	294.4	.5	886.1	287.3	4.62	.2166
95	323.9	295.1	.7	885.6	284.5	4.57	.2188
96	324.6	295.9	1181.0	885.0	281.7	4.53	.2210
97	325.4	296.7	.2	884.5	279.0	4.48	.2231
98	326.1	297.4	.4	884.0	276.3	4.44	.2253
99	326.8	298.2	.6	883.4	273.7	4.40	.2274
100	327.6	298.9	.8	882.9	271.1	4.36	.2296
101	328.3	299.7	1182.1	882.4	268.5	4.32	.2317
102	329.0	300.4	.3	881.9	266.0	4.28	.2339
103	329.7	301.1	.5	881.4	263.6	4.24	.2360
104	330.4	301.9	.7	880.8	261.2	4.20	.2382
105	331.1	302.6	.9	880.3	258.9	4.16	.2403
106	331.8	303.3	1183.1	879.8	256.6	4.12	.2425
107	332.5	304.0	.4	879.3	254.3	4.09	.2446
108	333.2	304.7	.6	878.8	252.1	4.05	.2467
109	333.9	305.4	.8	878.3	249.9	4.02	.2489
110	334.5	306.1	1184.0	877.9	247.8	3.98	.2510
111	335.2	306.8	.2	877.4	245.7	3.95	.2531
112	335.9	307.5	.4	876.9	243.6	3.92	.2553
113	336.5	308.2	.6	876.4	241.6	3.88	.2574
114	337.2	308.8	.8	875.9	239.6	3.85	.2596
115	337.8	309.5	1185.0	875.5	237.6	3.82	.2617
116	338.5	310.2	.2	875.0	235.7	3.79	.2638
117	339.1	310.8	.4	874.5	233.8	3.76	.2660
118	339.7	311.5	.6	874.1	231.9	3.73	.2681
119	340.4	312.1	.8	873.6	230.1	3.70	.2703
120	341.0	312.8	.9	873.2	228.3	3.67	.2724
121	341.6	313.4	1186.1	872.7	226.5	3.64	.2745
122	342.2	314.1	.3	872.3	224.7	3.62	.2766
123	342.9	314.7	.5	871.8	223.0	3.59	.2788
124	343.5	315.3	.7	871.4	221.3	3.56	.2809
125	344.1	316.0	.9	870.9	219.6	3.53	.2830
126	344.7	316.6	1187.1	870.5	218.0	3.51	.2851
127	345.3	317.2	.3	870.0	216.4	3.48	.2872
128	345.9	317.8	.4	869.6	214.8	3.46	.2894
129	346.5	318.4	.6	869.2	213.2	3.43	.2915
130	347.1	319.1	.8	868.7	211.6	3.4[illegible]	[illegible]
131	347.6	319.7	1188.0	868.3	210.1	3.[illegible]	[illegible]
132	348.2	320.3	.2	867.9	208.6	3[illegible]	[illegible]
133	348.8	320.8	.3	867.5	207.1	[illegible]	[illegible]
134	349.4	321.5	.5	867.0	205.7	[illegible]	[illegible]

Properties of Saturated Steam.

Gauge Pressure, lbs. per sq. in.	Absolute Pressure, lbs. per square inch.	Temperature Fahrenheit.	Total Heat above 32° F. In the Water h Heat-units.	Total Heat above 32° F. In the Steam H Heat-units.	Latent Heat L. $= H - h$. Heat-units.	Relative Volume, Vol. of water at 39° F. = 1.	Volume, Cu. ft. in 1 lb. of Steam.	Weight of 1 cu. ft. [illegible]
120.3	135	350.0	322.1	1188.7	866.6	204.2	3.29	[illegible]
121.3	136	350.5	322.6	.9	866.2	202.8	3.27	[illegible]
122.3	137	351.1	323.2	1189.0	865.8	201 4	3.24	[illegible]
123.3	138	351.8	323.8	.2	865.4	200.0	3.22	[illegible]
124.3	139	352.2	324.4	.4	865.0	198.7	3.20	[illegible]
125.3	140	352.8	325.0	.5	864.6	197.3	3.18	[illegible]
126.3	141	353.3	325.5	.7	864.2	196.0	3.16	[illegible]
127.3	142	353.9	326.1	.9	863.8	194.7	3.14	[illegible]
128.3	143	354.4	326.7	1190.0	863.4	193.4	3.11	[illegible]
129.3	144	355.0	327.2	.2	863.0	192.2	3.09	[illegible]
130.3	145	355.5	327.8	.4	862.6	190.9	3.07	[illegible]
131.3	146	356.0	328.4	.5	862.2	189.7	3.05	[illegible]
132.3	[illegible]	356.6	328.9	.7	861.8	185.5	3.04	[illegible]
133.3	[illegible]	357.1	329.5	.9	861.4	187.3	3.02	[illegible]
134.3	[illegible]	357.6	330.0	1191.0	861.0	186.1	3.00	[illegible]
135.3	150	358.2	330.6	.2	860.6	184.9	2.98	[illegible]
136.3	151	358.7	331.1	.3	860.2	183.7	2.96	[illegible]
137.3	152	359.2	331.6	.5	859.9	182.6	2.94	[illegible]
138.3	153	359.7	332.2	.7	859.5	181.5	2.92	[illegible]
139.3	154	360.2	332.7	.8	859.1	180.4	2.91	[illegible]
140.3	155	360.7	333.2	1192.0	858.7	179.2	2.89	[illegible]
141.3	156	361.3	333.8	.1	858.4	178.1	2.87	[illegible]
142.3	157	361.8	334.3	.3	858.0	177.0	2.85	[illegible]
143.3	158	362.3	334.8	.4	857.6	175.0	2.84	[illegible]
144.3	159	362.8	335.3	.6	857.2	174.9	2.82	[illegible]
145.3	160	363.3	335.9	.7	856.9	173.9	2.80	[illegible]
146.3	161	363.8	336.4	.9	856.5	172.9	2.79	[illegible]
147.3	162	364.3	336.9	1193.0	856.1	171.9	2.77	[illegible]
148.3	163	364.8	337.4	.2	855.8	171.0	2.76	[illegible]
149.3	164	365.3	337.9	.3	855.4	170.0	2.74	[illegible]
150.3	165	365.7	338.4	.5	855.1	169.0	2.72	[illegible]
151.3	166	366.2	338.9	.6	854.7	168.1	2.71	[illegible]
152.3	167	366.7	339.4	.8	854.4	167.1	2.69	[illegible]
153.3	168	367.2	339.9	.9	854.0	166.2	2.68	[illegible]
154.3	169	367.7	340.4	1194.1	853.6	165.3	2.66	[illegible]
155.3	170	368.2	340.9	.2	853.3	164.3	2.65	[illegible]
156.3	171	368.6	341.4	.4	852.9	163.4	2.63	[illegible]
157.3	172	369.1	341.9	.5	852.6	162.5	2.62	[illegible]
158.3	173	369.6	342.4	.7	852.3	161.6	2.61	[illegible]
159.3	174	370.0	342.9	.8	851.9	160.7	2.59	[illegible]
160.3	175	370.5	343.4	.9	851.6	159.8	2.58	[illegible]
161.3	176	371.0	343.9	1195.1	851.2	158.9	2.56	[illegible]
162.3	177	371.4	344.3	.2	850.9	158.1	2.55	[illegible]
163.3	178	371.9	344.8	.4	850.5	157.2	2.54	[illegible]
164.3	179	372.4	345.3	.5	850.2	156.4	2.52	[illegible]
[illegible]	180	372.8	345.8	.7	849.9	155.6	2.51	[illegible]
[illegible]	[illegible]	373.3	346.3	.8	849.5	154.8	[illegible]	[illegible]
[illegible]	[illegible]	373.7	346.7	.9	849.2	154.0	[illegible]	[illegible]
[illegible]	[illegible]	374.2	347.2	1196.1	848.9	153.2	[illegible]	[illegible]

Properties of Saturated Steam.

Absolute Pressure, lbs. per square inch.	Temperature Fahrenheit.	Total Heat above 32° F.		Latent Heat L. $= H - h$. Heat-units.	Relative Volume. Vol. of water at 39° F. = 1.	Volume. Cu. ft. in 1 lb. of Steam	Weight of 1 cu. ft. Steam, lb.
		In the Water h Heat-units.	In the Steam H Heat-units.				
184	374.6	347.7	1196.2	848.5	152.4	2.46	.4066
185	375.1	348.1	.3	848.2	151.6	2.45	.4087
186	375.5	348.6	.5	847.9	150.8	2.43	.4108
187	375.9	349.1	.6	847.6	150.0	2.42	.4129
188	376.4	349.5	.7	847.2	149.2	2.41	.4150
189	376.9	350.0	.9	846.9	148.5	2.40	.4170
190	377.3	350.4	1197.0	846.6	147.8	2.39	.4191
191	377.7	350.9	.1	846.3	147.0	2.37	.4212
192	378.2	351.3	.3	845.9	146.3	2.36	.4233
193	378.6	351.8	.4	845.6	145.6	2.35	.4254
194	379.0	352.2	.5	845.3	144.9	2.34	.4275
195	379.5	352.7	.7	845.0	144.2	[illegible]	.4296
196	380.0	353.1	.8	844.7	143.5	[illegible]	.4317
197	380.3	353.6	.9	844.4	142.8	[illegible].31	.4337
198	380.7	354.0	1198.1	844.1	142.1	2.29	.4358
199	381.2	354.4	.2	843.7	141.4	2.28	.4379
200	381.6	354.9	.3	843.4	140.8	2.27	.4400
201	382.0	355.3	.4	843.1	140.1	2.26	.4420
202	382.4	355.8	.6	842.8	139.5	2.25	.4441
203	382.8	356.2	.7	842.5	138.8	2.24	.4462
204	383.2	356.6	.8	842.2	138.1	2.23	.4482
205	383.7	357.1	1199.0	841.9	137.5	2.22	.4503
206	384.1	357.5	.1	841.6	136.9	2.21	.4523
207	384.5	357.9	.2	841.3	136.3	2.20	.4544
208	384.9	358.3	.3	841 0	135.7	2.19	.4564
209	385.3	358.8	.5	840.7	135.1	2.18	.4585
210	385.7	359.2	.6	840.4	134.5	2.17	.4605
211	386.1	359.6	.7	840.1	133.9	2.16	.4626
212	386.5	360.0	.8	839 8	133.3	2.15	.4646
213	386.9	360.4	.9	839.5	132.7	2.14	.4667
214	387.3	360.9	1200.1	839.2	132.1	2.13	.4687
215	387.7	361.3	.2	838.9	131.5	2.12	.4707
216	388.1	361.7	.3	838.6	130.9	2.12	.4728
217	388.5	362.1	.4	838.3	130.3	2.11	.4748
218	388.9	362.5	.6	838.1	129.7	2.10	.4768
219	389.3	362.9	.7	837.8	129.2	2.09	.4788
220	389.7	362.2*	1200.8	838.6*	128.7	2.06	.4852
230	393.6	366.2	1202.0	835.8	123.3	1.98	.5061
240	397.3	370.0	1203.1	833.1	118.5	1.90	.5270
250	400.9	373.8	1204.2	830.5	114.0	1.83	.5478
260	404.4	377.4	1205.3	827.9	109.8	1.76	.5686
270	407.8	380.9	1206.3	825.4	105.9	1.70	.5894
280	411.0	384.3	1207.3	823.0	102.3	1.64	.6101
290	414.2	387.7	1208.3	820.6	99.0	1.585	.6308
300	417.4	390.9	1209.2	818.3	95.8	1.535	
350	432.0	406.3	1213.7	807.5	82.7	1.325	

...repancies at 205.3 lbs. gauge are due to the change fr...
...ures.

Properties of Saturated Steam.

Gauge Pressure, lbs per sq. in.	Absolute Pressure, lbs. per square inch.	Temperature Fahrenheit.	Total Heat above 32° F. In the Water h Heat-units.	Total Heat above 32° F. In the Steam H Heat-units.	Latent Heat $L = H - h$. Heat-units.	Relative Volume, Vol. of water at 39° F. = 1.	Volume, Cu. ft. of Steam in 1 lb.
385.3	400	444.9	419.8	1217.7	797.9	72.8	1.167
435.3	450	456.6	432.2	1221.3	789.1	65.1	1.042
485.3	500	467.4	443.5	1224.5	781.0	58.8	.942
535.3	550	477.5	454.1	1227.6	773.5	52.6	.859
585.3	600	486.9	464.2	1230.5	766.3	49.3	.790
635.3	650	495.7	473.6	1233.2	759.6	45.6	.731
685.3	700	504.1	482.4	1235.7	753.3	42.4	.680
735.3	750	512.1	490.9	1238.0	747.2	39.6	.636
785.3	800	519.6	498.9	1240.3	741.4	37.1	.597
835.3	850	526.8	506.7	1242.5	735.8	34.9	.563
885.3	900	533.7	514.0	1244.7	730.6	33.0	.531
935.3	950	540.3	521.3	1246.7	725.4	31.4	.501
985.3	1000	546.8	528.3	1248.7	720.3	30.0	.480

FLOW OF STEAM.

Flow of Steam through a Nozzle. (From Clark on the S
engine.)—The flow of steam of a greater pressure into an atmosphere
less pressure increases as the difference of pressure is increased, u
external pressure becomes only 58% of the absolute pressure in the
The flow of steam is neither increased nor diminished by the fall of
ternal pressure below 58%, or about 4/7ths of the inside pressure, even
extent of a perfect vacuum. In flowing through a nozzle of the best
the steam expands to the external pressure, and to the volume due
pressure, so long as it is not less than 58% of the internal pressure.
external pressure of 58%, and for lower percentages, the ratio of exp
is 1 to 1.624. The following table is selected from Mr. Brownlee's data
plifying the rates of discharge under a constant internal pressure
various external pressures:

Outflow of Steam; from a Given Initial Pressure i Various Lower Pressures.

Absolute initial pressure in boiler, 75 lbs. per sq. in.

Absolute Pressure in Boiler per square inch.	External Pressure per square inch.	Ratio of Expansion in Nozzle.	Velocity of Outflow at Constant Density.	Actual Velocity of Outflow Expanded.
lbs.	lbs.	ratio.	feet per sec.	feet p. sec.
75	74	1.012	227.5	230
75	72	1.037	386.7	401
75	70	1.063	490	521
75	65	1.136	660	749
75	61.62	1.198	736	876
	60	1.219	765	983
	50	1.434	873	1252
	45	1.575	890	1401
	43.46 (58 p. cent)	1.624	890.6	1446.5
		1.624	890.6	1446.5
		1.624	890.6	1446.5

eam of varying initial pressures is discharged into the atmos- atmospheric pressure being not more than 58% of the initial he velocity of outflow at constant density, that is, supposing the ity to be maintained, is given by the formula $V = 3.5953\sqrt{h}$.

locity of outflow in feet per minute, as for steam of the initial ity;
ight in feet of a column of steam of the given absolute initial sure of uniform density, the weight of which is equal to the pres- on the unit of base.

st initial pressure to which the formula applies, when the steam ed into the atmosphere at 14.7 lbs. per square inch, is (14.7 × 5.37 lbs. per square inch. Examples of the application of the e given in the table below.

contents of this table it appears that the velocity of outflow into here, of steam above 25 lbs. per square inch absolute pressure, ffective, increases very slowly with the pressure, obviously be- ensity, and the weight to be moved, increase with the pressure. e of 900 feet per second may, for approximate calculations, be he velocity of outflow as for constant density, that is, taking the the steam at the initial volume.

v of Steam into the Atmosphere.—External pressure inch 14.7 lbs. absolute. Ratio of expansion in nozzle, 1.624.

stant Density.	Actual Velocity of Outflow Expanded.	Discharge per square inch of Orifice per min	Horse-power per sq. in. of Orifice if H. P. = 30 lbs. per hour.	Absolute Initial Pressure per square inch.	Velocity of Outflow as at Constant Density.	Actual Velocity of Outflow Expanded.	Discharge per square inch of Orifice per minute.	Horse-power per sq. in. of Orifice if H. P. = 30 lbs. per hour.
et ec.	feet per sec.	lbs.	H.P.	lbs.	feet p.sec.	feet per sec.	lbs.	H.P.
63	1401	22.81	45.6	90	895	1454	77.94	155.9
67	1408	26.84	53.7	100	898	1459	86.34	172.7
74	1419	35.18	70.4	115	902	1466	98.76	197.5
80	1429	44.06	88.1	135	906	1472	115.61	231.2
85	1437	52.59	105.2	155	910	1478	132.21	264.4
89	1444	61.07	122.1	165	912	1481	140.46	280.9
91	1447	65.30	130.6	215	919	1493	181.58	363.2

s Approximate Rule.—Flow in pounds per second = ab- sure × area in square inches ÷ 70. This rule gives results which espond with those in the above table, as shown below.

lbs. p. sq. in.	25.37	40	60	75	100	135	165	215
per min., by								
....	22.81	35.18	52.59	65.30	86.34	115.61	140.46	181.58
rule.........	21.74	34.29	51.43	64.29	85.71	115.71	141.43	184.29

body, in Trans. A. S. M. E., xi, 187, reports a series of experi- w of steam through tubes ¼ inch in diameter, and ¼, ½, and 1½ ith rounded entrances, in which the results agreed closely with mula, the greatest difference being an excess of the experimental culated result of 3.2%. An equation derived from the theory of mics is given by Prof. Peabody, but it does not agree with the al results as well as Napier's rule, the excess of the actual flow

f Steam in Pipes.—A formula commonly used for velocity eam in pipes is the same as Downing's for the flow of water in -iron pipes, viz., $V = 50\sqrt{\frac{H}{L}D}$, in which V = velocity in feet

L = length and D = diameter of pipe in feet, [umn of steam, of the pressure of the steam at which would produce a pressure equal to the difference of pressure at two ends of the pipe. (For derivation of the coefficient 50, see Briggs "Warming Buildings by Steam," Proc. Inst. C. E. 1882.)

If Q = quantity in cubic feet per minute, d = diameter in inches, L being in feet, the formula reduces to

$$Q = 4.7233\sqrt{\frac{H}{L}d^5},\quad H = .0448\frac{Q^2L}{d^5},\quad d = .5374\sqrt[5]{\frac{Q^2L}{H}}.$$

(These formulæ are applicable to air and other gases as well as steam.)

If p_1 = pressure in pounds per square inch of the steam (or gas) at the entrance to the pipe, p_2 = the pressure at the exit, then $144(p_1 - p_2)$ = difference in pressure per square foot. Let w = density or weight per cubic foot of steam at the pressure p_1, then the height of column equivalent to difference in pressures

$$= H = \frac{144(p_1 - p_2)}{w},\quad \text{and}\quad Q = 60 \times .7854 \times 50D^2\sqrt{\frac{144(p_1 - p_2)D}{wL}}.$$

If W = weight of steam flowing in pounds per minute $= Qw$, and d taken in inches, L being in feet,

$$W = 56.68\sqrt{\frac{w(p_1 - p_2)d^5}{L}};\quad Q = 56.68\sqrt{\frac{(p_1 - p_2)d^5}{Lw}};$$

$$d = 0.199\sqrt[5]{\frac{W^2L}{w(p_1 - p_2)}} = 0.199\sqrt[5]{\frac{Q^2wL}{p_1 - p_2}}.$$

Velocity in feet per minute $= V = Q \div .7854\frac{d^2}{144} = 10392\sqrt{\frac{(p_1 - p_2)d}{wL}}.$

For a velocity of 6000 feet per minute, $d = \frac{wL}{3(p_1 - p_2)}$; $p_1 - p_2 = \frac{wL}{3d}$.

For a velocity of 6000 feet per minute, a steam-pressure of 100 lbs. gauge, or $w = .264$, and a length of 100 feet, $d = \frac{8.8}{p_1 - p_2}$; $p_1 - p_2 = \frac{8.8}{d}$. That is, a pipe 1 inch diameter, 100 feet long, carrying steam of 100 lbs. gauge-pressure at 6000 feet velocity per minute, would have a loss of pressure of 8.8 lbs. per square inch, while steam travelling at the same velocity in a pipe 8.8 inches diameter would lose only 1 lb. pressure.

G. H. Babcock, in "Steam," gives the formula

$$W = 87\sqrt{\frac{w(p_1 - p_2)d^5}{L\left(1 + \frac{3.6}{d}\right)}}.$$

In earlier editions of "Steam" the coefficient is given as 300,—evidently an error,—and this value has been reprinted in Clark's Pocket-Book (1892 edition). It is apparently derived from one of the numerous formulæ for flow of water in pipes, the multiplier of L in the denominator being used for the expression of the increased resistance of small pipes. Putting this formula in the form $W = c\sqrt{\frac{w(p_1 - p_2)d^5}{L}}$, in which c will vary with the diameter of the pipe, we have,

For diameter, inches....	1	2	3	4	6	9
Value of c................	40.7	52.1	58.8	63	68.8	73.7

instead of the constant value 56.68, given with the simpler formula.

One of the most widely accepted formulæ for flow of water is D'Arcy's

$\sqrt{\frac{HD}{4}}$, in which c has values ranging from 65 for a ½-inch pipe up

24-inch. Using D'Arcy's coefficients, and modifying his formula to apply to steam, to the form

$$Q = c\sqrt{\frac{(p_1 - p_2)d^5}{wL}}, \quad \text{or} \quad W = c\sqrt{\frac{w(p_1 - p_2)d^5}{L}},$$

n,

eter, inches....	½	1	2	3	4	5	6	7	8
c................	36.8	45.3	52.7	56.1	57.8	58.4	59.5	60.1	60.7
eter, inches....	9	10	12	14	16	18	20	22	24
c................	61.2	61.8	62.1	62.3	62.6	62.7	62.9	63.2	63.2

absence of direct experiments these coefficients are probably as as any that may be derived from formulæ for flow of water.

f pressure in lbs. per sq. in. $= p_1 - p_2 = \frac{Q^2wL}{c^2d^5}$.

of Pressure due to Radiation as well as Friction.—udiger (*Mechanics*, June 30, 1883) gives the following formulæ and r flow of steam in pipes. He takes into consideration the losses in due both to radiation and to friction.

f power, expressed in heat-units due to friction, $Hf = \frac{W^2fl}{10p^2d^5}$.

ue to radiation, $Hr = 0.262rld.$

W is the weight in lbs. of steam delivered per hour, f the coefficient of friction of the pipe, l the length of the pipe in feet, p the absolute pressure, d the diameter of the pipe in inches, and r the coefficient tion. f is taken as from .0165 to .0175, and r varies as follows:

TABLE OF VALUES FOR r.

Pipe Covering.	Absolute Pressure.			
	40 lbs.	65 lbs.	90 lbs.	115 lbs.
ed pipe	437	555	620	684
ment composition..........	146	178	193	209
bestos "	157	192	202	222
bestos flock................	150	185	197	210
ooden log..................	100	122	145	151
ineral wool................	61	76	85	93
air felt....................	48	58	66	73

pended table shows the loss due to friction and radiation in a steam- ere the quantity of steam to be delivered is 1000 lbs. per hour, l = , the pipe being so protected that loss by radiation $r = 64$, and the terminal pressure being 90 lbs.:

r ,	Loss by Friction, Hf.	Loss by Radiation, Hr.	Total Loss, L.	Diam. of Pipe, inches.	Loss by Friction, Hf.	Loss by Radiation, Hr.	Total Loss, L.
	197,531	16,768	214,300	3½	376	58,688	59,064
	64,727	20,960	85,687	4	193	67,072	67,265
	26,012	25,152	51,164	5	63	[illegible]	[illegible]
	12,035	29,344	41,379	6	25	[illegible]	
	6,173	33,536	39,709	7	12	[illegible]	
	2,023	41,920	*43,943*	8	6	[illegible]	
	813	*50,304*	*51,117*				

If the pipes are carrying steam with minimum loss, then for sa[illegible]
and p, the loss of pressure L for pipes of different diameters va[illegible]
versely as the diameters.

The general equation for the loss of pressure for the minimal lo[illegible]
friction and radiation is

$$L = \frac{0.000[illegible]\; d[illegible]p}{W}.$$

The loss of pressure for pipes of 1 inch diameter for different [illegible]
terminal pressures when steam is flowing with minimal loss is expre[illegible]
the formula $L = Cl\sqrt{r^2}$, in which the coefficient C has the following [illegible]

For 65 lbs. abs. term. pressure........................C = 0.000[illegible]
" 75 " " " " .... 0.000[illegible]
" 90 " " " " .... 0.000[illegible]
" 100 " " " " .... 0.000[illegible]
" 115 " " " " .... 0.000[illegible]

In order to find the loss of pressure for any other diameter, divide [illegible]
of pressure in a 1-inch pipe for the given terminal pressure by th[illegible]
diameter, and the quotient will be the loss of pressure for that diam[illegible]

The following is a general summary of the results of Mr. Rudiger's [illegible]
tigation:

The flow of steam in a pipe is determined in the same manner as t[illegible]
of water, the formula for the flow of steam being modified only by [illegible]
tuting the equivalent loss of pressure, divided by the density of the [illegible]
for the loss of head.

The losses in the flow of steam are two in number—the loss du[illegible]
friction of flow and that due to radiation from the sides of the pip[illegible]
sum of these is a minimum when the equivalent of the loss due [illegible]
tion of flow is equal to one fifth of the loss of heat by radiation. [illegible]
greater or less loss of pressure—i.e., for a less or greater diameter [illegible]
—the total loss increases very rapidly.

For delivering a given quantity of steam at a given terminal p[illegible]
with minimal total loss, the better the non-conducting material u[illegible]
the larger the diameter of the steam-pipe to be used.

The most economical loss of pressure for a pipe of given diameter [illegible]
to the most economical loss of pressure in a pipe of 1 inch diameter [illegible]
conditions, divided by the diameter of the given pipe in inches.

The following table gives the capacity of pipes of different diam[illegible]
deliver steam at different terminal pressures through a pipe one h[illegible]
long for loss of pressure of 10 lbs., and a mean value of $f = 0.0175$, [illegible]
denote the number of pounds of steam delivered per hour:

Diameter of Pipe, inches.	Abs. Term. Pressure.			Diameter of Pipe, inches.	Abs. Term. Pr[illegible]	
	65 lbs.	80 lbs.	100 lbs.		65 lbs.	80 lbs.
	W	W	W		W	W
1	102	113	125	4½	4,397	4,872
1¼	179	198	219	5	5,721	[illegible]
1½	282	312	346	6	9,024	10,000
1¾	415	459	508	7	13,268	14,701
2	579	641	710	8	18,526	[illegible]
2½	1,011	1,121	1,240	9	24,870	27,556
3	1,595	1,768	1,956	10	32,364	35,860
3½	2,346	2,599	2,875	11	41,081	45,507
4	3,275	3,629	4,042	12	51,049	[illegible]

Resistance to Flow by Bends, Valves, etc. (From B[illegible]
[illegible]ing Buildings by Steam.)—The resistance at the entrance t[illegible]
special bel[illegible] [illegible]iven consists of two parts. The hea[illegible]
[illegible]d in g[illegible] [illegible] of flow; and the head 0.505 [illegible]

sistance of the mouth of the tube. Hence the whole loss of ıtrance is $1.505 \frac{v^2}{2g}$. This resistance is equal to the resistance tube of a length equal to about 60 times its diameter.

t each sharp right-angled elbow is the same as in flowing gth of straight tube equal to about 40 times its diameter. For stop-valve the resistance is taken to be 1½ times that of the lbow.

Steam-pipes for Stationary Engines.—Authorities -engine generally agree that steam-pipes supplying engines such size that the mean velocity of steam in them does not et per minute, in order that the loss of pressure due to friction ccessive. The velocity is calculated on the assumption that the ed at each stroke. In very long pipes, 100 feet and upward, it e them larger than this rule would give, and to place a large r on the pipe near the engine, especially when the engine cuts e stroke.

ı *Power*, May, 1893, on proper area of supply-pipes for engines showing the practice of leading builders. To facilitate com- e engines have been rated in horse-power at 40 pounds mean sure. The table contains all the varieties of simple engines, -valve to the Corliss, and it appears that there is no general he sizes of pipe used in the different types.

s selected from this table are as follows:

, in......	2	2½	3	3½	4	4½	5	6	7	8	9	10
gines....	25	39	56	77	100	126	156	225	306	400	506	625
rmula (1)	23	36	51	70	91	116	143	206	278	366	463	571
rmula (2)	24	37.5	54	73	96	121	150	216	294	384	486	600

is: 1 H P. requires .1375 sq. in. of steam-pipe area.

is: Horse-power $= 6d^2$. d = diam. of pipe in inches.

.1375 in formula (1) is thus derived: Assume that the linear eam in the pipe should not exceed 6000 feet per minute, then yl. area × piston-speed ÷ 6000 (a). Assume that the av. mean sure is 40 lbs. per sq. in., then cyl. area × piston-speed × 40 ÷ -power (b). Dividing (a) by (b) and cancelling, we have pipe = .1375 sq. in. If we use 8000 ft. per min. as the allowable the factor .1375 becomes .1031; that is, pipe area ÷ H.P. = area × .97 = horse-power. This, however, gives areas of pipe re used in the most recent practice. A formula which gives agreeing with practice, as shown in the above table is

$$\text{ower} = 6d^2, \text{ or pipe diameter} = \sqrt{\frac{\text{H.P.}}{6}} = .408\sqrt{\text{H.P.}}$$

F CYLINDERS CORRESPONDING TO VARIOUS SIZES OF STEAM- ED ON PISTON-SPEED OF ENGINE OF 600 FT. PER MINUTE, AND E MEAN VELOCITY OF STEAM IN PIPE OF 4000, 6000, AND 8000 IN. (STEAM ASSUMED TO BE ADMITTED DURING FULL STROKE.)

, Inches.........	2	2½	3	3½	4	4½	5	6
	5.2	6.5	7.7	9.0	10.3	11.6	12.9	15.5

plan is followed, the pipe fills with water whenever this boiler is sto and the others are running, and breakage of the pipe may cause serio sults. Never let a junction-pipe run into the bottom of the main pipe into the side or top. Always use an angle-valve where convenient, as is more room in them. Never use a gate valve under high pressure un by-pass is used with it. Never open a blow-off valve on a boiler a littl then shut it; it is sure to catch the sediment and ruin the valve; thr well open before closing. Never use a globe-valve on an indicator-pipe water, always use gate or angle valves or stop-cocks to obtain a clear sage. Buy if possible valves with renewable disks. Lastly, never let a go inside a boiler to work, especially if he is to hammer on it, unless break the joint between the boiler and the valve and put a plate of between the flanges.

Flanges for Steam-nozzles and Steam-pipe, used wit Gill Water-tube Boiler, Phila., 1892.

Size of pipe	3	4	5	6	7	8
Outside diameter of flange, inches	9	10	11	12	13	14
Pitch-circle for bolts, diam., "	7	8	9	10	11	12
Outside diam. of gaskets, "	5½	6½	7½	8½	9½	10½
Inside diam. of gaskets, "	3½	4½	5½	6½	7½	8½
Number of bolts	5	6	7	8	9	10
Size of pipe	10	11	12	13	14	15
Outside diameter of flange, inches	16	17	18	19	20	21
Pitch-circle for bolts, diam., "	14	15	16	17	18	19
Outside diam. of gaskets, "	12½	13½	14½	15½	16½	17½
Inside diam. of gaskets, "	10½	11½	12½	13½	14½	15½
Number of bolts	12	13	14	15	16	17

All holes drilled 15/16 in., with a jig accurately laid out.
All bolts to be ⅞ in. diam. by 3½ in. long under the head.
All bolts to have square heads and hexagon nuts.

The "Steam Loop" is a system of piping by which water of densation in steam-pipes is automatically returned to the boiler. simplest form it consists of three pipes, which are called the riser, the zontal, and the drop-leg. When the steam-loop is used for returning to boiler the water of condensation and entrainment from the steam through which the steam flows to the cylinder of an engine, the riser is erally attached to a separator; this riser empties at a suitable height the horizontal, and from thence the water of condensation is led into drop-leg, which is connected to the boiler, into which the water of conde tion is fed as soon as the hydrostatic pressure in drop-leg in connection the steam-pressure in the pipes is sufficient to overcome the boiler-press The action of the device depends on the following principles: Differen pressure may be balanced by a water-column; vapors or liquids tend to to the point of lowest pressure; rate of flow depends on difference of p sure and mass; decrease of static pressure in a steam-pipe or chamb proportional to rate of condensation; in a steam-current water will be ried or swept along rapidly by friction. (Illustrated in Modern Mecha p. 807.)

Loss from an Uncovered Steam-pipe. (Bjorling on Pum engines.)—The amount of loss by condensation in a steam-pipe carried a deep mine-shaft has been ascertained by actual practice at the Clay C Colliery, near Chesterfield, where there is a pipe 7½ in. internal diam ft. long. The loss of steam by condensation was ascertained by measurement of the water deposited in a receiver, and was found equivalent to about 1 lb. of coal per I.H.P. per hour for every 10 steam-pipe; but there is no doubt that if the pipes had been in the u shaft, and well covered with a good non-conducting material, the loss have been less. (For Steam-pipe Coverings, see p. 469, *ante*.)

THE STEAM-BOILER.

e **Horse-power of a Steam-boiler.**—The term horse power o meanings in engineering: *First, an absolute unit or measure of the f work*, that is, of the work done in a certain definite period of time, ource of energy, as a steam-boiler, a waterfall, a current of air or or by a prime mover, as a steam-engine, a water-wheel, or a wind- The value of this unit, whenever it can be expressed in foot-pounds rgy, as in the case of steam-engines, water-wheels, and waterfalls, is foot-pounds per minute. In the case of boilers, where the work done, nversion of water into steam, cannot be expressed in foot-pounds of ble energy, the usual value given to the term horse-power is the evap- n of 30 lbs. of water of a temperature of 100° F. into steam at 70 lbs. re above the atmosphere. Both of these units are arbitrary; the first, foot-pounds per minute, first adopted by James Watt, being considered lent to the power exerted by a good London draught-horse, and the of water evaporated per hour being considered to be the steam re- nent per indicated horse-power of an average engine.

second definition of the term horse-power *is an approximate measure size, capacity, value, or "rating"* of a boiler, engine, water-wheel, or source or conveyer of energy, by which measure it may be described, t and sold, advertised, etc. No definite value can be given to this re, which varies largely with local custom or individual opinion of s and users of machinery. The nearest approach to uniformity which arrived at in the term "horse-power," used in this sense, is to say boiler, engine, water-wheel, or other machine, "rated" at a certain power, should be capable of steadily developing that horse-power for period of time under ordinary conditions of use and practice, leaving l custom, to the judgment of the buyer and seller, to written contracts chase and sale, or to legal decisions upon such contracts, the interpre- of what is meant by the term "ordinary conditions of use and ce." (Trans. A. S. M. E., vol. vii. p. 226.)

committee of the A. S. M. E. on Trials of Steam-boilers in 1884 (Trans., p. 265) discussed the question of the horse-power of boilers as follows: Committee of Judges of the Centennial Exhibition, to whom the trials peting boilers at that exhibition were intrusted, met with this same m, and finally agreed to solve it, at least so far as the work of that ittee was concerned, by the adoption of the unit, 30 lbs. of water evap- l into dry steam per hour from feed-water at 100° F., and under a re of 70 lbs. per square inch above the atmosphere, these conditions considered by them to represent fairly average practice. The quan- heat demanded to evaporate a pound of water under these conditions 0.2 British thermal units, or 1.1496 units of evaporation. The unit of proposed is thus equivalent to the development of 33,305 heat units ur, or 34.488 units of evaporation. . . .

r committee, after due consideration, has determined to accept the nnial Standard, the first above mentioned, and to recommend that in ndard trials the commercial horse-power be taken as an evaporation bs. of water per hour from a feed-water temperature of 100° F. into at 70 lbs. gauge pressure, which shall be considered to be equal to 34½ f evaporation, that is, to 34½ lbs. of water evaporated from a feed- temperature of 212° F. into steam at the same temperature. This rd is equal to 33,305 thermal units per hour.

the opinion of this committee that a boiler rated at any stated number se-powers should be capable of developing that power with easy firing, ate draught, and ordinary fuel, while exhibiting good economy; and r, that the boiler should be capable of developing at least one third han its rated power to meet emergencies at times when maximum ny is not the most important object to be attained.

t of Evaporation.—It is the custom to reduce results of boiler- o the common standard of weight of water evaporated by the unit of the combustible portion of the fuel, the evaporation being consid- have taken place at mean atmospheric pressure, and at the temper- lue that pressure, the feed-water being also assumed to have been d at that temperature. *This is, in technical language, said ent evaporation from and at the boiling-point at atmosp "from and at 212° F."* This unit of evaporation, or on water evaporated from and at 212°, is equivalent to 965.7 British thermal units.

Measures for Comparing the Duty of Boilers.—The measure of the efficiency of a boiler is the number of pounds of water evaporated per pound of combustible, the evaporation being reduced to the standard "from and at 212°;" that is, the equivalent evaporation from feed-water at a temperature of 212° F. into steam at the same temperature.

The measure of the capacity of a boiler is the amount of "boiler horse-power" developed, a horse-power being defined as the evaporation of 30 lbs. of water per hour from 100° F. into steam at 70 lbs. pressure, or 34½ lbs. per hour from and at 212°.

The measure of relative rapidity of steaming of boilers is the number of pounds of water evaporated per hour per square foot of water-heating surface.

The measure of relative rapidity of combustion of fuel in boiler-furnaces is the number of pounds of coal burned per hour per square foot of grate surface.

STEAM-BOILER PROPORTIONS.

Proportions of Grate and Heating Surface required for a given Horse-power.—The term horse-power here means capacity to evaporate 30 lbs. of water from 100° F., temperature of feed-water, to steam of 70 lbs., gauge-pressure = 34.5 lbs. from and at 212° F.

Average proportions for maximum economy for land boilers fired with good anthracite coal:

Heating surface per horse-power	11.5	sq. ft.
Grate " "	1/3	"
Ratio of heating to grate surface	34.5	"
Water evap'd from and at 212° per sq. ft. H.S. per hour	3	lbs.
Combustible burned per H.P. per hour	3	"
Coal with 1/6 refuse, lbs. per H.P. per hour	3.6	"
Combustible burned per sq. ft. grate per hour	9	"
Coal with 1/6 refuse, lbs. per sq. ft. grate per hour	10.8	"
Water evap'd from and at 212° per lb. combustible	11.5	"
" " " " " " " " coal (1/6 refuse)	9.6	"

The rate of evaporation is most conveniently expressed in pounds evaporated from and at 212° per sq. ft. of water-heating surface per hour, and rate of combustion in pounds of coal per sq. ft. of grate-surface per hour.

Heating-surface.—For maximum economy with any kind of fuel a boiler should be proportioned so that at least one square foot of heating-surface should be given for every 3 lbs. of water to be evaporated from and at 212° F. per hour. Still more liberal proportions are required if a portion of the heating-surface has its efficiency reduced by: 1. Tendency of the heated gases to short-circuit, that is, to select passages of least resistance and flow through them with high velocity, to the neglect of other passages. 2. Deposition of soot from smoky fuel. 3. Incrustation. If the heating-surfaces are clean, and the heated gases pass over it uniformly, little if any increase in economy can be obtained by increasing the heating-surface beyond the proportion of 1 sq. ft. to every 3 lbs. of water to be evaporated, and with all conditions favorable but little decrease of economy will take place if the proportion is 1 sq. ft. to every 4 lbs. evaporated; but in order to provide for driving of the boiler beyond its rated capacity, and for possible decrease of efficiency due to the causes above named, it is better to adopt 1 sq. ft. to 3 lbs. evaporation per hour as the minimum standard proportion.

Where economy may be sacrificed to capacity, as where fuel is very cheap, it is customary to proportion the heating-surface much less liberally. The following table shows approximately the relative results that may be expected with different rates of evaporation, with anthracite coal.

Lbs. water evapor'd from and at 212° per sq. ft. heating-surface per hour:

2	2.5	3	3.5	4	5	6	7	8	9

Sq. ft. heating-surface required per horse-power:

17.3	13.8	11.5	9.8	8.6	6.8	5.8	4.9	4.3	3.8

Ratio of heating to grate surface if 1/3 sq. ft. of G. S. is required per H.P.:

52	41.4	34.5	29.4	25.8	20.4	17.4	13.7	12.9	11.4

Probable relative economy:

100	100	100	95	90	85	80	75	70	65

Probable temperature of chimney gases, degrees F.:

[illegible]	450	[illegible]	[illegible]	585	652	720	787	[illegible]	[illegible]

economy will vary not only with the amount of heating-sur- -power, but with the efficiency of that heating-surface as acity for transfer of heat from the heated gases to the water, end on its freedom from soot and incrustation, and upon the the water and the heated gases.

ous coal the efficiency will largely depend upon the thorough- h the combustion is effected in the furnace.

y with any kind of fuel will greatly depend upon the amount l to the furnace in excess of that required to support com- strong draught and thin fires this excess may be very great, us loss of economy.

nent of Heating-surface.—Authorities are not agreed ods of measuring the heating-surface of steam-boilers. The o consider as heating-surface all the surfaces that are sur- ter on one side and by flame or heated gases on the other, but rence of opinion as to whether tubular heating-surface should n the inside or from the outside diameter. Some writers say, eating-surface always on the smaller side—the fire side of the ontal return tubular boiler and the water side in a water-tube s would deduct from the heating-surface thus measured an portions supposed to be ineffective on account of being cov- or being out of the direct current of the gases.

of uniformity, however, it would appear to be the best method l surfaces as heating-surfaces which transmit heat from the to the water, making no allowance for different degrees of also, to use the external instead of the internal diameter greater convenience in calculation, the external diameter of ually being made in even inches or half inches. There would good reason for considering the smaller surface in a tube as rface, for the transmission of heat through plates that are ugated on one side does not appear to be proportional to the e, but rather to the larger. Thus the Serve ribbed tube trans- t to the water per foot of length than a plain tube of same eter, and a ribbed steam-radiator radiates more heat than a having the same internal or smaller surface.

ding the heating-surface of vertical tubular boilers: Multiply ence of the fire-box (in inches) by its height above the grate; ombined circumference of all the tubes by their length, and roducts add the area of the lower tube-sheet; from this sum rea of all the tubes, and divide by 144: the quotient is the are feet of heating-surface.

ding the heating-surface of horizontal tubular boilers: Take s in inches. Multiply two thirds of the circumference of the gth; multiply the sum of the circumferences of all the tubes on length; to the sum of these products add two thirds of the be-sheets; from this sum subtract twice the combined area of livide the remainder by 144 to obtain the result in square feet.

ding the square feet of heating-surface in tubes: Multiply the es by the diameter of a tube in inches, by its length in feet,

wer, Builder's Rating. Heating-surface per er.—It is a general practice among builders to furnish about of heating-surface per horse-power, but as the practice is not and contracts should always specify the amount of heating- urnished. Not less than one third square foot of grate-surface ished per horse-power.

News, July 5, 1894, gives the following rough-and-ready rule proximately the commercial horse-power of tubular or water- Number of tubes × their length in feet × their nominal ches $\div$ 50 $= nLd \div 50$. The number of square feet of surface $\frac{n\pi dL}{12} = \frac{nLd}{3.82}$, and the horse-power at 12 square feet of surface orse-power, not counting the shell, $= nLd \div 45.8$. If 15 square e of tubes be taken, it is $nLd \div 57.3$. Making allowance for rface in the shell will reduce the divisor to about 50.

wer of Marine and Locomotive Boilers.—The wer is not *generally* used in connection with boilers in ... *th locomotives.* The boilers are designed to suit the *y extent of grate* and heating-surface only.

Grate-surface.—The amount of grate-surface required [illegible]
power, and the proper ratio of heating-surface to grate-surface [illegible]
tremely variable, depending chiefly upon the character of the coal [illegible]
the rate of draught. With good coal, low in ash, approximately equ[illegible]
may be obtained with large grate-surface and light draught and [illegible]
grate-surface and strong draught, the total amount of coal burned
being the same in both cases. With good bituminous coal, like [illegible]
low in ash, the best results apparently are obtained with stro[illegible]
and high rates of combustion, provided the grate-surfaces are [illegible]
that the total coal burned per hour is not too great for the capa[illegible]
heating-surface to absorb the heat produced.

With coals high in ash, especially if the ash is easily fusible, [illegible]
choke the grates, large grate-surface and a slow rate of comb[illegible]
required, unless means, such as shaking grates, are provided to [illegible]
the ash as fast as it is made.

The amount of grate-surface required per horse-power under [illegible]
ditions may be estimated from the following table:

	Lbs. Water from and at 212° per lb., Coal.	Lbs. Coal per H.P. per hour.	Pounds of Coal burned per s[illegible] of Grate per hour						[illegible]
			8	10	12	15	20	25	[illegible]
			Sq. Ft. Grate per H[illegible]						
Good coal and boiler,	10	3.45	.43	.35	.28	.23	.17	.14	[illegible]
	9	3.83	.48	.38	.32	.25	.19	.15	[illegible]
Fair coal or boiler,	8.61	4.	.50	.40	.33	.26	.20	.16	[illegible]
	8	4.31	.54	.43	.36	.29	.22	.17	[illegible]
	7	4.93	.62	.49	.41	.33	.24	.20	[illegible]
Poor coal or boiler,	6.9	5.	.63	.50	.42	.34	.25	[illegible]	[illegible]
	6	5.75	.72	.58	.48	.38	.29	[illegible]	[illegible]
	5	6.9	.86	.69	.58	.46	.35	.28	[illegible]
Lignite and poor boiler,	3.45	10.	1.25	1.00	.83	.67	.50	.40	[illegible]

In designing a boiler for a given set of conditions, the grate-sur[illegible]
be made as liberal as possible, say sufficient for a rate of combu[illegible]
lbs. per square foot of grate for anthracite, and 15 lbs. per squa[illegible]
bituminous coal, and in practice a portion of the grate-surfa[illegible]
bricked over if it is found that the draught, fuel, or other condit[illegible]
it advisable.

Proportions of Areas of Flues and other Gas-p[illegible]
—Rules are usually given making the area of gas-passages bea[illegible]
ratio to the area of the grate-surface; thus a common rule for
tubular boilers is to make the area over the bridge wall 1/7 of
surface, the flue area 1/8, and the chimney area 1/9.

For average conditions with anthracite coal and moderate dra[illegible]
rate of combustion of 12 lbs. coal per square foot of grate per hour,
of heating to grate surface of 30 to 1, this rule is as good as any, [illegible]
dent that if the draught were increased so as to cause a rate of [illegible]
of 24 lbs., requiring the grate-surface to be cut down to a ratio of
areas of gas-passages should not be reduced much, because the
face is reduced. The coal burned being the same under the cha[illegible]
tions, and there being no reason why the gases should travel [illegible]
velocity, the actual areas of the passages should remain as befo[illegible]
ratio of the area to the grate-surface would in that case be doub[illegible]

Mr. Barrus states that the highest efficiency with anthrac[illegible]
obtained when the tube area is 1/9 to 1/10 of the grate-surface
bituminous coal when it is 1/6 to 1/7, for the conditions of mediu[illegible]
combustion, such as 10 to 12 lbs. per square foot of grate per h[illegible]
square feet of heating-surface allowed to the horse-power.

The tube area should be made large enough not to choke the d[illegible]
[illegible] lessen the capacity of the boiler; if made too large the gases
[illegible] the p[illegible] least resistance and escape from the[illegible]
[illegible] state.
[illegible] monly found in horizontal tubular [illegible]

go chiefly through the upper rows of tubes; sometimes also in ibular boilers, where the gases are apt to pass most rapidly e tubes nearest to the centre.

ssages through Grate-bars.—The usual practice is, air-30% to 50% of area of the grate; the larger the better, to avoid f the air-supply by clinker; but with coal free from clinker much r-space may be used without detriment. See paper by F. A. Trans. A. S. M. E., vol. xv. p. 503.

PERFORMANCE OF BOILERS.

team-engine, vol. i. p. 327) gives the following formulas for the f coal and water consumed in steam-boilers per square foot of per hour, and the ratio of the heating-surface to the area of the Water taken as evaporated from and at 212° F.

boilers..................................	$w = .0222r^2 + 9.56c$
ilers....................................	$w = .016r^2 + 10.25c$
ngine boilers............................	$w = .008r^2 + 8.6c$
e boilers (coal-burning).................	$w = .009r^2 + 9.7c$
e boilers (coke-burning).................	$w = .0178r^2 + 7.94c$

w = weight of water in pounds per square foot of grate per hour;
c = pounds of fuel per square foot of grate per hour;
r = ratio of heating to grate surface.

re minimum rates of consumption of fuel below which these are not applicable. The limit varies for each kind of boiler, and it h the surface-ratio. It is imposed by the fact that the maximum ve power of fuel is a fixed quantity, and is naturally at that point reduction of the rate of combustion for a given ratio procures the n into the boiler of the whole of the proportion of the heat which le for evaporation. In the combustion of good coal the limit of ve efficiency may be taken as measured by 12½ lbs. of water from 12° F.; and in that of good coke by 12 lbs. of water from and at Based on these formulæ Clark gives the following table:

ative Performance of Steam-boilers for Increasing es of Combustion and different Surface-ratios.

For best coal; surface-ratio 30.

f	Water from and at 212° F. per hour.	Fuel per Square Foot of Grate per hour, in pounds.						
		5	10	15	20	30	40	50
		lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.
y.	Per sq. ft. of grate	62.5*	116	163	211	307	402	498
	Per lb. of coal......	12.5	11.56	10.89	10.56	10.23	10.06	9.96
	Per sq. ft. of grate	62.5*	117	168	219	322	424	527
	Per lb. of coal.....	12.5	11.69	11.25	10.95	10.69	10.61	10.54
	Per sq. ft. of grate	50	93	136	179	265	351	437
	Per lb. of coal......	10	9.3	9.01	8.95	8.83	8.77	8.74
ive.	Per sq. ft. of grate	57	105	154	202	299	396	493
	Per lb. of coal.. ...	11.4	10.5	10.26	10.10	9.97	9.90	9.86

Surface-ratio 50.

		5	10	15	20	30	40	50
		lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.
y.	Per sq. ft. of grate	62.5*	125*	187.5*	247	342	438	534
	Per lb. of coal......	12.5	12.5	12.5	12.33	11.41	10.95	10.67
	Per sq. ft. of grate	62.5*	125*	187.5*	245	348	450	552
	Per lb. of coal.....	12.5	12.5	12.5	12.25	11.58	11.25	11.05
	Per sq. ft. of grate	62.5*	106	149	192	278	364	450
	Per lb. of coal.. ...	12.5	10.6	9.93	9.6	9.27	9.10	9.00
ive.	Per sq. ft. of grate	62.5*	120	168	217	314	411	508
	Per lb. of coal......	12.5	11.95	11.20	10.85	10.4[illegible]	[illegible]	[illegible]

quantities fall below the scope of the formulæ for t[illegible] in the text.

Surface ratio 75.

		30	40	50	60	75
		lbs.	lbs.	lbs.	lbs.	lbs.
Locomotive.	Per sq. ft. of grate.	342	439	536	633	778
"	Per lb. of coal......	11.39	10.97	10.71	10.6[illegible]	10.[illegible]

General Conditions which secure Economy o[f] boilers.—In general, the highest results are produced where t[he temperature] of the escaping gases is the least. An examination of this [is] made by Mr. G. H. Barrus in his book on "Boiler Tests," by se[lecting] tests made by him, six in number, in which the temperature [is above the] average, that is, 375° F., and comparing with five tests in which [the temper]ature is less than 375°. The boilers are all of the common hor[izontal tubular type,] and all use anthracite coal of either egg or broken size. The a[verage] temperatures in the two series was 444° and 343° respectively, a[nd the dif]ference was 101°. The average evaporations are 10.40 lbs. and [...] spectively, and the lowest result corresponds to the case of the [...] temperature. In these tests it appears, therefore, that a redu[ction] in the temperature of the waste gases secured an increase in the [...] of 6%. This result corresponds quite closely to the effect of [...] temperature of the gases by means of a flue-heater where a r[...] 107° was attended by an increase of 7% in the evaporation per p[...]

A similar comparison was made on horizontal tubular boilers [...] berland coal. The average flue temperature in four tests is 4[...] average evaporation is 11.34 lbs. Six boilers have temperatures [...] the average of which is 383°, and these give an average evaporat[ion ...] lbs. With 67° less temperature of the escaping gases the eva[poration is] higher by about 4%.

The wasteful effect of a high flue temperature is exhibited by [...] than those of the horizontal tubular class. This source of waste [...] to be the main cause of the low economy produced in those ver[...] which are deficient in heating-surface.

Relation between the Heating-surface and Grate-surface fo[r] Highest Efficiency.—A comparison of three tests of horizon[tal] boilers with anthracite coal, the ratio of heating-surface to gr[ate] being 36.4 to 1, with three other tests of similar boilers, in whi[ch it] was 48 to 1, showed practically no difference in the results. T[his] shows that a ratio of 36 to 1 provides a sufficient quantity of hea[ting-surface] to secure the full efficiency of anthracite coal where the rate of [...] is not more than 12 lbs. per sq. ft. of grate per hour.

In tests with bituminous coal an increase in the ratio from 36[...] cured a small improvement in the evaporation per pound of coal [...] temperature of the escaping gases indicated that a still furt[her ...] would be beneficial. Among the high results produced on comm[on horizon]tal tubular boilers using bituminous coal, the highest occurs wh[ere the ratio] is 53.1 to 1. This boiler gave an evaporation of 12.47 lbs. A [...] boiler furnishes another example of high performance, an eva[poration of] 12.42 lbs. having been obtained with bituminous coal, and in t[his the] ratio is 65 to 1. These examples indicate that a much larger [...] heating-surface is required for obtaining the full efficiency of [bituminous] coal than for boilers using anthracite coal. The temperature o[f the escap]ing gases in the same boiler is invariably higher when bitumi[nous coal is] used than when anthracite coal is used. The deposit of soot on [...] when bituminous coal is used interferes with the full efficiency [of the sur]face, and an increased area is demanded as an offset to the lo[ss the] deposit occasions. It would seem, then, that if a ratio of 36 to [1 suffices] for anthracite coal, from 45 to 50 should be provided when bitu[minous coal] is burned, especially in cases where the rate of combustion is a[...] lbs. per sq. ft. of grate per hour.

The number of tubes controls the ratio between the area of g[rate] and area of tube-opening. A certain minimum amount of tub[e-opening is] required for efficient work.

The best results obtained with anthracite coal in the commo[n ...] boiler are in cases where the ratio of area of grate-surface to [...] opening is lar[...] to 1. The conclusion is drawn that the [...] ciency with [...] oal is obtained when the tube-opening i[...] 1/9 of the [...]

inous coal is burned the requirements appear to be different. large tube opening does not seem to make the extra tubes n bituminous coal is used. The highest result on any boiler of tubular class, fired with bituminous coal, was obtained where ng was the largest. This gave an evaporation of 12.47 lbs., the surface to tube-opening being 5.4 to 1. The next highest re- lbs., the ratio being 5.2 to 1. Three high results, averaging obtained when the average ratio was 7.1 to 1. Without going the ratio to be desired when bituminous coal is used is that tube-opening having an area of from 1/6 to 1/7 of the grate- applies to medium rates of combustion of, say, 10 to 12 lbs. per per hour, 12 sq. ft. of water-heating surface being allowed per

n of results obtained from different types of boilers leads to nclusion that the economy with which different types of e depends much more upon their proportions and the condi- hich they work, than upon their type; and, moreover, that oportions are suitably carried out, and when the conditions the various types of boilers give substantially the same eco-

of a Steam-boiler.—The efficiency of a boiler is the the total heat generated by the combustion of the fuel ed in heating the water and in raising steam. With anthracite ing-value of the combustible portion is very nearly 14,500 ., equal to an evaporation from and at 212° of 14,500 ÷ 966 ter. A boiler which when tested with anthracite coal shows of 12 lbs. of water per lb. of combustible, has an efficiency of a figure which is approximated, but scarcely ever quite e best practice. With bituminous coal it is necessary to have n of its heating-power made by a coal calorimeter before the e boiler using it can be determined, but a close estimate may the chemical analysis of the coal. (See Coal.)

ce between the efficiency obtained by test and 100% is the sum us wastes of heat, the chief of which is the necessary loss due ature of the chimney-gases. If we have an analysis and a etermination of the heating-power of the coal (properly sam- average analysis of the chimney-gases, the amounts of the ay be determined with approximate accuracy by the method w.

IS OF THE COAL. Semi-bituminous.	
. ...	80.55
. ...	4.50
.. ..	2.70
....	1.08
....	2.92
....	8.25
	100.00

2. ANALYSIS OF THE DRY CHIMNEY-GASES, BY WEIGHT.

				C.	O.	N.
CO_2	= 13.6	=		3.71	9.89	.....
CO	= .2	=		.09	.11	.....
O	= 11.2	=		....	11.20	.....
N	= 75.0	=		...	.....	75.00
	100.0			3.80	21.20	75.00

e of the coal by Dulong's formula, 14,243 heat-units.
eing collected over water, the moisture in them is not deter-

efuse as determined by boiler-test, 10.25, or 2% more than that ysis, the difference representing carbon in the ashes obtained est.
ure of external atmosphere, 60° F.
umidity of air, 60%, corresponding (see air tables) to .007 lb. of lb. of air.
ure of chimney-gases, 560° F.
esults:
in the chimney-gases being 3.8% of their weight, the total gases per lb. of carbon burned is 100 ÷ 3.8 = 26.32 lbs. Since rned is 80.55 − 2 = 78.55% of the weight of the coal ... ...ght es per lb. of coal is 26.32 × 78.55 ÷ 100 = 20.67 l... of coal furnishes to the dry chimney-gases .78
$\frac{.50}{8}$) ÷ *100 = .0214* lb. *O*; a total of .8177, say .8

tracted from 20.67 lbs. leaves 19.85 lbs. as the quantity of dry air (not in- ing moisture) which enters the furnace per pound of coal, not counting air required to burn the available hydrogen, that is, the hydrogen minus eighth of the oxygen chemically combined in the coal. Each lb. of burned contained .045 lb. H, which requires .045 × 8 = .36 lb. O for its combustion. Of this, .027 lb. is furnished by the coal itself, leaving .333 to come from the air. The quantity of air needed to supply this oxygen containing 23% by weight of oxygen) is .333 ÷ .23 = 1.45 lb., which added to the 19.85 lbs. already found gives 21.30 lbs. as the quantity of dry air supplied to the furnace per lb. of coal burned.

The air carried in as vapor is .0071 lb. for each lb. of dry air, or 21.3 × = 0.15 lb. for each lb. of coal. Each lb. of coal contained .029 lb. of moisture, which was evaporated and carried into the chimney-gases. The .045 of H per lb. of coal when burned formed .045 × 9 = .405 lb. of H_2O.

From the analysis of the chimney-gas it appears that .09 ÷ 3.80 = the carbon in the coal was burned to CO instead of to CO_2.

We now have the data for calculating the various loses of heat as fol- for each pound of coal burned:

		Heat-units.	Per cent Heat of the C
20.67 lbs. dry gas × (560° − 60°) × sp. heat 0.24	=	2480.4	[illegible]
.15 lb. vapor in air × (560° − 60°) × sp. heat .48	=	36.0	[illegible]
.029 lb. moisture in coal heated from 60° to 212°	=	4.4	[illegible]
" evaporated from and at 212°; .029 × 966	=	28.0	[illegible]
" steam (heated from 212° to 560°) × 348 × .48	=	4.8	[illegible]
.405 lb. H_2O from H in coal × (152 + 966 + 348 × .48)	=	520.4	[illegible]
.0237 lb. C burned to CO; loss by incomplete combustion, .0237 × (14544 − 4451)	=	239.2	[illegible]
.02 lb coal lost in ashes; .02 × 14544	=	290.9	[illegible]
Radiation and unaccounted for, by difference	=	621.0	[illegible]
		4228.1	[illegible]
Utilized in making steam, equivalent evaporation 10.37 lbs. from and at 212° per lb. of coal	=	10,014.9	[illegible]
		14,243.0	[illegible]

The heat lost by radiation from the boiler and furnace is not easily determined directly, especially if the boiler is enclosed in brickwork, or protected by non-conducting covering. It is customary to estimate the heat lost by radiation by difference, that is, to charge radiation with all the heat lost which is not otherwise accounted for.

One method of determining the loss by radiation is to block off a portion of the grate-surface and build a small fire on the remainder, and drive the fire with just enough draught to keep up the steam-pressure and supply the heat lost by radiation without allowing any steam to be discharged, weighing the coal consumed for this purpose during a test of several hours duration.

Estimates of radiation by difference are apt to be greatly in error, since in this difference are accumulated all the errors of the analyses of the coal and of the gases. An average value of the heat lost by radiation from a boiler set in brickwork is about 4 per cent. When several boilers are in a battery and enclosed in a boiler-house the loss by radiation may be much less, since much of the heat radiated from the boiler is returned in the air supplied to the furnace, which is taken from the boiler-room.

An important source of error in making a "heat balance" such as the one above given, especially when highly bituminous coal is used, is due to the non-combustion of part of the hydrocarbon gases distilled from the coal immediately after firing, when the temperature of the furnace may be reduced below the point of ignition of the gases. Each pound of hydrogen which escapes burning is equivalent to a loss of heat in the furnace of 62,500 heat-units.

In analyzing the chimney gases by the usual method the percentages of the constituent gases are obtained by volume instead of by weight. To reduce percentages by volume to percentages by weight, multiply the percentage by volume of each gas by its specific gravity as compared with air, and divide each product by the sum of the products.

ds of air required to burn a pound of carbon may be obtained
m the analysis by volume by the following formula:

$$\left.\begin{matrix}\text{of air required to burn}\\ \text{pound of carbon}\end{matrix}\right\} = \frac{4}{3}\left\{\frac{2(CO_2 + O) + CO}{CO_2 + CO}\right\} + 0.23;$$

, CO_2, and CO are the per cents, by volume, of the several con- the flue gases.

$$\text{per pound} \Big\} = \left\{\begin{matrix}\text{Lbs. of air per pound}\\ \text{of carbon}\end{matrix}\right\} \times \left\{\begin{matrix}\text{Per cent of carbon}\\ \text{in coal.}\end{matrix}\right.$$

to volume at temperature of 32° F. make use of the formula

$$V_0 = 12.387 \times \text{lbs. of air per pound of coal.}$$

TESTS OF STEAM-BOILERS.

tests at the Centennial Exhibition, Philadel- 76.—(See Reports and Awards Group XX, International Exhibi- , 1876; also, Clark on the Steam-engine, vol. i, page 253.)
ive tests were made of fourteen boilers, using good anthracite oiler, the Galloway, being tested with both anthracite and semi- coal. Two tests were made with each boiler: one called the ial, to determine the economy and capacity at a rapid rate of d the other called the economy trial, to determine the economy n at a rate supposed to be near that of maximum economy and city. The following table gives the principal results obtained in y trial, together with the capacity and economy figures of the ial for comparison.

e r.	Economy Tests.									Capacity Tests.	
	Ratio Water-heating Surface to Grate-surface.	Coal burned per sq. ft. Grate per hour.	Per cent Ash and Refuse.	Water evap. from 100° to 70 lbs. p. s.ft. H.S. per hr.	Water evap. from and at 212° p. lb. comb'ble, cor. for Quality of Steam.	Temperature in Uptake.	Moisture in Steam.	Superheating of Steam.	Horse-power.	Horse-power.	Water evap. from and at 212° per lb. Combustible.
		lbs.	p.ct	lbs.	lbs.	deg	%	deg	H.P.	H.P.	lbs.
....	34.6	9.1	10.4	2.25	12.094	393	...	41.4	119.8	148.6	10.441
....	64.3	12.0	10.4	1.68	11.988	415	...	32.6	57.8	68.4	11.064
....	30.6	6.8	11.3	1.87	11.923	333	...	9.4	47.0	69.3	11.163
....	45.8	12.1	11.1	2.42	11.906	411	1.3	...	99.8	125.0	11.925
Wilcox	37.7	10.0	11.0	2.43	11.822	296	2.7	....	135.6	186.6	10.330
....	23.7	9.6	11.1	3.63	11 583	308	...	1.4	108.3	133.8	11.216
bit. coal	23.7	7.9	8.8	3.20	12.125	325	0.3	....	90.9	125.1	11.609
....	15.6	8.0	10.3	2.32	11.039	420	...	71.7	42.6	58.7	9.745
....	27.3	12.4	8.5	2.75	10.930	517	0.9	....	82.4	108.4	9.889
....	30.7	12.3	9.5	3.30	10.834	524	....	20.5	147.5	162.8	9.145
....	17.5	9.7	9.3	2.64	10.618	417	....	15.7	98.0	132.8	9.568
....	20.9	10.8	9.0	3.82	10.312	....	5.6	...	81.0	99.9	8 397
....	33.5	9.3	11.4	1.38	10.041	430	4.2	....	72.1	108.0	9.974
....	14.0	8.0	11.0	4.44	10.021	374	5.2	....	51.7	67.8	9.865
Black ...	19.0	8.6	9.9	3.43	9.613	572	2.1	....	45.7	67.2	9.429
....	....	...	....	2.77	11.123	...	...	....	85.0	110.8	10.2

parison of the economy and capacity trials shows that an avera capacity of 30 per cent was attended by a decrease in econon nt, but the relation of economy to rate of driving varied great rent boilers. In the Kelly boiler an increase in capacity of [illegible] tended by a decrease in economy of over 18 per [illegible] *er with an increase of* 25 per cent in capacity [illegible] *economy.*

One of the most important lessons gained from the above tests there is no necessary relation between the type of a boiler and econ the five boilers that gave the best results, the total range of varia tween the highest and lowest of the five being only 2.3%, three were tube boilers, one was a horizontal tubular boiler, and the fifth was bination of the two types. The next boiler on the list, the Galloway, internally fired boiler, all of the others being externally fired. The f is a brief description of the principal constructive features of the boilers:

Root..................	4-in. water-tubes, inclined 20° to horizontal; r draught.
Firmenich.............	3-in. water-tubes, nearly vertical; reversed d
Lowe..................	Cylindrical shell, multitubular flue.
Smith.................	Cylindrical shell, multitubular flue—water-side flues.
Babcock & Wilcox....	3½-in. water-tubes, inclined 15° to horizo versed draught.
Galloway......	Cylindrical shell, furnace-tubes and water-t
Andrews....	Square fire-box and double return multitubu
Harrison..............	8 slabs of cast-iron spheres, 8 in. in diam versed draught.
Wiegand...............	4-in. water-tubes, vertical, with internal cir tubes.
Anderson..............	3-in. flue-tubes, nearly horizontal; return cir
Kelly.................	3-in. water-tubes, slightly inclined; each div internal diaphragm to promote circulatio
Exeter................	27 hollow rectangular cast-iron slabs.
Pierce	Rotating horizontal cylinder, with flue-tubes
Rogers & Black.......	Vertical cylindrical boiler, with external wate

Tests of Tubulous Boilers.—The following tables are giv H. Leonard, Asst. Engr. U. S. N., in *Jour. Am. Soc. Naval Engrs.* 18 tests were made at different times by boards of U. S. Naval Engin cept the test of the locomotive-torpedo boiler, which was made in E

No.	Type.	Coal burned per sq. ft. Grate per hour.	Evaporation from and at 212° F. Per lb. Com'ble.	Per sq. ft. H. Surface.	Per cu. ft. Space.	Weights, lbs. E, Empty. S, Steaming Level.	Per I.H.P.	Per sq. ft. H. Surface.	Per lb. Water evaporated.	Air-pressure, in. of Water.
1	Belleville..	12.8	10.42	5.2	6.4	E 40,670 S 42,770	204	53.2	10.1	Nat'
2	Herreshoff	9.3 25.8	10.23 8.68	3.1 8	9.1 23.8	E 2,945 S 3,050	96 35	14.8	4.8 1.8	Jet Jet
3	Towne.....	4.3 24.5	13.4 6 77	2.7 8.2	10 30.4	E 1,380 S 1,640	172 56	21.8	8.1 2.6	Nat' 1.
4	Ward	7.9 15.5 62.5	10.77 10.01 7.01	1.7 3.2 10	5.8 11 34.2	E 1,682 S 1,930	154 82 26	13.2	7.7 4.07 1.3	Nat' Jet Jet
5	Scotch.....	24.8 38	9.93 9.06	8.6 12.8	11 16.3	E 18,900 S 30,000	120 80	41.2	4.7 3.1	2. 4.
6	Locom'tive torpedo,	98.3 120.8	....	17.1 20.05	30.5 36.2	S 34,990	47.7 33.3	31.3	1.8 1.2	3. 4.
7	Ward	55.04	8.44	9.47	32.1	E 26,533 S 30,474	26	12.3	1.3	2
8	Thorny-croft. (U. S.S. Cushing.)	45	.....	.....	.....	E 20,160 S 24,640	*31	10.3	.....	4

* Approximate.

[illegible] cent moisture [illegible] m: Belleville, 6.31; Herreshoff (6 [illegible]
[illegible] ist, 3.44; [illegible] d, 11.6; others not given.

DIMENSIONS OF THE BOILERS.

	1	2	3	4	5	6	7	8
and in..	8′ 6″	4′ 9″	2′ 6″	3′ 2″	9′ 0″	16′ 8	10′ 3″*	10′ 0″‡
" " ..	7 0	3 8	2 6	1 7	9 0	6 4	4 6 †	7 0‡
" " ..	11 0	4 0	3 3	7 2	......	7 6	11 8	8 0‡
t........	645.5	69.6	20 3	42.7	572.5	630.3	729.3	560‡
sq. ft..	34.17	9	4.25	3.68	31.16	28	66.5	38.3
face, ...	804	205	75	146	727	1116	2490	2375
-G ...	23.5	22	17.6	39.5	23.3	39.8	37.4	62

* Diameter. † Diam. of drum. ‡ Approximate.

it per I.H.P. is estimated on a basis of 20 lbs. of water per hour expecting the Scotch boiler, where 25 lbs. have been used, as this imited to 80 lbs. pressure of steam.

ving approximation is made from the large table, on the assump- e evaporation varies directly as the combustion, and 25 lbs. of iare foot of grate per hour used as the unit.

f Boiler.	Com bustion.	Evaporation per cu. ft. of Space.	Weight per I.H.P.	Weight per sq. ft. Heating-surface.	Weight per lb. Water Evaporated.
....	0.50	0.50	2.02	2.10	2.50
....	1.00	0.95	0.72	0.60	0.90
.. ..	1.00	1.20	1.12	0.87	1.30
....	1.00	0.44	2.40	1.64	2.30
....	3.90	0.31	3.70	1.25	3.50
....	2.20	0.58	1.27	0.50	1.53

ville boiler has no practical advantage over the Scotch either in pied or weight. All the other tubulous boilers given greatly Scotch in these advantages of weight and space.

High Rates of Evaporation.—*Eng'g*, May 9, 1884, p. 415.

	Locomotive.		Torpedo-boat.	
. per sq. ft. H.S. per hour.	12.57	13.73	12.54	20.74
" lb. fuel from and at 212°.	8.22	8.94	8.37	7 04
its transf'd per sq. ft. of H.S.	12,142	13,263	12,113	20,034
....	.586	.637	.542	.468

tful if these figures were corrected for priming.

ny Effected by Heating the Air Supplied to irnaces. (Clark, S. E.)—Meunier and Scheurer-Kestner ob- ut 7% greater evaporative efficiency in summer than in winter, me boilers under like conditions,—an excess which had been ex- the difference of loss by radiation and conduction. But Mr. surmising that the gain might be due in some degree also to the perature of the air in summer, made comparative trials with of three boilers, each working one week with the heated air, t week with cold air. The following were the several efficien-

FIRST TRIALS: THREE BOILERS; RONCHAMP COAL.

	Water per lb. of Coal.	Water per lb. of Combustible.
h heated air (128° F.)	7.77 lbs.	8.95 lbs.
h cold air (69°.8).................	7.33 "	8.63 "
erence in favor of heated air	0.44 "	"

SECOND TRIALS: SAME COAL; THREE OTHER

heated air (120°.4 F.)............	8.70 lbs.
cold air (75°.2)	8.09 "
nce in favor of heated air.....	0.61 "

These results show economies in favor of heating the air of 6% and 7[illegible]

Mr. Poupardin believes that the gain in efficiency is due chiefly to [illegible] better combustion of the gases with heated air. It was observed that [illegible] heated air the flames were much shorter and whiter, and that there [illegible] notably less smoke from the chimney.

An extensive series of experiments was made by J. C. Hoadley (Trans. A. S. M. E., vol. vi., 676) on a "Warm-blast Apparatus," for utilizing [illegible] heat of the waste gases in heating the air supplied to the furnace. The [illegible]paratus, as applied to an ordinary horizontal tubular boiler 60 in. diam[illegible] 21 feet long, with 65 3½-inch tubes, consisted of 240 2-inch tubes, 18 feet [illegible] through which the hot gases passed while the air circulated around the[illegible] The net saving of fuel effected by the warm blast was from 10.7% to 1[illegible] the fuel used with cold blast. The comparative temperatures averaged [illegible] follows, in degrees F.:

	Cold-blast Boiler.	Warm-blast Boiler.	Difference.
In heat of fire	2493	2793	300
At bridge wall	1340	1600	260
In smoke box	373	375	2
Air admitted to furnace	32	332	300
Steam and water in boiler	300	300	0
Gases escaping to chimney	373	162	211
External air	32	32	0

With anthracite coal the evaporation from and at 212° per lb. combustible was, for the cold-blast boiler, days 10.85 lbs., days and nights 10.51; and [illegible] the warm-blast boiler, days 11.83, days and nights 11.03.

Results of Tests of Heine Water-tube Boilers with Different Coals.

(Communicated by E. D. Meier, C.E., 1894.)

Number	1	2	3	4	5	6	7	8
Kind of Coal.	Cumberland, Semi-bitum.	2d Pool, Youghiogheny.		Turkey Hill, Ill.	Carbon Hill, Wash.	Hocking Val., Ohio.	[illegible]	[illegible]
Per cent ash	5.1	4.89	......	11.6	16.1	11.5	[illegible]	[illegible]
Heating-surface, sq. ft.	2900	2040	2040	2300	1260	3730	[illegible]	[illegible]
Grate-surface, sq. ft.	54	44.8	44.8	50	21	73.3	[illegible]	[illegible]
Ratio H.S. to G.S.	53.7	45.5	45.5	46	60	50.9	[illegible]	[illegible]
Coal per sq. ft. G. per hr.	24.7	23.5	22.7	35	33.7	26.2	[illegible]	[illegible]
Water per sq. ft. H.S. per hr. from and at 212°	5.03	5.14	5.24	5.56	4.26	4.28	[illegible]	[illegible]
Water evap. from and at 212° per lb. coal	10.91	9.94	10.51	7.31	7.59	8.33	[illegible]	[illegible]
Per lb. combustible	11.50	10.48	...	8.27	9.05	9.41	[illegible]	[illegible]
Temp. of chimney gases	530°	.....	400	567	571	...	[illegible]	[illegible]
Calorific value of fuel	13,800	12,936	12,936	10,487	11,785	11,670	[illegible]	[illegible]
Efficiency of boiler per c.	77.0	74.3	78.5	67.2	62.5	69.3	[illegible]	[illegible]

Tests Nos. 7 and 8 were made with the Hawley Down-draught Furnace, the others with ordinary furnaces.

These tests confirm the statement already made as to the difficulty [illegible] obtaining, with ordinary grate-furnaces, as high a percentage of the ca[illegible]rific value of the fuel with the Western as with the Eastern coals.

Test No 3, 78.5% efficiency, is remarkably good for Pittsburgh (Youghiogheny) coal. If the Washington coal had given equal efficiency, the saving [illegible] fuel would be $\frac{78.5 - 62.5}{78.5} = 20.2\%$. The results of tests Nos. 7 and 8 indicate [illegible] the downward [illegible] furnace is well adapted for burning Illi[illegible]

um Boiler Efficiency with Cumberland Coal.—
bs. of water per lb. combustible from and at 212° is about the
poration that can be obtained from the best steam fuels in the
es, such as Cumberland, Pocahontas, and Clearfield. In excep-
13 lbs. has been reached, and one test is on record (F. W. Dean,
, Feb. 1, 1894) giving 13.23 lbs. The boiler was internally fired,
ire type, 82 inches diameter, 31 feet long, with 160 3-inch tubes
g. Heating surface, 1998 square feet; grate-surface, 45 square feet,
ing the test to 30½ square feet. Double furnace, with fire-brick
a long combustion-chamber. Feed-water heater in smoke-box.
ng are the principal results:

	1st Test.	2d Test.
rned per sq. ft. of grate per hour, lbs.......	8.85	16.06
. per sq. ft. of heating-surface per hour, lbs	1.63	3.00
. from and at 212° per lb. combustible, in- feed-water heater..........................	13.17	13.23
orated, excluding feed-water heater.........	12.88	12.90
re of gases after leaving heater, F...........	360°	463°

BOILERS USING WASTE GASES.

tioning Boilers for Blast-Furnaces.—(F. W. Gordon, Trans. A. I. M. E., vol. xii., 1883.)

on's recommendation for proportioning boilers when properly set
blast-furnace gas is, for coke practice, 30 sq. ft. of heating-sur-
n of iron per 24 hours, which the furnace is expected to make,
the heating-surface thus: For double-flued boilers, all shell-
osed to the gases, and half the flue-surface; for the French type,
osed surface of the upper boiler and half the lower boiler-
r cylindrical boilers, not more than 60 ft. long, all the heating-

ove must be added a battery for relay in case of cleaning, repairs,
ore than one battery extra in large plants, when the water carries

acite practice add 50% to above calculations. For charcoal prac-
20%.

er to the author in May, 1894, Mr. Gordon says that the blast-
ctice at the time when his article (from which the above extract
as written was very different from that existing at the present
es, more economical engines are being introduced, so that less
ft. of boiler-surface per ton of iron made in 24 hours may now be
He says further: Blast-furnace gases are seldom used for other
e requirements, which of course is throwing away good fuel. In
furnace in an ordinary good condition, and a condition where it
maximum of blast, which is in the neighborhood of 200 to 225
mospheric measurement, per sq. ft. of sectional area of hearth,
te the necessary H.P. with very small heating-surface, owing to
at of the escaping gases from the boilers, which frequently is
s.

making 200 tons of iron a day will consume about 900 H.P. in
engine. About a pound of fuel is required in the furnace per
g metal.

e it requires 70 cu. ft. of air-piston displacement per lb. of fuel
or 22,400 cu. ft. per minute for 200 tons of metal in 1400 working
day, at, say, 10 lbs. discharge-pressure. This is equal to 9¼ lbs.
e steam-piston of equal area to the blast-piston, or 900 I.H.P. To
for hoisting, pumping and other purposes for which steam is em-
nd blast-furnaces, and we have 1100 H.P., or say 5½ H.P. per
per day. Dividing this into 30 gives approximately 5½ sq. ft. of
face of boiler per H.P.

tube Boilers using Blast-furnace Gases.—D. S
ans. A. I. M. E., xvii, 50) reports a test of a water-tube boiler usin
e gas as fuel. The heating-surface was 2535 sq. ft. It develope
ntennial standard), or 5.01 lbs. of water f[illegible] pe
ating-surface per hour. Some of the pr[illegible]
ows: Calorific value of 1 lb. of the gas, [illegible]
its initial temperature, which was 650° F
of the gas = 0.9 lb. Chimney draught, 1½
sq. in.; of air inlet, 100 sq. in. Temper

gases, 775° F. Efficiency of the boiler calculated from the tempe and analyses of the gases at exit and entrance, 61%. The average a were as follows, hydrocarbons being included in the nitrogen:

	By Weight.		By Volume.	
	At Entrance.	At Exit.	At Entrance.	At
CO_2	10.69	26.37	7.08	
O	.11	3.05	.10	
CO	26.71	1.78	27.80	
Nitrogen	62.48	68.80	65.02	
C in CO_2	2.92	7.19		
C in CO	11.45	.76		
Total C	14.37	7.95		

Steam-boilers Fired with Waste Gases from Pud and Heating Furnaces.—The *Iron Age*, April 6, 1893, contains a of a number of tests of steam-boilers utilizing the waste heat fro dling and heating furnaces in rolling-mills. The following principal d selected: In Nos. 1, 2, and 4 the boiler is a Babcock & Wilcox wate boiler, and in No. 3 it is a plain cylinder boiler, 42 in. diam. and 2 ft. No. 4 boiler was connected with a heating-furnace, the others with pud furnaces.

	No. 1.	No. 2.	No. 3.	N
Heating-surface, sq. ft	1026	1196	143	
Grate-surface, sq. ft	19.9	13.6	13.6	
Ratio H.S. to G.S.	52	87.2	10.5	
Water evap. per hour, lbs	3358	2159	1812	
" " per sq. ft. H.S. per hr., lbs	3.3	1.8	12.7	
" " per lb. coal from and at 212°	5.9	6.24	3.78	
" " " " comb. " " " "	....	7.20	4.31	

In No. 2, 1.38 lbs. of iron were puddled per lb. of coal.

In No. 3, 1.14 lbs. of iron were puddled per lb. of coal.

No. 3 shows that an insufficient amount of heating-surface was pro for the amount of waste heat available.

RULES FOR CONDUCTING BOILER-TESTS.

The Committee of the A. S. M. E. on Boiler-tests, consisting of Wm. (chairman), J. C. Hoadley, R. H. Thurston, Chas. E. Emery, and Ch Porter, recommended the following code of rules for boiler-tests (Tr vol. vi. p. 256):

Preliminaries to a Test.

I. In preparing for and conducting trials of steam-boilers the spe object of the proposed trial should be clearly defined and steadily kep view.

II. Measure and record the dimensions, position, etc., of grate and h ing surfaces, flues and chimneys, proportion of air-space in the grate face, kind of draught, natural or forced.

III. Put the boiler in good condition. Have heating-surface clean in and out, grate-bars and sides of furnace free from clinkers, dust and a removed from back connections, leaks in masonry stopped, and all obst tions to draught removed. See that the damper will open to full extent, that it may be closed when desired. Test for leaks in masonry by fi little smoky fuel and immediately closing damper. The smoke will escape through the leaks.

IV. Have an understanding with the parties in whose interest the te to be made as to the character of the coal to be used. The coal must be or, if wet, a sample must be dried carefully and a determination of amount of moisture in the coal made, and the calculation of the result the test corrected accordingly. Wherever possible, the test should be with standard coal of a known quality. For that portion of the co of the Alle ntains good anthracite egg coal or Cumberl -bitumin be taken as the standard for making tests.

lleghany Mountains and east of the Missouri River, Pittsburgh lump y be used.*

all important tests a sample of coal should be selected for chemical s.

stablish the correctness of all apparatus used in the test for weighing asuring. These are: 1. Scales for weighing coal, ashes, and water. s, or water-meters for measuring water. Water-meters, as a rule, only be used as a check on other measurements. For accurate work er should be weighed or measured in a tank. 3. Thermometers and ters for taking temperatures of air, steam, feed-water, waste gases, Pressure-gauges, draught-gauges, etc.

Before beginning a test, the boiler and chimney should be thoroughly to their usual working temperature. If the boiler is new, it should ntinuous use at least a week before testing, so as to dry the mortar hly and heat the walls.

Before beginning a test, the boiler and connections should be free aks, and all water connections, including blow and extra feed pipes, be disconnected or stopped with blank flanges, except the particular rough which water is to be fed to the boiler during the trial. In lo- where the reliability of the power is so important that an extra feed- ust be kept in position, and in general when for any other reason pipes other than the feed-pipes cannot be disconnected, such pipes drilled so as to leave openings in their lower sides, which should be pen throughout the test as a means of detecting leaks, or accidental uthorized opening of valves. During the test the blow-off pipe should exposed.

injector is used it must receive steam directly from the boiler being and not from a steam-pipe or from any other boiler.

hat the steam-pipe is so arranged that water of condensation cannot ck into the boiler. If the steam-pipe has such an inclination that the of condensation from any portion of the steam-pipe system may run nto the boiler, it must be trapped so as to prevent this water getting e boiler without being measured.

STARTING AND STOPPING A TEST.

t should last at least ten hours of continuous running, and twenty- ours whenever practicable. The conditions of the boiler and furnace espects should be, as nearly as possible, the same at the end as at ginning of the test. The steam-pressure should be the same, the level the same, the fire upon the grates should be the same in quan- d condition, and the walls, flues, etc., should be of the same tempera- To secure as near an approximation to exact uniformity as possible ditions of the fire and in temperatures of the walls and flues, the ing method of starting and stopping a test should be adopted :

Standard Method.—Steam being raised to the working pressure, re- rapidly all the fire from the grate, close the damper, clean the ash-pit, quickly as possible start a new fire with weighed wood and coal, the time of starting the test and the height of the water-level while ter is in a quiescent state, just before lighting the fire.

he end of the test remove the whole fire, clean the grates and ash-pit, te the water-level when the water is in a quiescent state ; record the f hauling the fire as the end of the test. The water-level should be rly as possible the same as at the beginning of the test. If it is not me, a correction should be made by computation, and not by operat- mp after test is completed. It will generally be necessary to regulate scharge of steam from the boiler tested by means of the stop-valve ime while fires are being hauled at the beginning and at the end of t. in order to keep the steam-pressure in the boiler at those times up average during the test.

Alternate Method.—Instead of the Standard Method above described, lowing may be employed where local conditions render it necessary : e regular time for slicing and cleaning fires have them burned rather s is usual before cleaning, and then thoroughly cleaned ; note the t of coal left on the grate as nearly as it can be estimated ; note the

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ese coals are selected because they are about the only c the essentials *of excellence of quality*, adaptability to v *aces, grates, boilers, and methods* of firing, and wide di *accessibility in the markets.*

pressure of steam and the height of the water-level—which should medium height to be carried throughout the test—at the same ti note this time as the time of starting the test. Fresh coal, which weighed, should now be fired. The ash-pits should be thoroughly at once after starting. Before the end of the test the fires should low, just as before the start, and the fires cleaned in such a man leave the same amount of fire, and in the same condition, on the gr the start. The water-level and steam-pressure should be brough same point as at the start, and the time of the ending of the test noted just before fresh coal is fired.

During the Test.

XII. *Keep the Conditions Uniform.*—The boiler should be run ously, without stopping for meal-times or for rise or fall of pr steam due to change of demand for steam. The draught being ad the rate of evaporation or combustion desired before the test is should be retained constant during the test by means of the damp

If the boiler is not connected to the same steam-pipe with other an extra outlet for steam with valve in same should be provided, case the pressure should rise to that at which the safety-valve is be reduced to the desired point by opening the extra outlet, withou ing the fires.

If the boiler is connected to a main steam-pipe with other bo safety-valve on the boiler being tested should be set a few pound than those of the other boilers, so that in case of a rise in pres other boilers may blow off, and the pressure be reduced by clos dampers, allowing the damper of the boiler being tested to rema and firing as usual.

All the conditions should be kept as nearly uniform as possible, force of draught, pressure of steam, and height of water. The cleaning the fires will depend upon the character of the fuel, the ra combustion, and the kind of grates. When very good coal is used, combustion not too rapid, a ten-hour test may be run without any of the grates, other than just before the beginning and just before of the test. But in case the grates have to be cleaned during the intervals between one cleaning and another should be uniform.

XIII. *Keeping the Records.*—The coal should be weighed and deli the firemen in equal portions, each sufficient for about one hour's a fresh portion should not be delivered until the previous one has fired. The time required to consume each portion should be no time being recorded at the instant of firing the first of each new po is desirable that at the same time the amount of water fed into th should be accurately noted and recorded, including the height of th in the boiler and the average pressure of steam and temperature during the time. By thus recording the amount of water evapor successive portions of coal, the record of the test may be divi several divisions, if desired, at the end of the test, to discover the of uniformity of combustion, evaporation, and economy at differen of the test.

XIV. *Priming Tests.*—In all tests in which accuracy of results i tant, calorimeter tests should be made of the percentage of moistur steam, or of the degree of superheating. At least ten such tests sh made during the trial of the boiler, or so many as to reduce the average error to less than one per cent, and the final records of the test corrected according to the average results of the calorimeter te

On account of the difficulty of securing accuracy in these tests, th est care should be taken in the measurements of weights and tempe The thermometers should be accurate within a tenth of a degree, scales on which the water is weighed to within one hundredth of a

Analyses of Gases.—Measurement of Air-supply, etc.

XV. In tests for purposes of scientific research, in which the det tion of all the variables entering into the test is desired, certain obse should be made which are in general not necessary in tests for com purposes. These are the measurement of the air-supply, the determ of its contained moisture, the measurement and analysis of the flu determination of the amount of heat lost by radiation, of the am ration of air [illegible] the setting, the direct determination by [illegible] experiments [illegible] ute heating value of the fuel, and [illegible] team made by the boiler) of the total heat imparted to the

the flue-gases is an especially valuable method of deter-
e value of different methods of firing, or of different kinds
naking these analyses great care should be taken to pro-
nples—since the composition is apt to vary at different
and the analyses should be intrusted only to a thoroughly
st, who is provided with complete and accurate apparatus.
inations of the other variables mentioned above are not
taken except by engineers of high scientific attainments,
for making them is likely to be improved in the course of
h, it is not deemed advisable to include in this code any
s for making them.

RECORD OF THE TEST.

of the test should be kept on properly prepared blanks,
ngs as follows:

res.		Temperatures.					Fuel.		Feed-water.	
	Draught-gauge.	External Air.	Boiler-room.	Flue.	Feed-water.	Steam.	Time.	Lbs.	Time.	Lbs. or cu. ft.

REPORTING THE TRIAL.

al results should be recorded upon a properly prepared
d include as many of the following items as are adapted for
t for which the trial is made. The items marked with a *
for ordinary trials, but are desirable for comparison with
n other sources.

ts of the trials of a..............................
r at...
termine..

l..................................			
trial	hours.		
IONS AND PROPORTIONS.			
complete description.			
e....wide....long....area........	sq. ft.		
ng surface..............................	sq. ft.		
g surface..............................	sq. ft.		
ter-heating surface to grate-sur-			
ERAGE PRESSURES.			
ure in boiler, by gauge....	lbs.		
am-pressure..........................	lbs.		
e pressure, per barometer.........	in.		
ught in inches of water..	in.		
AGE TEMPERATURES.			
air......................................	deg.		
l..	deg.		
....	deg.		
gases....................................	deg.		
er..	deg.		

See reference in paragraph preceding tal

[illegible]

[illegible] Item [illegible] factor [illegible]
[illegible] Item [illegible] + [illegible] per cent [illegible]
[illegible] Item [illegible] 6.5.
[illegible] Item 20 / 0.9028.

Item [illegible] Item [illegible] 0.00003, or $\frac{\text{Item 20}}{[illegible]}$, or $\frac{\text{Item 29}}{33,305}$.

[illegible] 43 and 44 [illegible]

...t water evaporated per pound of ...l from and at 212° F. §	lbs.		
...t water evaporated per pound of ...tible from and at 212° F. §	lbs.		
...MMERCIAL EVAPORATION.			
...t water evaporated per pound of ...l with one sixth refuse, at 70 pounds ...ressure, from temperature of 100° ...em 33 × 0.7249	lbs.		
RATE OF COMBUSTION.			
...actually burned per square foot of ...rface per hour..	lbs.		
...umption of dry ...er hour. Coal ...d with one ...efuse.§ — Per sq. ft. of grate-surface......	...ts.		
Per sq. ft. of water-heating surface..	lbs.		
Per sq. ft. of least area for draught.	lbs.		
RATE OF EVAPORATION.			
...aporated from and at 212° F. per ...f heating-surface per hour.........	lbs.		
...r evaporated ...ur from tem-...re of 100° F. ...eam of 70 lbs. pressure. § — Per sq. ft. of grate-surface..........	lbs.		
Per sq. ft. of water-heating surface..	lbs.		
Per sq. ft. of least area for draught.	lbs.		
...MMERCIAL HORSE-POWER.			
...of thirty pounds of water per hour ...ated from temperature of 100° F. ...eam of 70 pounds gauge-pressure ...lbs. from and at 212°) §	H.P.		
...ower, builders' rating, at.. ...square ...r horse-power.....	H.P.		
...developed above, or below, rating§.	per cent		

...of Evaporation.—The table on the following pages was ...blished by the author in Trans. A. S. M. E., vol. vi., 1884, under ...bles for Facilitating Calculations of Boiler-tests. The table ...tors for every 3° of temperature of feed-water from 32° to 212° ...ery two pounds pressure of steam within the limits of ordinary ...m-pressures.

...nce in the factor corresponding to a difference of 3° tempera-...s always either .0031 or .0032. For interpolation to find a factor ...ter temperature between 32° and 212°, not given in the table, ...or for the nearest temperature and add or subtract, as the case ...if the difference is .0031, and .0011 if the difference is .0032. As ...cases a factor of evaporation to three decimal places is accu-...any error which may be made in the fourth decimal place by ...is of no practical importance.

...used in calculating these factors of evaporation are those given ... Porter's Treatise on the Richards' Steam-engine Indicator.

...*is Factor* $= \frac{H - h}{965.7}$, *in which H is the total heat of steam at the* ...*sure, and h the total heat of feed-water of the observed*

Gauge-pressures … Lbs. 0 + / Absolute pressures 15	10 + / 25	20 + / 35	30 + / 45	40 + / 55	45 + / 60	50 + / 65	52 + / 67
Feed-water Temperature.	FACTORS OF EVAPORATION.						
212° F.	1.0003 1.0088	1.0149	1.0197	1.0237	1.0254	1.0271	1.0277

Feed-water Temperature	0 + 15	10 + 25	20 + 35	30 + 45	40 + 55	45 + 60	50 + 65	52 + 67
212° F.	1.0003	1.0088	1.0149	1.0197	1.0237	1.0254	1.0271	1.0277
209	35	1.0120	80	1.0228	68	86	1.0302	1.0309
206	66	51	1.0212	60	99	1.0317	34	40
203	98	83	43	91	1.0331	49	65	72
200	1.0129	1.0214	75	1.0323	62	80	97	1.0403
197	60	46	1.0306	54	94	1.0412	1.0428	34
194	92	77	38	85	1.0425	43	60	66
191	1.0223	1.0308	69	1.0417	57	74	91	97
188	55	40	1.0400	48	88	1.0506	1.0522	1.0528
185	86	71	32	80	1.0519	37	54	60
182	1.0317	1.0403	63	1.0511	51	68	85	91
179	49	34	95	42	82	1.0600	1.0616	1.0623
176	80	65	1.0526	74	1.0613	31	48	54
173	1.0411	97	57	1.0605	45	63	79	85
170	43	1.0528	89	36	76	94	1.0710	1.0717
167	74	59	1.0620	68	1.0707	1.0725	42	48
164	1.0505	91	51	99	39	56	73	80
161	37	1.0622	82	1.0730	70	88	1.0804	1.0811
158	68	53	1.0714	62	1.0801	1.0819	36	42
155	99	84	45	93	33	50	67	73
152	1.0631	1.0716	76	1.0824	64	82	98	1.0905
149	62	47	1.0808	55	95	1.0913	1.0930	36
146	93	78	39	87	1.0926	44	61	67
143	1.0724	1.0810	70	1.0918	58	75	92	98
140	56	41	1.0901	49	89	1.1007	1.1023	1.1030
137	87	72	33	80	1.1020	38	55	61
134	1.0818	1.0903	64	1.1012	51	69	86	92
131	49	34	95	43	83	1.1100	1.1117	1.1123
128	81	66	1.1026	74	1.1114	32	48	55
125	1.0912	97	57	1.1105	45	63	79	86
122	43	1.1028	89	36	76	94	1.1211	1.1217
119	74	59	1.1120	68	1.1207	1.1225	42	48
116	1.1005	90	51	99	39	56	73	79
113	36	1.1122	82	1.1230	70	88	1.1304	1.1310
110	68	53	1.1213	61	1.1301	1.1319	35	42
107	99	84	45	92	32	50	66	73
104	1.1130	1.1215	76	1.1323	63	81	98	1.1404
101	61	46	1.1307	55	94	1.1412	1.1429	35
98	92	77	38	86	1.1426	43	60	66
95	1.1223	1.1309	69	1.1417	57	75	91	97
92	55	40	1.1400	48	88	1.1506	1.1522	1.1529
89	86	71	31	79	1.1519	37	53	60
86	1.1317	1.1402	63	1.1510	50	68	84	91
83	48	33	94	41	81	99	1.1616	1.1622
80	79	64	1.1525	72	1.1612	1.1630	47	53
77	1.1410	95	56	1.1604	44	61	78	84
74	41	1.1526	87	35	75	92	1.1709	1.1715
71	72	58	1.1618	66	1.1706	1.1723	40	46
68	1.1504	89	49	97	37	55	71	78
65	35	1.1620	80	1.1728	68	86	1.1802	1.1809
62	66	51	1.1711	59	99	1.1817	33	40
59	97	82	43	90	1.1830	48	64	71
56	1.1628	1.1713	74	1.1821	61	79	96	1.1902
53	59	44	1.1805	52	92	1.1910	1.1927	33
50	90	75	36	84	1.1923	41	58	64
47	1.1721	1.1806	67	1.1915	54	72	89	95
44	52	37	98	46	86	1.2003	1.2020	1.2026
41	83	68	1.1929	77	1.2017	34	51	57
38	1.1814	[illegible]	60	1.2008	48	65	82	88
35	[illegible]	[illegible]	91	39	79	96	1.2113	1.2119
32	[illegible]	[illegible]	[illegible]	70	1.2110	1.2128	44	50

...ress., lbs. / ...e Pressures..	58 + 73	60 + 75	62 + 77	64 + 79	66 + 81	68 + 83	70 + 85	72 + 87	74 + 89	76 + 91
...ater ...p.	Factors of Evaporation.									
F.	1.0295	1.0301	1.0307	1.0312	1.0318	1.0323	1.0329	1.0334	1.0339	1.0344
	1.0327	33	38	44	49	55	60	65	70	75
	58	64	70	75	81	86	91	97	1.0402	1.0407
	90	96	1.0401	1.0407	1.0412	1.0418	1.0423	1.0428	33	38
	1.0421	1.0427	33	38	44	49	54	59	65	69
	53	58	64	70	75	80	86	91	96	1.0501
	84	90	96	1.0501	1.0507	1.0512	1.0517	1.0522	1.0527	32
	1.0515	1.0521	1.0527	33	38	43	49	54	59	64
	47	53	58	64	69	75	80	85	90	95
	78	84	90	95	1.0601	1.0606	1.0611	1.0616	1.0622	1.0626
	1.0610	1.0615	1.0621	1.0627	32	37	43	48	53	58
	41	47	52	58	63	69	74	79	84	89
	72	78	84	89	95	1.0700	1.0705	1.0711	1.0716	1.0721
	1.0704	1.0709	1.0715	1.0721	1.0726	32	37	42	47	52
	35	41	46	52	57	63	68	73	78	83
	[illegible]	72	78	83	89	94	99	1.0805	1.0810	1.0815
	[illegible]	1.0803	1.0809	1.0815	1.0820	1.0825	1.0831	36	41	46
	1.0829	35	40	46	51	57	62	67	72	77
	60	66	72	77	83	88	93	98	1.0904	1.0908
	92	97	1.0903	1.0909	1.0914	1.0919	1.0925	1.0930	35	40
	1.0923	1.0929	34	40	45	51	56	61	66	71
	54	60	66	71	77	82	87	92	97	1.1002
	85	91	97	1.1002	1.1008	1.1013	1.1018	1.1024	1.1029	34
	1.1017	1.1022	1.1028	34	39	44	50	55	60	65
	48	54	59	65	70	76	81	86	91	96
	79	85	91	96	1.1102	1.1107	1.1112	1.1117	1.1122	1.1127
	1.1110	1.1116	1.1122	1.1127	33	38	43	49	54	59
	42	47	53	59	64	69	75	80	85	90
	73	79	84	90	95	1.1201	1.1206	1.1211	1.1216	1.1221
	1.1204	1.1210	1.1215	1.1221	1.1226	32	37	42	47	52
	35	41	47	52	58	63	68	73	78	83
	66	72	78	83	89	94	99	1.1305	1.1310	1.1315
	98	1.1303	1.1309	1.1315	1.1320	1.1325	1.1331	36	41	46
	1.1329	34	40	46	51	57	62	67	72	77
	60	66	71	77	82	88	93	98	1.1403	1.1408
	91	97	1.1403	1.1408	1.1414	1.1419	1.1424	1.1429	34	39
	1.1422	1.1428	34	39	45	50	55	60	65	70
	53	59	65	70	76	81	86	92	97	1.1502
	85	90	96	1.1502	1.1507	1.1512	1.1518	1.1523	1.1528	33
	1.1516	1.1521	1.1527	33	38	43	49	54	59	64
	47	53	58	64	69	75	80	85	90	95
	78	84	89	95	1.1600	1.1606	1.1611	1.1616	1.1621	1.1626
	1.1609	1.1615	1.1621	1.1626	32	37	42	47	52	57
	40	46	52	57	63	68	73	78	83	88
	71	77	83	88	94	99	1.1704	1.1710	1.1715	1.1720
	1.1702	1.1708	1.1714	1.1719	1.1725	1.1730	35	41	46	51
	34	39	45	51	56	61	67	72	77	82
	65	70	76	82	87	92	98	1.1803	1.1808	1.1813
	96	1.1802	1.1807	1.1813	1.1818	1.1824	1.1829	34	39	44
	1.1827	33	38	44	49	55	60	65	70	75
	58	64	69	75	80	86	91	96	1.1901	1.1906
	89	95	1.1901	1.1906	1.1912	1.1917	1.1922	1.1927	32	37
	1.1920	1.1926	32	37	43	48	53	58	63	[illegible]
	51	57	63	68	74	79	84	89	94	[illegible]
	82	88	94	99	1.2005	1.2010	1.2015	1.2021	1.2026	1.203[illegible]
	1.2013	1.2019	1.2025	1.2030	36	41	[illegible]	[illegible]	[illegible]	[illegible]
	44	50	56	61	67	72	[illegible]	[illegible]	[illegible]	[illegible]
	76	81	87	93	98	1.2103	[illegible]	[illegible]	[illegible]	[illegible]
	1.2107	1.2112	1.2118	1.2124	1.2129	34	[illegible]	[illegible]	[illegible]	[illegible]
	38	43	49	55	60	65	[illegible]	[illegible]	[illegible]	[illegible]
	69	75	80	86	91	97	[illegible]	[illegible]	[illegible]	[illegible]

Gauge-pressures lbs.,	78 +	80 +	82 +	84 +	86 +	88 +	90 +	92 +	94 +	96 +
Absolute Pressures,	93	95	97	99	101	103	105	107	109	111
Feed-water Temp.	FACTORS OF EVAPORATION.									
212	1.0349	1.0353	1.0358	1.0363	1.0367	1.0372	1.0376	1.0381	1.0385	1.0389
209	80	85	90	94	99	1.0403	1.0408	1.0412	1.0416	1.0421
206	1.0411	1.0416	1.0421	1.0426	1.0430	35	39	43	48	52
203	43	48	52	57	62	66	71	75	79	83
200	74	79	84	89	93	[illegible]	1.0502	1.0506	1.0511	1.0515
197	1.0506	1.0511	1.0515	1.0520	1.0525	1.05[illegible]	33	38	42	[illegible]
194	37	42	47	51	56	[illegible]	65	69	73	[illegible]
191	69	73	78	83	87	92	96	1.0601	1.0605	[illegible]
188	1.0600	1.0605	1.0610	1.0614	1.0619	1.0623	1.0628	32	36	40
185	31	36	41	46	50	55	59	63	68	[illegible]
182	63	68	72	77	81	86	90	95	99	[illegible]
179	94	99	1.0704	1.0708	1.0713	1.0717	1.0722	1.0726	1.0730	35
176	1.0725	1.0730	35	40	44	49	53	57	62	66
173	57	62	66	71	75	80	84	89	93	97
170	88	93	98	1.0802	1.0807	1.0811	1.0816	1.0820	1.0824	[illegible]
167	1.0819	1.0824	1.0829	34	38	43	47	51	56	60
164	51	56	60	65	69	74	78	83	87	91
161	82	87	92	96	1.0901	1.0905	1.0910	1.0914	1.0918	1.0923
158	1.0913	1.0918	1.0923	1.0927	32	37	41	45	50	[illegible]
155	45	49	54	59	63	68	72	77	81	[illegible]
152	76	81	85	90	95	99	1.1004	1.1008	1.1012	[illegible]
149	1.1007	1.1012	1.1017	1.1021	1.1026	1.1030	35	39	43	[illegible]
146	38	43	48	53	57	62	66	70	75	[illegible]
143	70	74	79	84	88	93	97	1.1102	1.1106	1.1110
140	1.1101	1.1106	1.1110	1.1115	1.1120	1.1124	1.1129	33	37	41
137	32	37	42	46	51	55	60	64	68	[illegible]
134	63	68	73	78	82	87	91	95	1.1200	[illegible]
131	95	99	1.1204	1.1209	1.1213	1.1218	1.1222	1.1227	31	[illegible]
128	1.1226	1.1231	35	40	45	49	53	58	62	[illegible]
125	57	62	67	71	76	80	85	89	93	[illegible]
122	88	93	98	1.1302	1.1307	1.1311	1.1316	1.1320	1.1325	[illegible]
119	1.1320	1.1324	1.1329	34	38	43	47	51	56	[illegible]
116	51	55	60	65	69	74	78	83	87	91
113	82	87	91	96	1.1401	1.1405	1.1409	1.1414	1.1418	[illegible]
110	1.1413	1.1418	1.1422	1.1427	32	36	41	45	49	[illegible]
107	44	49	54	58	63	67	72	76	80	[illegible]
104	75	80	85	89	94	99	1.1503	1.1507	1.1512	1.1516
101	1.1506	1.1511	1.1516	1.1521	1.1525	1.1530	34	38	43	[illegible]
98	38	42	47	52	56	61	65	70	71	[illegible]
95	69	74	78	83	87	92	96	1.1601	1.1605	[illegible]
92	1.1600	1.1605	1.1609	1.1614	1.1619	1.1623	1.1628	32	36	[illegible]
89	31	36	41	45	50	51	59	63	67	72
86	62	67	72	76	81	85	90	94	98	1.1703
83	93	98	1.1703	1.1707	1.1712	1.1717	1.1721	1.1725	1.1730	34
80	1.1724	1.1729	34	39	43	48	52	56	61	[illegible]
77	56	60	65	70	74	79	83	88	92	[illegible]
74	87	91	96	1.1801	1.1805	1.1810	1.1814	1.1819	1.1823	[illegible]
71	1.1818	1.1823	1.1827	32	36	41	45	50	54	[illegible]
68	49	54	58	63	68	72	77	81	85	[illegible]
65	80	85	89	94	99	1.1903	1.1908	1.1912	1.1916	[illegible]
62	1.1911	1.1916	1.1921	1.1925	1.1930	34	39	43	47	52
59	42	47	52	56	61	65	70	74	78	83
56	73	78	83	87	92	96	1.2001	1.2005	1.2009	1.2014
53	1.2004	1.2009	1.2014	1.2018	1.2023	1.2028	32	36	41	[illegible]
50	35	40	45	50	54	59	63	67	72	[illegible]
47	66	71	76	81	85	90	94	98	[illegible]	[illegible]
44	98	1.2102	1.2107	1.2112	1.2116	1.2121	1.2125	1.2130	[illegible]	[illegible]
41	1.2129	33	[illegible]	43	47	52	56	61	65	[illegible]
[illegible]	60	[illegible]	[illegible]	74	78	83	87	92	96	[illegible]
[illegible]	91	[illegible]	[illegible]	[illegible]	1.2209	1.2214	1.2218	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	1.[illegible]	[illegible]	[illegible]	41	45	49	54	58	[illegible]

essures 100 + Press. s. 115.	105 + 120	110 + 125	115 + 130	120 + 135	125 + 140	130 + 145	135 + 150	140 + 155	145 + 160	150 + 165
er	FACTORS OF EVAPORATION.									
1.0397	1.0407	1.0417	1.0427	1.0436	1.0445	1.0453	1.0462	1.0470	1.0478	1.0486
1.0429	39	49	58	67	76	85	93	1.0501	1.0509	1.0517
60	70	80	89	99	1.0508	1.0516	1.0525	33	41	48
92	1.0502	1.0511	1.0521	1.0530	39	48	56	64	72	80
1.0523	33	43	52	62	70	79	87	96	1.0604	1.0611
55	65	74	84	93	1.0602	1.0610	1.0619	1.0627	35	43
86	96	1.0606	1.0615	1.0624	33	42	50	58	66	74
1.0617	1.0627	37	47	56	65	73	82	90	98	1.0706
49	59	69	78	87	96	1.0705	1.0713	1.0721	1.0729	37
80	90	1.0700	1.0709	1.0719	1.0727	36	44	53	61	68
1.0712	1.0722	31	41	50	59	67	76	84	92	1.0800
43	53	63	72	81	90	99	1.0807	1.0815	1.0823	31
74	84	94	1.0803	1.0813	1.0821	1.0830	39	47	55	62
1.0806	1.0816	1.0825	35	44	53	61	70	78	86	94
37	47	57	66	75	84	93	1.0901	1.0909	1.0917	1.0925
68	78	88	97	1.0907	1.0915	1.0924	32	41	49	56
1.0900	1.0910	1.0919	1.0929	38	47	55	64	72	80	88
31	41	51	60	69	78	87	95	1.1003	1.1011	1.1019
62	72	82	91	1.1000	1.1009	1.1018	1.1026	35	43	50
93	1.1003	1.1013	1.1023	32	41	49	58	66	74	82
1.1025	35	44	54	63	72	81	89	97	1.1105	1.1113
56	66	76	85	94	1.1103	1.1112	1.1120	1.1128	36	44
87	97	1.1107	1.1116	1.1126	34	43	51	60	68	75
1.1118	1.1129	38	48	57	66	74	83	91	99	1.1207
50	60	70	79	88	97	1.1206	1.1214	1.1222	1.1230	38
81	91	1.1201	1.1210	1.1219	1.1228	37	45	53	61	69
1.1212	1.1222	32	41	51	59	68	76	85	93	1.1300
43	53	63	73	82	91	99	1.1308	1.1316	1.1324	32
75	85	94	1.1304	1.1313	1.1322	1.1331	39	47	55	63
1.1306	1.1316	1.1326	35	44	53	62	70	78	86	94
37	47	57	66	75	84	93	1.1401	1.1409	1.1417	1.1425
68	78	88	97	1.1407	1.1415	1.1424	32	41	49	56
99	1.1409	1.1419	1.1429	38	47	55	64	72	80	88
1.1431	41	50	60	69	78	86	95	1.1503	1.1511	1.1519
62	72	82	91	1.1500	1.1509	1.1518	1.1516	34	42	50
93	1.1503	1.1513	1.1522	31	40	49	57	65	73	81
1.1524	34	44	53	62	71	80	88	97	1.1605	1.1612
55	65	75	84	94	1.1602	1.1611	1.1620	1.1628	36	43
86	96	1.1606	1.1616	1.1625	34	42	51	59	67	75
1.1618	1.1628	37	47	56	65	73	82	90	98	1.1706
49	50	68	78	87	96	1.1705	1.1713	1.1721	1.1729	37
80	90	1.1700	1.1709	1.1718	1.1727	36	44	52	60	68
1.1711	1.1721	31	40	49	58	67	75	83	91	99
42	52	62	71	80	89	98	1.1806	1.1815	1.1823	1.1830
73	83	93	1.1802	1.1812	1.1820	1.1829	37	46	54	61
1.1804	1.1814	1.1824	34	43	52	60	69	77	85	93
35	45	55	65	74	83	91	1.1900	1.1908	1.1916	1.1924
67	77	86	96	1.1905	1.1914	1.1922	31	39	47	55
98	1.1908	1.1917	1.1927	36	45	54	62	70	78	86
1.1929	39	49	58	67	76	85	93	1.2001	1.2009	1.2017
60	70	80	89	98	1.2007	1.2016	1.2024	32	40	48
91	1.2001	1.2011	1.2020	1.2029	38	47	55	63	71	79
1.2022	32	42	51	60	69	78	86	94	1.2102	1.2110
53	63	73	82	91	1.2100	1.1209	1.2117	1.2126	34	41
84	94	1.2104	1.2113	1 2123	31	40	48	57	65	72
1.2115	1.2125	35	44	54	63	71	80	88	96	1.2203
46	56	66	76	85	94	1.2202	1.2211	1.2219	1.2227	35
77	87	97	1.2207	1.2216	1.2225	33	42	50	58	66
1.2208	1.2219	1.2228	38	47	56	64	73	81	89	97
40	50	59	69	78	87	95	1.2304	1.2312	1.2320	1.2328
71	81	90	1.2300	1.2309	1.2318	1.2326	35	43	51	59

STRENGTH OF STEAM-BOILERS. VARIOUS R[ULES] FOR CONSTRUCTION.

There is a great lack of uniformity in the rules prescribed b[y differ]ent writers and by legislation governing the construction of stea[m-boilers]. In the United States, boilers for merchant vessels must be constr[ucted ac]cording to the rules and regulations prescribed by the Board of Su[pervising] Inspectors of Steam Vessels; in the U. S. Navy, according to ru[les of the] Navy Department, and in some cases according to special acts of C[ongress]. On land, in some places, as in Philadelphia, the construction of b[oilers is] governed by local laws; but generally there are no laws upon the [subject], and boilers are constructed according to the idea of individual engi[neers and] boiler-makers. In Europe the construction is generally regulated [by strin]gent inspection laws. The rules of the U. S. Supervising Insp[ectors of] Steam-vessels, the British Lloyd's and Board of Trade, the French [Bureau] Veritas, and the German Lloyd's are ably reviewed in a paper b[y Nathaniel] Foley, M. Inst. Naval Architects, etc., read at the Chicago Engineer[ing Con]gress, Division of Marine and Naval Engineering. From this paper [the fol]lowing notes are taken, chiefly with reference to the U. S. and Briti[sh rules].

(*Abbreviations.*—T. S., for tensile strength; El., elongation; Con[traction] traction of area.)

Hydraulic Tests.—*Board of Trade, Lloyd's, and Bureau V[eritas]*. Twice the working pressure.

United States Statutes.—One and a half times the working pres[sure].

Mr. Foley proposes that the proof pressure should be 1½ times th[e work]ing pressure + one atmosphere.

Established Nominal Factors of Safety.—*Board of* [*Trade*]. 4.5 for a boiler of moderate length and of the best construction a[nd work]manship.

Lloyd's.—Not very apparent, but appears to lie between 4 and 5.

United States Statutes.—Indefinite, because the strength of the [joint is] not considered, except by the broad distinction between single and [double] riveting.

Bureau Veritas: 4.4.

German Lloyd's: 5 to 4.65, according to the thickness of the plate.

Material for Riveting.—*Board of Trade.*—Tensile stre[ngth of] rivet bars between 26 and 30 tons, el. in 10″ not less than 25%, and c[ontraction of] area not less than 50%.

Lloyd's.—T. S., 26 to 30 tons; el. not less than 20% in 8″. The [material] must stand bending to a curve, the inner radius of which is not grea[ter than] 1½ times the thickness of the plate, after having been uniformly h[eated to] a low cherry-red, and quenched in water at 82° F.

United States Statutes.—No special provision.

Rules Connected with Riveting.—*Board of Trade.*—Th[e shear]ing resistance of the rivet steel to be taken at 23 tons per square i[nch, 5 to] be used for the factor of safety independently of any addition to th[e factor] for the plating. Rivets in double shear to have only 1.75 times th[e single] section taken in the calculation instead of 2. The diameter must no[t be less] than the thickness of the plate and the pitch never greater than 8¼ [inches. The] thickness of double butt-straps (each) not to be less than ⅝ the thic[kness of] the plate; single butt-straps not less than 9/8.

Distance from centre of rivet to edge of hole = diameter of rivet [× 1½].

Distance between rows of rivets

$$= 2 \times \text{diam. of rivet or} = [(\text{diam.} \times 4) + 1] \div 2, \text{ if chain, and}$$

$$= \frac{\sqrt{[(\text{pitch} \times 11) + (\text{diam.} \times 4)] \times (\text{pitch} + \text{diam.} \times 4)}}{10} \text{ if zig[zag].}$$

Diagonal pitch = (pitch × 6 + diam. × 4) ÷ 10.

Lloyd's.—Rivets in double shear to have only 1.75 times the single [section] taken in the calculation instead of 2. The shearing strength of rivet[s to] be taken at 85% of the T. S. of the material of shell plates. In [cases] where the strength of the longitudinal joint is satisfactorily show[n by ex]periment to be greater than given by the formula, the actual streng[th may] be taken in the calculation.

United States Statutes.—No rules.

Material for Cylindrical Shells Subject to Intern[al Pres]sure.—*Board of T*[...] between 27 and 32 tons. In the [...] el. not less [...], but should be about 25%; [...]

...han 20%. Strips 2″ wide should stand bending until the sides are ...el at a distance from each other of not more than three times the ...'s thickness.

...yd's.—T. S. between the limits of 26 and 30 tons per square inch. El. ...s than 20% in 8″. Test strips heated to a low cherry-red and plunged ...water at 82° F. must stand bending to a curve, the inner radius of ...h is not greater than 1½ times the plate's thickness.

...S. *Statutes.*—Plates of ½″ thick and under shall show a contr. of not ...than 50%; when over ½″ and up to ¾″, not less than 45%; when over ...not less than 40%.

...Foley's comments: The Board of Trade rules seem to indicate a steel ...o high T. S. when a lower and more ductile one can be got: the lower ...limit should be reduced, and the bending test might with advantage ...de after tempering, and made to a smaller radius. Lloyd's rule for ...y seems more satisfactory, but the temper test is not severe. The ...d States Statutes are not sufficiently stringent to insure an entirely ...factory material.

...r. Foley suggests a material which would meet the following: 25 tons ...r limit in tension; 25% in 8″ minimum elongation; radius for bending after tempering = the plate's thickness.

...hell-plate Formulæ.—*Board of Trade:* $P = \frac{T \times B \times t \times 2}{D \times F}$.

... = diameter of boiler in inches;
... = working-pressure in lbs. per square inch;
... = thickness in inches;
... = percentage of strength of joint compared to solid plate;
... = tensile strength allowed for the material in lbs. per square inch;
... = a factor of safety, being 4.5, with certain additions depending on method of construction.

...oyd's: $P = \frac{C \times (t - 2) \times B}{D}$.

... = thickness of plate in sixteenths; B and D as before; C = a constant ...ending on the kind of joint.

...hen longitudinal seams have double butt-straps, $C = 20$. When longi...nal seams have double butt-straps of unequal width, only covering on ...side the reduced section of plate at the outer line of rivets, $C = 19.5$. ...hen the longitudinal seams are lap-jointed, $C = 18.5$.

...S. *Statutes.*—Using same notation as for Board of Trade,

$$P = \frac{t \times 2 \times T}{D \times 6} \text{ for single-riveting; add 20\% for double-riveting;}$$

...re T is the lowest T. S. stamped on any plate.

...r. Foley criticises the rule of the United States Statutes as follows: The ... ignores the riveting, except that it distinguishes between single and ...ble, giving the latter 20% advantage; the circumferential riveting or ...s of seam is altogether ignored. The rule takes no account of workman... or method adopted of constructing the joints. The factor, one sixth, ...ly covers the actual nominal factor of safety as well as the loss of ...ngth at the joint, no matter what its percentage; we may therefore ...miss it as unsatisfactory.

...ules for Flat Plates.—*Board of Trade;* $P = \frac{C(t+1)^2}{S-6}$.

P = working-pressure in lbs. per square inch;
S = surface supported in square inches;
t = thickness in sixteenths of an inch;
C = a constant as per following table:

= 125 for plates not exposed to heat or flame, the stays fitted with nuts and washers, the latter at least three times the diameter of the stay and ⅔ the thickness of the plate;
= 187.5 for the same condition, but the washers ⅔ the pitch of stays in diameter, and thickness not less than plate;
= 200 for the *same condition, but doubling plates in place of washers, the width of which is ⅔ the pitch and thickness the same as the plate;*
= *112.5 for the same condition, but the stays with nuts only;*
75 when exposed to impact of heat or flame and steam ... the plates, and the stays fitted with nuts and washers ... diameter of the stay and ⅔ the plate's thickness;

$C = 67.5$ for the same condition, but stays fitted with nuts only;
$C = 100$ when exposed to heat or flame, and water in contact with the pl
and stays screwed into the plates and fitted with nuts;
$C = 66$ for the same condition, but stays with riveted heads.

U. S. Statutes.—Using same notation as for Board of Trade. $P = \frac{C}{}$
where p = greatest pitch in inches, P and t as above;

$C = 112$ for plates 7/16″ thick and under, fitted with screw stay-b and nuts, or plain bolt fitted with single nut and socket riveted head and socket;
$C = 120$ for plates above 7/16″, under the same conditions;
$C = 140$ for flat surfaces where the stays are fitted with nuts ins and outside;
$C = 200$ for flat surfaces under the same condition, but with the ad tion of a washer riveted to the plate at least ½ plate's thi ness, and of a diameter equal to 2/5 pitch.

N.B.—Plates fitted with double angle-irons and riveted to plate, with l at least ⅝ the thickness of plate and depth at least ¼ of pitch, would allowed the same pressure as determined by formula for plate with was riveted on.

N.B.—No brace or stay-bolt used in marine boilers to have a greater pi than 10½″ on fire-boxes and back connections.

Certain experiments were carried out by the Board of Trade which show that the resistance to bulging does not vary as the square of the plat thickness. There seems also good reason to believe that it is not invers as the square of the greatest pitch. Bearing in mind, says Mr. Foley, t mathematicians have signally failed to give us true theoretical foundati for calculating the resistance of bodies subject to the simplest form stresses, we therefore cannot expect much from their assistance in t matter of flat plates.

The Board of Trade rules for flat surfaces, being based on actual expe ment, are especially worthy of respect; sound judgment appears also have been used in framing them.

Furnace Formulæ.—Board of Trade.—*Long Furnaces.*—

$P = \frac{C \times t^2}{(L + 1) \times D}$, but not where L is shorter than $(11.5t - 1)$, at which leng the rule for short furnaces comes into play.

P = working-pressure in pounds per square inch; t = thickness in inch
D = outside diameter in inches; L = length of furnace in feet up to 10
C = a constant, as per following table, for drilled holes:

$C = 99{,}000$ for welded or butt-jointed with single straps, dou riveted;
$C = 88{,}000$ for butts with single straps, single-riveted;
$C = 99{,}000$ for butts with double straps, single-riveted.

Provided always that the pressure so found does not exceed that give the following formulæ, which apply also to short furnaces:

$P = \frac{C \times t}{D}$ for all the patent furnaces named;

$P = \frac{C \times t}{3 \times D}\left(5 - \frac{L \times 12}{67.5 \times t}\right)$ when with Adamson rings.

$C = 8{,}800$ for plain furnaces;
$C = 14{,}000$ for Fox; minimum thickness 5/16″, greatest ⅝″; plain not to exceed 6″ in length;
$C = 13{,}500$ for Morison; minimum thickness 5/16″, greatest ⅝″; p part not to exceed 6″ in length;
$C = 14{,}000$ for Purves-Brown; limits of thickness 7/16″ and ⅝″; p part 9″ in length;
$C = 8{,}800$ for Adamson rings; radius of flange next fire 1½″.

U. S. Statutes.—*Long Furnaces.*—Same notation.

$P = \frac{89{,}600 \times t^2}{L \times D}$, but L not to exceed 8 ft.

N.B.—If rings of wrought iron are fitted and riveted on properly an and to the flue in such a manner that the tensile stress on the rivets s

5000 lbs. per sq. in., the distance between the rings shall be taken h of the flue in the formulæ.

rnaces, Plain and Patent.—P, as before, when not 8 ft.

$\frac{0 \times t^2}{\times D}$;

$\frac{C}{}$ when

$C = 14,000$ for Fox corrugations where $D =$ mean diameter;
$C = 14,000$ for Purves-Brown where $D =$ diameter of flue;
$C = 5677$ for plain flues over 16″ diameter and less than 40″, when not over 3 ft. lengths.

comments on the rules for long furnaces as follows: The Board neral formula, where the length is a factor, has a very limited ed, viz., 10 ft. as the extreme length, and 135 thicknesses — 12″, t limit. The original formula, $P = \frac{C \times t^2}{L \times D}$, is that of Sir W. and was, I believe, never intended by him to apply to short fur- the very face of it, it is apparent, on the other hand, that if it is derately long furnaces, it cannot be so for very long ones. We re driven to the conclusion that any formula which includes a factor must be founded on a wrong basis.

Traill's form of the formula, namely, substituting $(L + 1)$ for L, appear sufficiently satisfactory for practical purposes, and in- r as can be judged, tally with the results obtained from experi- arly as could be expected. The experiments to which I refer number, and of great variety of length to diameter; the actual afety ranged from 4.4 to 6.2, the mean being 4.78, or practically to me, therefore, that, within the limits prescribed, the Board of ula may be accepted as suitable for our requirements.

ed States Statutes give Fairbairn's rule pure and simple, except treme limit of length to which it applies is fixed at 8 feet. As be seen, no limit for the shortest length is prescribed, but the are by no means clear, flues and furnaces being mixed or not uished.

al for Stays.—The qualities of material prescribed are as

Trade.—The tensile strength to lie between the limits of 27 and square inch, and to have an elongation of not less than 20% in stays which have been welded or worked in the fire should not

-26 to 30 ton steel, with elongation not less than 20% in 8″.

tutes.—The only condition is that the reduction of area must not 40% if the test bar is over ¾″ diameter.

llowed on Stays.—*Board of Trade.*—9000 lbs. per square ved on the net section, provided the tensile strength ranges from s. Steel stays are not to be welded or worked in the fire.

For screwed and other stays, not exceeding 1½″ diameter effec- s. per square inch is allowed; for stays above 1½″, 9000 lbs. No be welded.

utes.—Braces and stays shall not be subjected to a greater stress s. per square inch.

S. E., p. 459, says: "The iron of the stays ought not to be ex- greater working tension than 3000 lbs. on the square inch, in vide against their being weakened by corrosion. This amounts the factor of safety for the working pressure about 20." It is vever, that an allowance in the factor of safety for corrosion may be decreased with increase of diameter. W. K.]

.—*Board of Trade.* $P = \frac{C \times d^2 \times t}{(W - p)D \times L}$. $P =$ working pres- per sq. in.; $W =$ width of flame-box in inches; $L =$ length of hes; $p =$ pitch of bolts in inches; $D =$ distance between girders to centre in inches; $d =$ depth of girder in inches; $t =$ thick- of same in inches; $C =$ a constant $= 6600$ for 1 bolt, 9900 for 2 nd 11,220 for 4 bolts.

The same formula and constants, except that $C = 11,000$ for 4 or 0 for 6 or 7, and 11,880 for 8 or more.

utes.—The matter appears to be left to the designers.

Tube-Plates.—*Board of Trade.* $P = \frac{t(D - d) \times 20{,}000}{W \times D}$ [illegible]
horizontal distance between centres of tubes in inches; d = inside [illegible]
of ordinary tubes; t = thickness of tube-plate in inches; W = [illegible]
width of combustion-box in inches from front tube-plate to back [illegible]
box, or distance between combustion-box tube-plates when the [illegible]
double-ended and the box common to both ends.

The crushing stress on tube-plates caused by the pressure on the [illegible]
box top is to be limited to 10,000 lbs. per square inch.

Material for Tubes.—Mr. Foley proposes the following: If [illegible]
quality to be such as to give at least 22 tons per square inch as the [illegible]
tensile strength, with an elongation of not less than 15% in 8". If [illegible]
elongation to be not less than 26% in 8" for the material before being [illegible]
into strips; and after tempering, the test bar to stand completely [illegible]
together. Provided the steel welds well, there does not seem to be [illegible]
ject in providing tensile limits.

The ends should be annealed after manufacture, and stay-tube ends [illegible]
be annealed before screwing.

Holding-power of Boiler-tubes.—Experiments made [illegible]
ington Navy Yard show that with 2½ in. brass tubes in no case was the [illegible]
power less, roughly speaking, than 6000 lbs., while the average was [illegible]
of 20,000 lbs. It was further shown that with these tubes nuts were [illegible]
fluous, quite as good results being obtained with tubes simply expan[illegible]
the tube-plate and fitted with a ferrule. When nuts were fitted it wa[illegible]
that they drew off without injuring the threads.

In Messrs. Yarrow's experiments on iron and steel tubes of [illegible]
diameter the first 5 tubes gave way on an average of 23,740 lbs., which [illegible]
appear to be about ⅝ the ultimate strength of the tubes themselves [illegible]
these cases the hole through the tube-plate was parallel with a sl[illegible]
to it, and a ferrule was driven into the tube.

Tests of the next 5 tubes were made under the same conditions as [illegible]
5, with the exception that in this case the ferrule was omitted, the [illegible]
ing simply expanded into the plates. The mean pull required was [illegible]
or considerably less than half the ultimate strength of the tubes.

Effect of beading the tubes, the holes through the plate being par[illegible]
ferrules omitted. The mean of the first 3, which are tubes of [illegible]
kind, gives 26,876 lbs. as their holding-power, under these conditions [illegible]
pared with 23,740 lbs. for the tubes fitted with ferrules only. [illegible]
figure is, however, mainly due to an exceptional case where the [illegible]
power is greater than the average strength of the tubes themselves [illegible]

It is disadvantageous to cone the hole through the tube-plate [illegible]
sharp edge is removed, as the results are much worse than those [illegible]
with parallel holes, the mean pull being but 16,031 lbs., the experim[illegible]
ing made with tubes expanded and ferruled but not beaded over.

In experiments on tubes expanded into tapered holes, beaded [illegible]
fitted with ferrules, the net result is that the holding-power is, for [illegible]
experimented on, about ¾ of the tensile strength of the tube, the m[illegible]
being 28,797 lbs.

With tubes expanded into tapered holes and simply beaded ov[illegible]
results were obtained than with ferrules; in these cases, however, [illegible]
edge of the hole was rounded off, which appears in general to have [illegible]
effect.

In one particular the experiments are incomplete, as it is impo[illegible]
reproduce on a machine the racking the tubes get by the expan[illegible]
boiler as it is heated up and cooled down again, and it is quite [illegible]
therefore, that the fastening giving the best results on the testing [illegible]
may not prove so efficient in practice.

N.B.—It should be noted that the experiments were all made [illegible]
cold condition, so that reference should be made with caution, the [illegible]
stances in practice being very different, especially when there is [illegible]
the tube-plates, or when the tube-plates are thick and subject to [illegible]
heat.

Iron versus Steel Boiler-tubes. (Foley.)—Mr. Ble[illegible]
prefers iron tubes to those of steel, but how far he would go in att[illegible]
the leaky-tube defect to the use of steel tubes we are not aware. [illegible]
however, that the results of his experiments would warrant him [illegible]
[illegible] distance in this direction. The test consisted of [illegible]
[illegible] two tubes, one [illegible] wrought iron and the other of steel, [illegible]
[illegible] in. in diam[illegible] [illegible]16 in. thickness of metal. Th[illegible]

ame furnace, made red-hot, and then dipped in water. The
auged at a temperature of 46° F.
ion was twice repeated, with results as follows:

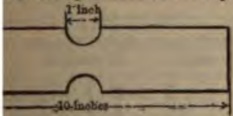
	Steel.	Iron.
th....................................	55.495 in.	55.495 in.
° F.; increase..........	.052 "	.048 "
expansion per degree F.........	.0000067	.0000062
ot and dipped in water; decrease	.007 in.	.003 in.
ng and cooling, decrease.........	.031 in.	.004 in.
g and cooling, decrease.....	.017 in.	.006 in.
traction	.055 in.	.013 in.

Kirk writes: That overheating of tube ends is the cause of the
e tubes in boilers is proved by the fact that the ferrules at
by the Admiralty prevent it. These act by shielding the tube
e action of the flame, and consequently reducing evaporation,
ing free access of the water to keep them cool.
many causes contribute, there seems no doubt that thick tube-
bear a share of causing the mischief.

Construction of Boilers in Merchant Vessels in the United States.

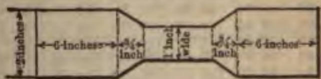
om General Rules and Regulations of the Board of Supervising
spectors of Steam-vessels (as amended 1893 and 1894).)

Strength of Plate. (Section 3.)—To ascertain the tensile
l other qualities of iron plate there shall be taken from each



sheet to be used in shell or other parts of boiler which are subject to tensile strain a test piece prepared in form according to the following diagram, viz.: 10 inches in length, 2 inches in width, cut out in the centre in the manner indicated.

To ascertain the tensile strength
ualities of steel plate, there shall be taken from each sheet to be
l or other parts of boiler which are subject to tensile strain, a test-
red in form according
ring diagram, the length
part in centre varying as
y different thickness of
follows:



ght portion shall be in
ast eight times the width multiplied by the thickness of said part,
reduction of area as called for by the present rules of the Board,
gation of at least 25%. The straight part shall be of a width of 1
rule to take effect on and after July 1, 1894.
that where contracts for boilers for ocean-going steamers re-
of material in compliance with the British Board of Trade,
yd's, or Bureau Veritas rules for testing, the inspectors shall
sts in compliance with the following rules:
es shall in all cases to have an ultimate elongation not less than 20%
of 8 inches. It is to be capable of being bent to a curve of which
dius is not greater than one and a half times the thickness of
after having been heated uniformly to a low cherry-red, and
water of 82° F.
1894 the shape of test-piece for steel was the same as that for iron,
oved shape. This shape has been condemned by authorities on
materials for over twenty years. It always gives results which
, the error sometimes amounting to 25 per cent. See pages 242,
lso, Strength of Materials, W. Kent, Van N. Science Series No. 41,
lee on Wrought-iron and Chain Cables.]

ty. (Section 6.)—To ascertain the ductility and other lawful
on of 45,000 lbs. tensile *strength shall show a contraction of area*
ent, *and each additional 1000* lbs. tensile strength shall show 1
ditional contraction of area, up to and including 55,000 ...
on of 55,000 tensile strength and upwards, showing
area, shall be deemed to have the lawful ducti...
h thickness and under shall show a contraction ...
r cent. Steel plate over ½ inch in thickness, u...

Thickness in 16ths of an Inch.	Diameter in Inches.									
	54	60	66	72	78	84	90	96	102	108
1	16.2	14.6	13.3	12.2	11.2	10.4	9.7	9.1	8.6	8.1
2	32.4	29.2	26.5	24.3	22.4	20.8	19.4	18.2	17.2	16.2
3	48.6	43.7	39.8	36.5	33.7	31.3	29.2	27.3	25.7	24.3
4	64.8	58.3	53.0	48.6	44.9	41.7	38.9	36.5	34.3	32.4
5	81.0	72.9	66.3	60.8	56.1	52.1	48.6	45.6	42.9	40.5
6	97.2	87.5	79.5	72.9	67.3	62.5	58.3	54.7	51.5	48.6
7	113.4	102.1	92.8	85.1	78.5	72.9	68.1	63.8	60.0	56.7
8	129.6	116.7	106.1	97.2	89.7	83.3	77.8	72.9	68.6	64.8
9	145.8	131.2	119.3	109.4	101.0	93.8	87.5	82.0	77.2	72.9
10	162.0	145.8	132.6	121.5	112.2	104.2	97.2	91.1	85.8	81.0
11	178.2	160.4	145.8	133.7	123.4	114.6	106.9	100.3	94.4	89.1
12	194.4	175.0	159.1	145.8	134.6	125.0	116.7	109.4	102.9	97.2
13	210.7	189.6	172.4	158.0	145.8	135.4	126.4	118.5	111.5	105.3
14	226.9	204.2	185.6	170.1	157.1	145.8	136.1	127.6	120.1	113.4
15	243.1	218.7	198.9	182.3	168.3	156.3	145.8	136.7	128.7	121.5
16	259.3	233.3	212.1	194.4	179.5	166.7	155.6	145.8	137.3	129.6

Rules governing Inspection of Boilers in Philad

In estimating the strength of the longitudinal seams in the shells of boilers the inspector shall apply two formulæ, A and B:

A, $\left\{ \dfrac{\text{Pitch of rivets} - \text{diameter of holes punched to receive the}}{\text{pitch of rivets}} \right.$

percentage of strength of the sheet at

B, $\left\{ \dfrac{\text{Area of hole filled by rivet} \times \text{No. of rows of rivets in seam} \times \text{ing strength of rivet}}{\text{pitch of rivets} \times \text{thickness of sheet} \times \text{tensile strength of sh}} \right.$

percentage of strength of the rivets in

Take the lowest of the percentages as found by formulæ A apply that percentage as the "strength of the seam" in the formula C, which determines the strength of the longitudinal seam

C, $\left\{ \dfrac{\text{Thickness of sheet in parts of inch} \times \text{strength of seam as ob by formula A or B} \times \text{ultimate strength of iron stamped on p}}{\text{internal radius of boiler in inches} \times 5 \text{ as a factor of safe}} \right.$

safe working

TABLE OF PROPORTIONS AND SAFE WORKING PRESSURES WITH FO AND C, @ 50,000 LBS., T.S.

Diameter of rivet.	5/8″	11/16	3/4	13/16
Diameter of rivet-hole.	11/16″	3/4	13/16	7/8
Pitch of rivets.	2″	2 1/16	2 1/8	2 3/16
Strength of seam, %.	.656	.636	.62	.60
Thickness of plate.	1/4″	5/16	3/8	7/16
Diameter of boiler, in...	Safe Working Pressure with Longitudin Single-riveted.			
24	137	165	193	220
30	109	132	154	176
32	102	124	144	165
34	96	117	136	155
36	91	110	129	147
38	86	104	122	139
40	82	99	116	132
44	74	91	105	120
48	68	83	96	110
54	60	73	86	[illegible]
60	55	66	77	[illegible]

of rivet........	5/8″	11/16	3/4	13/16	7/8
of rivet-hole...	11/16″	3/4	13/16	7/8	15/16
ivets...	3″	3 1/8	3 1/4	3 3/8	3 1/2
of seam, %.	.77	.76	.75	.74	.73
of plate..	1/4″	5/16	3/8	7/16	1/2
of boiler, in...	Safe Working Pressure with Longitudinal Seams, Double-riveted.				
24	160	198	235	269	305
30	127	158	188	215	243
32	119	148	176	202	228
34	112	140	166	190	215
36	106	132	156	179	203
38	101	125	148	170	192
40	96	119	141	161	183
44	87	108	128	147	166
48	79	99	118	135	152
54	70	88	104	120	135
60	64	79	94	108	122

s and Tubes for Steam-boilers.—(From Rules of U. S. sing Inspectors. Steam-pressures per square inch allowable on and lap-welded flues made in sections. Extract from table in Rules Supervising Inspectors.)

ast thickness of material allowable, D = greatest diameter in inches, wable pressure. For thickness greater than T with same diameter reased in the ratio of the thickness.

	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
	.18	.20	.21	.21	.22	.22	.23	.24	.25	.26	.27	.28	.29	30	.31	.32	.33
.	189	184	179	174	172	158	152	147	143	139	136	134	131	129	126	125	122
	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40
	.34	.35	.36	.37	.38	.39	.40	.41	.42	.43	.44	.45	.46	.47	.48	.49	.50
.	121	120	119	117	116	115	115	114	112	112	110	110	109	109	108	108	107

iameters not over 10 inches the greatest length of section allowable t; for diameters 10 to 23 inches, 3 feet; for diameters 23 to 40 inches, 30 If lengths of sections are greater than these lengths, the allowable e is reduced proportionately.

J. S. rule for corrugated flues, as amended in 1894, is as follows: Rule tion 14. The strength of all corrugated flues, when used for furnaces m chimneys (corrugation not less than 1 1/2 inches deep and not exceed- nches from centres of corrugation), and provided that the plain parts ends do not exceed 6 inches in length, and the plates are not less than ch thick, when new, corrugated, and practically true circles, to be ated from the following formula:

$$\frac{14,000}{D} \times T = \text{pressure}.$$

thickness, in inches; D = mean diameter in inches.

ed Flues.—The same formula is given for ribbed flues, with rib tions not less than 1 3/8 inches deep and not more than 9 inches

t Stayed Surfaces in Steam-boilers.—Rule II., Section 6, of les of the U. S. Supervising Inspectors provides as follows:

braces or stays hereafter employed in the construction of boilers be allowed a greater strain than 6000 lbs. per square inch of n.

k, in his treatise on the Steam-engine, also in his Pocket-book, gi llowing formula: $p = 407ts \div d$, in which p is the internal pressure s per square inch that will strain the plates to their elastic limit, ckness *of the plate in inches, d is the distance between two rows olts in the clear, and s is the tensile stress in the plate per square inch, at the elastic limit.* Substituti *steel, and copper, 12, 14, and 8 tons* respectively ;

FORMULÆ FOR ULTIMATE ELASTIC STRENGTH OF FLAT STAYED

	Iron.	Steel.
Pressure....................	$p = 5000\frac{t}{d}$	$p = 5700\frac{t}{d}$
Thickness of plate..........	$t = \frac{p \times d}{5000}$	$t = \frac{p \times d}{5700}$
Pitch of bolts..............	$d = \frac{5000t}{p}$	$d = \frac{5700t}{p}$

For Diameter of the Stay-bolts, Clark gives $d' =$

in which d' = diameter of screwed bolt at bottom of thread, P = nal and P' transverse pitch of stay-bolts between centres, p = pressure in lbs. per sq. in. that will strain the plate to its elastic elastic strength of the stay-bolts in lbs. per sq. in. Taking s = tons, respectively for iron, steel, and copper, we have

For iron, $d' = .00069 \sqrt{PP'p}$, or if $P = P'$, $d' = .00069P$
For steel, $d' = .00064 \sqrt{PP'p}$, " " $d' = .00064P$
For copper, $d' = .00084 \sqrt{PP'p}$, " " $d' = .00084P$

In using these formulæ a large factor of safety should be taken for reduction of size by corrosion. Thurston's Manual of Steam- 144, recommends that the factor be as large as 15 or 20. The Steam Boiler Insp. & Ins. Co. recommends not less than 10.

Strength of Stays.—A. F. Yarrow (*Engr.*, March 20, 1891) following results of experiments to ascertain the strength of stays:

Description.	Length between Plates.	Diameter of Stay over Threads.
Hollow stays screwed into plates and hole expanded	4.75 in.	1 in.(hole 7/16 in. and 5/16 in.
	4.64 in.	1 in.(hole 9/16 in. and 7/16 in.
Solid stays screwed into plates and riveted over.	4.80 in.	⅞ in.
	4.80 in.	⅝ in.

The above are taken as a fair average of numerous tests.

Stay-bolts in Curved Surfaces, as in Water-legs of cal Boilers.—The rules of the U. S. Supervising Inspectors follows: All vertical boiler-furnaces constructed of wrought plates, and having a diameter of over 42 in. or a height of over 40 in. stayed with bolts as provided by § 6 of Rule II, for flat surfaces; thickness of material required for the shells of such furnaces shall termined by the distance between the centres of the stay-bolts in nace and not in the shell of the boiler; and the steam-pressure shall be determined by the distance from centre of stay-bolts in the and the diameter of such stay-bolts at the bottom of the thread.

The Hartford Steam-boiler Insp. & Ins. Co. approves the above *Locomotive*, March, 1892) as far as it states that curved surfaces computed the same as flat ones, but prefers Clark's formulæ stayed surfaces to the rules of the U. S. Supervising Inspectors.

Fusible-plugs.—Fusible-plugs should be put in that portion heating-surface which first becomes exposed from lack of water. of the U. S. Supervising Inspectors specify Banca tin for the melting-point is about 445° F. The rule says: All steamers inserted in their boilers plugs of Banca tin, at least ½ in. in diameter smallest end of the internal opening, in the following manner: *Cylinder*-boilers with flues shall have one plug inserted in one boiler; and also one plug inserted in the shell of each boiler line and not less than 4 ft. from immediately before boilers shall have one plug end of the boiler. in the highest fire-surface of the back upright tubular boilers used for marine purposes shall have a fusible plug inserted in one of the tubes at a point at least 2 in. below the lower gauge-cock, and said plug may be placed in the upper head sheet when deemed advisable by the local inspectors.

Steam-domes.—Steam domes or drums were formerly almost universally used on horizontal boilers, but their use is now generally discontinued, as they are considered a useless appendage to a steam-boiler, and unless properly designed and constructed are an element of weakness.

Height of Furnace.—Recent practice in the United States makes the height of furnace much greater than it was formerly. With large sizes of anthracite there is no serious objection to having the furnace as low as 12 or 18 in., measured from the surface of the grate to the nearest portion of the heating surface of the boiler, but with coal containing much volatile matter and moisture a much greater distance is desirable. With very volatile coals the distance may be as great as 4 or 5 ft. Rankine (S. E., p. 457) says: The clear height of the "crown" or roof of the furnace above the grate-bars is seldom less than about 18 in., and often considerably more. In the fire-boxes of locomotives it is on an average about 4 ft. The height of 18 in. is suitable where the crown of the furnace is a brick arch. Where the crown of the furnace, on the other hand, forms part of the heating-surface of the boiler, a greater height is desirable in every case in which it can be obtained; for the temperature of the boiler-plates, being much lower than that of the flame, tends to check the combustion of the inflammable gases which rise from the fuel. As a general principle a high furnace is favorable to complete combustion.

IMPROVED METHODS OF FEEDING COAL.

Mechanical Stokers. (William R. Roney, Trans. A. S. M. E., vol. xii.)—Mechanical stokers have been used in England to a limited extent since 1785. In that year one was patented by James Watt. It was a simple device to push the coal, after it was coked at the front end of the grate, back towards the bridge. It was worked intermittently by levers, and was designed primarily to prevent smoke from bituminous coal. (See D. K. Clark's Treatise on the Steam-engine.)

After the year 1840 many styles of mechanical stokers were patented in England, but nearly all were variations and modifications of the two forms of stokers patented by John Jukes in 1841, and by E. Henderson in 1843.

The Jukes stoker consisted of longitudinal fire-bars, connected by links, so as to form an endless chain, similar to the familiar treadmill horse-power. The small coal was delivered from a hopper on the front of the boiler, on to the grate, which slowly moving from front to rear, gradually advanced the fuel into the furnace and discharged the ash and clinker at the back.

The Henderson stoker consists primarily of two horizontal fans revolving on vertical spindles, which scatter the coal over the fire.

Numerous faults in mechanical construction and in operation have limited the use of these and other mechanical stokers. The first American stoker was the Murphy stoker, brought out in 1878. It consists of two coal magazines placed in the side walls of the boiler furnace, and extending back from the boiler front 6 or 7 feet. In the bottom of these magazines are rectangular iron boxes, which are moved from side to side by means of a rack and pinion, and serve to push the coal upon the grates, which incline at an angle of about 35° from the inner edge of the coal magazines, forming a V-shaped receptacle for the burning coal. The grates are composed of narrow parallel bars, so arranged that each alternate bar lifts about an inch at the lower end, while at the bottom of the V, and filling the space between the ends of the grate-bars, is placed a cast-iron toothed bar, arranged to be turned by a crank. The purpose of this bar is to grind the clinker coming in contact with it. Over this V-shaped receptacle is sprung a fire-brick arch.

In the Roney mechanical stoker the fuel to be burned is dumped into a hopper on the boiler front. Set in the lower part of the hopper is a "pusher" to which is attached the "feed-plate" forming the bottom of the hopper. The "pusher," by a vibratory *motion, carrying* with it the "feed-plate," gradually *forces the fuel over the* "dead-plate" and on the grate. The *grate-bars, in their normal condition* form a series of steps, to the top step *which coal is fed from the* "dead-plate." Each bar rests in a concave *in the bearer, and is capable of a* rocking motion thro... adjustab... *All the grate-bars are coupled* together by a "ro... A v... *back-and-forth motion* being given to the "rocker-...

necting-rod, the grate-bars rock in unison, now forming a series and now approximating to an inclined plane, with the grates pa lapping, like shingles on a roof. When the grate-bars rock forwar will tend to work down in a body. But before the coal can mov the bars rock back to the stepped position, checking the downward breaking up the cake over the whole surface, and admitting a fre of air through the fire. The rocking motion is slow, being from strokes per minute, according to the kind of coal. This alternate and checking motion is continuous, and finally lands the cinder an the dumping-grate below.

Mr. Roney gives the following record of six tests to determine parative economy of the Roney mechanical stoker and hand-firing tubular boilers, 60 inches × 20 feet, burning Cumberland coal with draught. Rating of boiler at 12.5 square feet, 105 H. P.

	Three tests, hand-firing.			Three tests,	
Evaporation per pound, dry coal from and at 212° lbs	10.36	10.44	11.00	11.89	12.25
H.P. developed above rating, %	5.8	13.5	68	54.6	66.7

Results of comparative tests like the above should be used with in drawing generalizations. It by no means follows from these res a stoker will always show such comparative excellence, for in this results of hand-firing are much below what may be obtained un le able circumstances from hand-firing with good Cumberland coal.

The Hawley Down-draught Furnace.—A foot or mo the ordinary grate there is carried a second grate composed of a water-tubes, opening at both ends into steel drums or headers, throu water is circulated. The coal is fed on this second grate, and as tially consumed falls through it upon the lower grate, where the com is completed in the ordinary manner. The draught through the co upper grate is downward through the coal and the grate. The volat are therefore carried down through the bed of coal, where they a oughly heated, and are burned in the space beneath, where they m excess of hot air drawn through the fire on the lower grate. In Chicago, from 30 to 45 lbs. of coal were burned per square foot of gra this system, with good economical results. (See catalogue of the Down Draught Furnace Co., Chicago, 1894.)

Under-feed Stokers.—Results similar to those that may be with downward draught are obtained by feeding the coal at the bottom bed, pushing upward the coal already on the bed which has had its matter distilled from it. The volatile matter of the freshly fired co has to pass through a body of ignited coke. (See circular of the Jo der-feed Stoker, Fraser & Chalmers, Chicago, 1894.)

SMOKE PREVENTION.

A committee of experts was appointed in St. Louis in 1891 to repor smoke problem. A summary of its report is given in the *Iron Age* 7, 1892. It describes the different means that have been tried to smoke, such as gas-fuel, steam-jets, fire-brick arches and checke hollow walls for preheating air, coking arches or chambers, double tion furnaces, and automatic stokers. All of these means have been less effective in diminishing smoke, their effectiveness depending upon the skill with which they are operated; but none is entirely tory. Fuel-gas is objectionable chiefly on account of its expen average quality of fuel-gas made from a trial run of several car Illinois coal, in a well-designed fuel-gas plant, showed a calorific 243,391 heat-units per 1000 cubic feet. This is equivalent to 5022.8 h per lb. of coal, whereas by direct calorimeter test an average sampl coal gave 11,172 heat-units. One lb. of the coal showed a theoretic oration of 11.56 lbs. water, while the gas from 1 lb. showed a the evaporation of 5.23 lbs. 48.17 lbs. of coal were required to furnish 10 feet of the gas. In 39 tests the smoke-preventing furnaces showed of the capacity of the common furnaces, reduced the work of the 28%, and required about 2% more fuel to do the same work. In ma steam-jets the fuel co[illegible]ion was increased 12% for the same Prof. O. H. Land[illegible] port to the State Board of Health [illegible] 1893), writes as follows on t (*Engineering* [illegible] prevention steam-boilers, the object must be attained by one or more agencies:

gn and setting of the boiler-plant. This implies proper grate draught, the necessary air-space between grate-bars and e, and ample combustion-room under boilers.

n of firing that is best adapted to each particular furnace to ct combustion of bituminous coal. This may be either: (*a*) r charging all coal into the front of the furnace until paren pushing back and spreading; or (*b*) "alternate side-firreading," by which the coal is spread over the whole grate form layers at each charging.

ion of air through the furnace-door, bridge-wall, or side walls, and other artificial means for thoroughly mixing the air and ses.

the cooling of the furnace and boilers by the inrush of cold rnace-doors are opened for charging coal and handling the

g a gradation of the several steps of combustion so that the arged, dried, and warmed at the coolest part of the furnace, d by successive steps to the hottest place, where the final the coked coal is completed, and compelling the distilled ses to pass through this hottest part of the fire.

the cooling by radiation of the unburned combustible gases ixing and combustion have been accomplished.

e supply of air to suit the periodic variation in demand.

tution of a continuous uniform feeding of coal instead of arging.

ught burning or causing the air to enter above the grate and ugh the coal, carrying the distilled products down to the high ane at the bottom of the fire.

of smoke-prevention devices which have been invented is classification is:

al stokers. They effect a material saving in the labor of efficient smoke-preventers when not pushed above their hen the coal does not cake badly. They are rarely susceptilen changes in the rate of firing frequently demanded in

in side walls, bridge-wall, and grate-bars, through which air s heated. The results are always beneficial, but the flues are clean and in order.

ches, or spaces in front of the furnace arched over, in which s coked, both to prevent cooling of the distilled gases, and to ass through the hottest part of the furnace just beyond the ilts are good for normal conditions, but ineffective when the d. The arches also are easily burned out and injured by e.

tes, or a portion of the grate next the furnace-doors, reserved nd coking the coal before it is spread over the grate. These lts when the furnace is not forced above its normal capacity. the method of "coke-firing" mentioned before.

ught furnaces, or furnaces in which the air is supplied to the grate, and the products of combustion are taken away from ate, thus causing a downward draught through the coal, carryl gases down to the highly heated incandescent coal at the ayer of coal on the grate. This is the most perfect manner ombustion, and is absolutely smokeless.

ts to draw air in or inject air into the furnace above the grate, the air and the combustible gases together. A very efficient er, but one liable to be wasteful of fuel by inducing too rapid

ates placed in the furnace above the fire to aid in mixing the ases with the air.

urnaces, of which there are two different styles; the first of he second grate *below the first* grate; the coal is coked on the *ing which process the distilled* gases are made to pass over *e, where they are ignited* and burned; the coke from the first *d onto the second grate*: a very efficient and economical *but rather complicated to* construct and maintain. In the *products of combustion* from the first furnace pr the grate and fire of the second, each furnace being charged with f[illegible] when needed, the latter generally with a smokeless coal or coke; [illegible]tional and unpromising method.

Mr. C. F. White, Consulting Engineer to the Chicago Society for [illegible]vention of Smoke, writes under date of May 4, 1893:

The experience had in Chicago has shown plainly that it is perfe[illegible] to equip steam-boilers with furnaces which shall burn ordinary so[illegible] such a manner that the making of smoke dense enough to obstruct [illegible] shall be confined to one or two intervals of perhaps a couple of [illegible] duration in the ordinary day of 10 hours.

Gas-fired Steam-boilers.—Converting coal into gas in a [illegible] producer, before burning it under the steam-boiler, is an ideal m[illegible] smoke-prevention, but its expense has hitherto prevented its gener[illegible] duction. A series of articles on the subject, illustrating a great nu[illegible] devices, by F. J. Rowan, is published in the *Colliery Engineer*, 188[illegible] also Clark on the Steam-engine.

FORCED COMBUSTION IN STEAM-BOILERS

For the purpose of increasing the amount of steam that can be ge[illegible] by a boiler of a given size, forced draught is of great importance [illegible] universally used in the locomotive, the draught being obtained by a [illegible] jet in the smoke-stack. It is now largely used in ocean steamers, e[illegible] in ships of war, and to a small extent in stationary boilers. Economy is generally not attained by its use, its advantages being confine[illegible] securing of increased capacity from a boiler of a given bulk, weight, [illegible] The subject of forced draught is well treated in a paper by James H[illegible] entitled, "Forced Combustion in Steam-boilers" (Section G, Engi[illegible] Congress at Chicago, in 1893), from which we abstract the following:

Edwin A. Stevens at Bordentown, N. J., in 1827, in the steamer [illegible] America," fitted the boilers with closed ash-pits, into which the air [illegible]bustion was forced by a fan. In 1828 Ericsson fitted in a similar ma[illegible] steamer "Victory," commanded by Sir John Ross.

Messrs. E. A. and R. L. Stevens continued the use of forced dra[illegible] a considerable period, during which they tried three different modes [illegible] the fan for promoting combustion: 1, blowing direct into a closed [illegible] 2, exhausting the base of the funnel by the suction of the fan; 3, fo[illegible] into an air-tight boiler-room or stoke-hold. Each of these three [illegible] was attended with serious difficulties.

In the use of the closed ash-pit the blast-pressure would frequen[illegible] the gases of combustion, in the shape of a serrated flame, from [illegible] around the furnace doors in so great a quantity as to affect both [illegible]ciency and health of the firemen.

The chief defect of the second plan was the great size of the fan [illegible] to produce the necessary exhaustion. The size of fan required gr[illegible] rapidly increasing ratio as the combustion increases, both on accou[illegible] greater air-supply and the higher exit temperature enlarging the v[illegible] the waste gases.

The third method, that of forcing cold air by the fan into an [illegible] boiler-room—the present closed stoke-hold system—though it over[illegible] difficulties in working belonging to the two forms first tried, ha[illegible] defects of its own, as it cannot be worked, even with modern [illegible] boiler-construction, much, if at all, above the power of a good [illegible] draught, in most boilers, without damaging them.

In 1875 John I. Thornycroft & Co., of London, began the constr[illegible] torpedo-boats with boilers of the locomotive type, in which a hig[illegible] combustion was attained by means of the air-tight boiler-room, i[illegible] air was forced by means of a fan.

In 1882 H.B.M. ships "Satellite" and "Conqueror" were fitted [illegible] system, the former being a small ship of 1500 I.H.P., and the latte[illegible] clad of 4500 I.H.P. On the trials with forced draught, which lasted [illegible] to three hours each, the highest rates of combustion gave 16.9 I[illegible] square foot of fire-grate in the "Satellite," and 13.41 I.H.P. in [illegible]queror."

None of the short trials at these rates of combustion were mad[illegible] injury to the seams and tubes of the boilers, but the system wa[illegible] and it has been continued in the British Navy to this day (1893).

In Mr. Howden's [illegible] advantage arising from increased [illegible] over natural-draug[illegible] derived from using forced draught [illegible] ash-pit sufficient [illegible] the disadvantages arising fr[illegible] king, there being either excessive smoke from bituminous coal or d evaporative economy.

880 Mr. Howden designed an arrangement intended to overcome the s of both the closed ash-pit and closed stoke-hold systems.

ir-tight reservoir or chamber is placed on the front end of the boiler rrounding the furnaces. This reservoir, which projects from 8 to 10 from the end of the boiler, receives the air under pressure, which is d by the valves into the ash-pits and over the fires in proportions to the kind of fuel used and the rate of combustion required. The ed above the fires is admitted to a space between the outer and inner ce-doors, the inner having perforations and an air-distributing box gh which the air passes under pressure.

means of the balance of air-pressure above and below the fires all ncy for the fire to blow out at the furnace-door is removed.

regulating the admission of the air by the valves above and below the the highest rate of combustion possible by the air-pressure used can fected, and in same manner the rate of combustion can be reduced to elow that of natural draught, while complete and economical combus- at all rates is secured.

feature of the system is the combination of the heating of the air of bustion by the waste gases with the controlled and regulated admission r to the furnaces. This arrangement is effected most conveniently by ng the hot fire-gases after they leave the boiler through stacks of cal tubes enclosed in the uptake, their lower ends being immediately e the smoke-box doors.

tallations on Howden's system have hitherto been arranged for a rate mbustion to give at full sea-power an average of from 18 to 22 I.H.P. square foot of fire-grate with fire-bars from 5′ 0″ to 5′ 6″ in length.

is believed that with suitable arrangement of proportions even 30 P. per square foot can be obtained.

r an account of recent uses of exhaust-fans for increasing draught, see r by W. R. Roney, Trans. A. S. M. E., vol. xv.

FUEL ECONOMIZERS.

reen's Fuel Economizer.—Clark gives the following average re- s of comparative trials of three boilers at Wigan used with and without nomizers:

	Without Economizers.	With Economizers.
Coal per square foot of grate per hour......	21.6	21.4
Water at 100° evaporated per hour..........	73.55	79.32
Water at 212° per pound of coal	9.60	10.56

howing that in burning equal quantities of coal per hour the rapidity of poration is increased 9.3% and the efficiency of evaporation 10% by the ition of the economizer.

he average temperatures of the gases and of the feed-water before and r passing the economizer were as follows:

	With 6-ft. grate. Before.	With 6-ft. grate. After.	With 4-ft. grate. Before.	With 4-ft. grate. After.
erage temperature of gases.......	649	340	501	312
erage temperature of feed-water.	47	157	41	137

king averages of the two grates, to raise the temperature of the feed- r 100° the gases were cooled down 250°.

rformance of a Green Economizer with a Smoky Coal. e action of Green's Economizer was tested by M. W. Grosseteste for a d of three weeks. The apparatus consists of four ranges of vertical , 6½ feet high, 3¾ inches in diameter outside, nine pipes in each range, ected at top and bottom by horizontal pipes. The water enters all the from below, and leaves them from above. The system of pipes is en- ed in a brick casing, into which the gaseous products of combustion ntroduced from above, and which they leave from below. The pipes leared of soot externally by automatic scrapers. The capacity for is 24 cubic feet, and the total external heating-surface is 290 squar The apparatus is placed in connection with a boiler having 355 squa surface.

apparatus had been at work for seven weeks continuo[illegible] been cleaned, and had accumulated a ⅛-inch coating ash, when its performance, in the same condition, was observed in week. During the second week it was cleaned twice every day; [illegible] the third week, after having been cleaned on Monday morning, [illegible] worked continuously without further cleaning. A smoke-making [illegible] used. The consumption was maintained sensibly constant from day [illegible]

GREEN'S ECONOMIZER.—RESULTS OF EXPERIMENTS ON ITS EFFICI[ENCY] AFFECTED BY THE STATE OF THE SURFACE. (W. Grosseteste.)

TIME (February and March).	Temperature of Feed-water.			Temperature of [Gas]eous Prod[ucts].	
	Entering Feed-heater.	Leaving Feed-heater.	Difference.	Entering Feed-heater.	Leaving Feed-heater.
	Fahr.	Fahr.	Fahr.	Fahr.	Fahr.
1st Week	73.5°	161.5°	88.0°	849°	261°
2d Week	77.0	230.0	153.0	882	297
3d Week—Monday	73.4	196.0	122.6	831	284
Tuesday	73.4	181.4	108.0	871	309
Wednesday	79.0	178.0	99.0	——	——
Thursday	80.6	170.6	90.0	952	329
Friday	80.6	169.0	88.4	889	338
Saturday	79.0	172.4	93.4	901	351

	1st Week.	2d Week.	3d W[eek].
Coal consumed per hour	214 lbs.	216 lbs.	21[illegible]
Water evaporated from 32° F. per hour	1424	1525	14[illegible]
Water per pound of coal	6.65	7.06	6.7[illegible]

It is apparent that there is a great advantage in cleaning the pipes —the elevation of temperature having been increased by it from 88° [illegible] In the third week, without cleaning, the elevation of temperature re[turned] in three days to the level of the first week; even on the first day [illegible] quickly reduced by as much as half the extent of relapse. By cleani[ng the] pipes daily an increased elevation of temperature of 65° F., was ob[tained] whilst a gain of 6% was effected in the evaporative efficiency.

INCRUSTATION AND CORROSION.

Incrustation and Scale.—Incrustation (as distinguished [from] mere sediments due to dirty water, which are easily blown out, or ga[thered] up, by means of sediment-collectors) is due to the presence of salts [in the] feed-water (carbonates and sulphates of lime and magnesia for th[e most] part), which are precipitated when the water is heated, and form [de]posits upon the boiler-plates. (See Impurities in Water, p. 551, ante.)

Where the quantity of these salts is not very large (12 grains per [gallon,] say) scale preventives may be found effective. The chemical pre[ventives] either form with the salts other salts soluble in hot water; or pre[cipitate] them in the form of soft mud, which does not adhere to the plates, [and may] be washed out from time to time. The selection of the chemical m[ust de]pend upon the composition of the water, and it should be introduce[d regu]larly with the feed.

EXAMPLES.—Sulphate-of-lime scale prevented by carbonate of so[da]; sulphate of soda produced is soluble in water; and the carbonate [of lime] falls down in grains, does not adhere to the plates, and may there[fore be] blown out or gathered into sediment-collectors. The chemical reac[tion]:

Sulphate of lime + Carbonate of soda = Sulphate of soda + Carbonate [of lime]
$CaSO_4$ Na_2CO_3 Na_2SO_4 $CaC[O_3]$

Sodium phosphate will decompose the sulphates of lime and magn[esia]:

Sulphate of lime + Sodium phosphate = Calcium phos. + Sulphate [of soda]
$CaSO_4$ Na_2HPO_4 $CaHPO_4$ $Na[_2SO_4]$

Sul. of magnesia + [illegible]hosphate = Phosphate of magnesia + [illegible]
$MgSO_4$ [illegible]O_4 $MgHPO_4$

ty of salts is large, scale preventives are not of much urce of supply must be sought, or the bad water purified to enter the boilers. The damage done to boilers by un- ormous.

e obtained by collecting rain, or condensing steam by ondensers. The water thus obtained should be mixed ater, or treated with a little alkali, as undiluted, pure ; or, after each periodic cleaning, the bad may be used put a skin upon the plates.

and magnesia may be precipitated either by heating the milk of lime (Porter Clark process) with it, the water

produced by the use of pure water, or by the presence r, caused perhaps in the engine-cylinder by the action of upon the grease, resulting in the production of fatty may be neutralized by the addition of lime.

ediment which may collect in a 100-H.P. steam-boiler, bs. of water per hour, the water containing different y in solution, provided that no water is blown off:

urities per U. S. gallon:	30	40	50	60	70	80	90	100
r 100,000:	51.42	68.56	85.71	102.85	120	137.1	154.3	171.4
in 1 hour, pounds:	1.542	2.056	2.571	3.085	3.6	4.11	4.63	5.11
urs, pounds:	15.42	20.56	25.71	30.85	36.0	41.1	46.3	51.4
ys, pounds:	92.55	123.4	154.3	185.1	216.0	246.8	277.6	308.5

er has 1200 sq. ft. heating-surface, one week's running ff, with water containing 100 grains of solid matter per would make a scale nearly .02 in. thick, if evenly depos- eating-surface, assuming the scale to have a sp. gr. of ı. ft.; $.02 \times 1200 \times 156 \times 1/12 = 312$ lbs.

Compounds.—The Bavarian Steam-boiler Inspection ed as follows:

usual substances in water can be retained in soluble form mud by adding caustic soda or lime. This is especially boilers have small interior spaces.

have a chemical analysis of the water in order to fully d and quantity of the preparation to be used for the

unds for removing boiler-scale should be avoided. (A list ds manufactured and sold by German firms is then given nalyzed by the association.)

arations are either nonsensical or fraudulent, or contain wo substances recommended by the association for re- rally soda, which is colored to conceal its presence, and ated with useless or even injurious matter.

as well as giving the compound some strange, fanciful imply to deceive the boiler owner and conceal from him buying colored soda or similar substances, for which he is nt price.

waukee & St. P. R. R. uses for the prevention of scale in an alkaline compound consisting of 3750 gals. of water, stic soda, and 1600 lbs. of 58% soda-ash. Between Milwau- he water-supply contains from 1 to 4½ lbs. of incrusting ., principally calcium carbonate and sulphate and mag- The amount of compound necessary to prevent the in- o 7 pints per 1000 gals. of water. This is really only one tity needed for chemical combination, but the action of *egenerative. The soda-ash (sodium carbonate) extracts n the carbonates of lime and magnesia and precipitates form. The bicarbonate of soda thus form- id by the heat, and is again changed to etically this action might continue ind* so strong as in the case of the larger tube, so as to avoid any contracting the effective area by deposit from the solution; but th of the solution will be just sufficient to neutralize any acidity of (*Iron Age*, Nov. 2, 1893.)

Use of Zinc.—Zinc is often used in boilers to prevent the action of water on the metal. The action appears to be an elect the iron being one pole of the battery and the zinc being the ot hydrogen goes to the iron shell and escapes as a gas into the ste oxygen goes to the zinc.

On account of this action it is generally believed that zinc w prevent corrosion, and that it cannot be harmful to the boiler Some experiences go to disprove this belief, and in numerous case not only been of no use, but has even been harmful. In one case boiler had been troubled with a deposit of scale consisting chi ganic matter and lime, and zinc was tried as a preventive. The action of the zinc was so obvious that its continued use was adv frequent opening of the boiler and cleaning out of detached scale the old scale should be removed and the boiler become clean. Ei months later the water-supply was changed, it being now obtai another stream supposed to be free from lime and to contain onl matter. Two or three months after its introduction the tubes were found to be coated with an obstinate adhesive scale, and com zinc oxide and the organic matter or sediment of the water us deposit had become so heavy in places as to cause overheating an of the plates over the fire. (*The Locomotive.*)

Effect of Deposit on Flues. (Rankine.)—An external c carbonaceous kind is often deposited from the flame and smoke of naces in the flues and tubes, and if allowed to accumulate seriously the economy of fuel. It is removed from time to time by means of and wire brushes. The accumulation of this crust is the probable the fact that in some steamships the consumption of coal per horse-power per hour goes on gradually increasing until it reaches a half times its original amount, and sometimes more.

Dangerous Steam-boilers discovered by Inspec The Hartford Steam-boiler Inspection and Insurance Co. reports inspectors during 1893 examined 163,328 boilers, inspected 66,69[illegible] both internally and externally, subjected 7861 to hydrostatic pres found 597 unsafe for further use. The whole number of defects was 122,893, of which 12,390 were considered dangerous. A sum given below. (*The Locomotive*, Feb. 1894.)

SUMMARY, BY DEFECTS, FOR THE YEAR 1893.

Nature of Defects.	Whole No.	Dangerous.	Nature of Defects.	Whole No.
Deposit of sediment.....	9,774	548	Leakage around tubes..	21,2[illegible]
Incrustation and scale...	18,369	865	Leakage at seams.....	5,6[illegible]
Internal grooving	1,249	148	Water-gauges defective.	[illegible]
Internal corrosion........	6,252	397	Blow-outs defective......	1,6[illegible]
External corrosion........	8,600	536	Deficiency of water	[illegible]
Def'tive braces and stays	1,966	485	Safety-valves overloaded	[illegible]
Settings defective........	3,094	352	Safety-valves defective..	[illegible]
Furnaces out of shape...	4,575	254	Pressure-gauges def'tive	5,8[illegible]
Fractured plates..........	3,532	640	Boilers without pressure-gauges........	[illegible]
Burned plates.............	2,762	325	Unclassified defects....	[illegible]
Blistered plates..........	3,331	164		
Defective rivets.....	17,415	1,569		
Defective heads.	1,357	350	Total..................	122,8[illegible]

The above-named company publishes annually a classified list explosions, compiled chiefly from newspaper reports, showing t 200 to 300 explosions take place in the United States every year, kil 200 to 300 persons, and injuring from 300 to 450. The lists are not to be complete, and may include only a fraction of the actual explosions.

Steam-boilers as Magazines of Explosive Energ *R. H. Thurston* (Trans. A. S. M. E., vol. vi.), in a paper with *title, presents* calculations showing the stored energy in the *steam of various boilers.* Concerning the plain tubular *form and dimens*[illegible] [illegible]pted as a standard by the Hartford e Co., he says: It is 60 inches in diameter, containing 66 3-inch d is 15 feet long. It has 850 feet of heating and 30 feet of grate is rated at 60 horse-power, but is oftener driven up to 75; weighs nds, and contains nearly its own weight of water, but only 21 of steam when under a pressure of 75 pounds per square inch. below its safe allowance. It stores 52,000,000 foot-pounds of en- which but 4 per cent is in the steam, and this is enough to drive r just about one mile into the air, with an initial velocity of nearly per second.

SAFETY-VALVES.

ulation of Weight, etc., for Lever Safety-valves.

W = weight of ball at end of lever, in pounds;
w = weight of lever itself, in pounds;
V = weight of valve and spindle, in pounds;
L = distance between fulcrum and centre of ball, in inches;
l = " " " " " " valve, in inches;
g = " " " " " " gravity of lever, in in.;
A = area of valve, in square inches;
P = pressure of steam, in lbs. per sq. in., at which valve will open.

$$\text{Then}\quad PA \times l = W \times L + w \times g + V \times l;$$

$$\text{whence}\quad P = \frac{WL + wg + Vl}{Al};$$

$$W = \frac{PAl - wg - Vl}{L};$$

$$L = \frac{PAl - wg - Vl}{W}.$$

MPLE.—Diameter of valve, 4″; distance from fulcrum to centre of ball, o centre of valve, 4″; to centre of gravity of lever, 15½″; weight of and spindle, 3 lbs.; weight of lever, 7 lbs.; required the weight of ball ke the blowing-off pressure 80 lbs. per sq. in.; area of 4″ valve = 12.566 Then

$$= \frac{PAl - wg - Vl}{L} = \frac{80 \times 12.566 \times 4 - 7 \times 15\frac{1}{2} - 3 \times 4}{36} = 108.4 \text{ lbs.}$$

following rules governing the proportions of lever-valves are given by . S. Supervisors. The distance from the fulcrum to the valve-stem in no case be less than the diameter of the valve-opening; the length lever must not be more than ten times the distance from the fulcrum valve-stem; the width of the bearings of the fulcrum must not be han three quarters of an inch; the length of the fulcrum-link must not ss than four inches; the lever and fulcrum-link must be made of ght iron or steel, and the knife-edged fulcrum points and the bearings hese points must be made of steel and hardened; the valve must be d by its spindle, both above and below the ground seat and above the through supports either made of composition (gun-metal) or bushed it; and the spindle must fit loosely in the bearings or supports.

Rules for Area of Safety-valves.

le of U. S. Supervising Inspectors of Steam-vessels (as amended 1894).)

er safety-valves to be attached to marine boilers shall have an area of ess than 1 sq. in. to 2 sq. ft. of the grate surface in the boiler, and the of all such safety-valves shall have an angle of inclination of 45° to the e line of their axes.

ing-loaded safety-valves shall be required to have an area of not less 1 sq. in. to 3 sq. ft. of grate surface of the boiler, except as hereinafter wise provided for *water-tube or coil and sectional boilers, and each g-loaded valve shall be supplied with* a lever that will raise the valve *ts seat a distance of not less than* that equal to one eighth the diam- *the valve-opening, and the seats of all such* safety-valves shall have *e of inclination to the centre* line of their axes of 45°. All spring- *fety-valves for water-tube* or coil and sectional boilers r

If we combine this formula with the formulæ

Flow in lbs. per hour = area of opening in sq. in. × 51.43 × abs. pressure, and

Area = diameter of valve × lift × 2.23, we obtain the following, which the author suggests as probably a more correct formula for the discharging capacity of the ordinary lever safety-valve than either of those above given.

Flow in lbs. per hour $= d(.0603 - .0031d) \times 115 \times 2.23 \times 51.43 = d(795 - 41d)$.

From which we obtain:

Diameter, inches....	1	1½	2	2½	3	3½	4	5	6	7
Flow, lbs. per hour..	754	1100	1426	1733	2016	2282	2524	2950	3294	3556
Horse-power........	25	37	47	58	67	76	84	98	110	119

the horse-power being taken as an evaporation of 30 lbs. of water per hour.

If we solve the example, above given, of the boiler evaporating 3600 lbs. of water per hour by this table, we find it requires one 7-inch valve, or a 2½- and a 3-inch valve combined. The 7-inch valve has an area of 38.5 sq. in., and the two smaller valves taken together have an area of only 12 sq. in.; another evidence of the absurdity of considering the area of disk as the factor which determined the capacity of the valve.

It is customary in practice not to use safety-valves of greater diameter than 4 in. If a greater diameter is called for by the rule that is adopted, then two or more valves are used instead of one.

Spring-loaded Safety-valves.—Instead of weights, springs are sometimes employed to hold down safety-valves. The calculations are similar to those for lever safety-valves, the tension of the spring corresponding to a given rise being first found by experiment (see Springs, page 347).

The rules of the U. S. Supervisors allow an area of 1 sq. in. of the valve to 3 sq. ft. of grate, in the case of spring-loaded valves, except in water-tube, coil, or sectional boilers, in which 1 sq. in. to 6 sq. ft. of grate is allowed.

Spring-loaded safety-valves are usually of the reactionary or "pop" type, in which the escape of the steam is opposed by a lip above the valve-seat, against which the escaping steam reacts, causing the valve to lift higher than the ordinary valve.

A. G. Brown gives the following for the rise, effective area, and quantity of steam discharged per hour by valves of the "pop" or Richardson type. The effective is taken at only 50% of the actual area due to the rise, on account of the obstruction which the lip of the valve offers to the escape of steam.

Dia. valve, in	1	1½	2	2½	3	3½	4	4½	5	6
Lift, inches.	.125	.150	.175	.200	.225	.250	.275	.300	.325	[illegible]
Area, sq. in.	.196	.354	.550	.785	1.061	1.375	1.728	2.121	2.553	[illegible]
Gauge-pres.,	**Steam discharged per hour, lbs.**									
30 lbs.	474	856	1330	1897	2563	3325	4178	5128	6173	[illegible]
50	669	1209	1878	2680	3620	4695	5901	7242	8718	[illegible]
70	861	1556	2417	3450	4660	6144	7596	9324	11220	[illegible]
90	1050	1897	2947	4207	5680	7370	9260	11365	13685	[illegible]
100	1144	2065	3208	4580	6185	8322	10080	12375	14895	[illegible]
120	1332	2405	3736	5332	7202	9342	11735	14410	17340	[illegible]
140	1516	2738	4254	6070	8200	10635	13365	16405	19745	[illegible]
160	1696	3064	4760	6794	9175	11900	14955	18355	22095	[illegible]
180	1883	3400	5283	7540	10180	13250	16595	20370	24520	[illegible]
200	2062	3724	5786	8258	11150	14465	18175	22310	26855	[illegible]

If we take 30 lbs. of steam per hour, at 100 lbs. gauge-pressure = 1 H.P., we have from the above table:

Diameter, inches...	1	1½	2	2½	3	3½	4	4½	5	6
Horse-power.......	38	69	107	153	206	277	336	412	496	687

A safety-valve should be capable of discharging a much greater quantity of steam than that corresponding to the rated horse-power of a boiler, since a boiler having ample grate surface and strong draught may generate more than double the quantity of steam its rating calls for.

The Consolidated Safety-valve Co.'s circular gives the following rated capacity of its nickel-seat "pop" safety-valves:

Size, in	1	1¼	1½	2	2½	3	3½	4	4½	5	5½
Boiler H.P. from	8	10	20	35	60	75	100	125	150	175	200
Boiler H.P. to	10	15	30	50	75	100	125	150	175	200	275

The figures in the lower line from 2 inch to 5 inch inclusive, correspond to the formula H.P. = 50(diameter − 1 inch).

rule by Rankine is 1/150 to 1/180 of the number of pounds of water ated per hour, equals for the above case 27 to 20 sq. in. A communi- *Power*, July, 1890, gives two other rules:

l sq. in. disk area for 3 sq. ft. grate, which would give 13.3 sq. in.

¼ sq. in. disk area for 1 sq. ft. grate, which would give 30 sq. in.; but grate-surface were reduced to 30 sq. ft. on account of increased it, these rules would make the disk area only 10 and 22.5 sq. in., tively.

Philadelphia rule for 100 lbs. gauge pressure gives a disk area of 0.21 for each sq. ft. of grate area, which would give an area of 8.4 sq. in. sq. ft. grate, and only 6.3 sq. in. if the grate is reduced to 30 sq. ft. ording to the rule this aggregate area would have to be divided between alves. But if the boiler was driven by forced draught, then the in- or "must estimate the area of grate at 1 sq. ft. for each 16 lbs. of fuel d per hour."

ler this condition the actual grate-surface might be cut down to 400 ÷ 25 sq. ft., and by the rule the combined area of the two safety-valves l be only $25 \times 0.21 = 5.25$ sq. in.

strom's Pocket-book, edition of 1891, gives ¾ sq. in. for 1 sq. ft. grate; quoting from Weisbach, vol. ii, 1/3000 of the heating-surface. This in ase considered is $1200/3000 = .4$ sq. ft. or 57.6 sq. in.

thus have rules which give for the area of safety-valve of the same 100- -power boiler results ranging all the way from 5.25 to 57.6 sq. in.

of the rules above quoted give the area of the disk of the valve as the to be ascertained, and it is this area which is supposed to bear some t ratio to the grate-surface, to the heating-surface, to the water evap- d, etc. It is difficult to see why this area has been considered even oximately proportional to these quantities, for with small lifts the area tual opening bears a direct ratio, not to the area of disk, but to the mference.

as for various diameters of valve:

eter	1	2	3	4	5	6	7
	.785	3.14	7.07	12.57	19.64	28.27	38.48
mference	3.14	6.28	9.42	12.57	15.71	18.85	21.99
m. × lift of 0.1 in.	.31	.63	.94	1.26	1.57	1.89	2.20
to area	.4	.2	.13	.1	.08	.067	.057

e apertures, therefore, are therefore directly proportional to the diam- or to the circumference, but their relation to the area is a varying one. the lift = ¼ diameter, then the opening would be equal to the area of disk, for circumference × ¼ diameter = area, but such a lift is far nd the actual lift of an ordinary safety-valve.

correct rule for size of safety-valves should make the product of the eter and the lift proportional to the weight of steam to be discharged.

"logical" method for calculating the size of safety-valve is given in *Locomotive*, July, 1892, based on the assumption that the actual opening ld be sufficient to discharge all the steam generated by the boiler. er's rule for flow of steam is taken, viz., flow through aperture of one n. in lbs. per second = absolute pressure ÷ 70, or in lbs. per hour = 51.43 solute pressure.

the angle of the seat is 45°, as specified in the rules of the U. S. Super- s, the area of opening in sq. in. = circumference of the disk × the lift , .71 being the cosine of 45°; or diameter of disk × lift × 2.23.

G. Brown in his book on The Indicator and its Practical Working don, 1894) gives the following as the lift of the ordinary lever safety- for 100 lbs. gauge-pressure:

iam. of valve	2	2½	3	3½	4	4½	5	6	inches.
ise of valve	.0583	.0523	.0507	.0492	.0478	.0462	.0446	.0430	inch.

lift decreases with increase of steam-pressure; thus for a 4-inch valve:

pressure, lbs.	45	65	85	105	115	135	155	175	195	215
e-press., lbs.	30	50	70	90	100	120	140	160	180	200
inch	.1034	.0775	.0620	.0517	.0478	.0413	.0365	.0327	.0296	.02[illegible]

effective area of opening Mr. Brown takes at 70% of the rise multiplie e circumference.

approximate formula corresponding to Mr. Brown's figures for dian between 2½ and 6 in. and gauge-pressures between 70 and 200 lbs. is

$$= (.0603 - .0031d) \times \frac{115}{abs.\ pressure},$$ in which d = diam.

If we combine this formula with the formulæ

Flow in lbs. per hour = area of opening in sq. in. × 51.43 × abs. press

Area = diameter of valve × lift × 2.23, we obtain the following, wh author suggests as probably a more correct formula for the disc capacity of the ordinary lever safety-valve than either of those abov

Flow in lbs. per hour $= d(.0603 - .0031d) \times 115 \times 2.23 \times 51.43 = d(79$

From which we obtain:

Diameter, inches....	1	1½	2	2½	3	3½	4	5
Flow, lbs. per hour..	754	1100	1426	1733	2016	2282	2524	2950
Horse-power.........	25	37	47	58	67	76	84	98

the horse-power being taken as an evaporation of 30 lbs. of water pe

If we solve the example, above given, of the boiler evaporating 300 water per hour by this table, we find it requires one 7-inch valve, or and a 3-inch valve combined. The 7-inch valve has an area of 38.5 and the two smaller valves taken together have an area of only 12 another evidence of the absurdity of considering the area of disk factor which determined the capacity of the valve.

It is customary in practice not to use safety-valves of greater di than 4 in. If a greater diameter is called for by the rule that is a then two or more valves are used instead of one.

Spring-loaded Safety-valves.—Instead of weights, sprin sometimes employed to hold down safety-valves. The calculatio similar to those for lever safety-valves, the tension of the spring corre ing to a given rise being first found by experiment (see Springs, page

The rules of the U. S. Supervisors allow an area of 1 sq. in. of the to 3 sq. ft. of grate, in the case of spring-loaded valves, except in wate coil, or sectional boilers, in which 1 sq. in. to 6 sq. ft. of grate is allow

Spring-loaded safety-valves are usually of the reactionary or "pop in which the escape of the steam is opposed by a lip above the valv against which the escaping steam reacts, causing the valve to lift than the ordinary valve.

A. G. Brown gives the following for the rise, effective area, and q of steam discharged per hour by valves of the "pop" or Richardso The effective is taken at only 50% of the actual area due to the rise, on a of the obstruction which the lip of the valve offers to the escape of st

Dia. value, in	1	1½	2	2½	3	3½	4	4½	5
Lift, inches.	.125	.150	.175	.200	.225	.250	.275	.300	[illegible]
Area, sq. in.	.196	.354	.550	.785	1.061	1.375	1.728	2.121	[illegible]
Gauge-pres.,	Steam discharged per hour, lbs.								
30 lbs.	474	856	1330	1897	2563	3325	4178	5128	[illegible]
50	669	1209	1878	2680	3620	4695	5901	7242	[illegible]
70	861	1556	2417	3450	4660	6144	7596	9324	[illegible]
90	1050	1897	2947	4207	5680	7370	9260	11365	[illegible]
100	1144	2065	3208	4580	6185	8322	10080	12375	[illegible]
120	1332	2405	3736	5332	7202	9342	11735	14410	[illegible]
140	1516	2738	4254	6070	8200	10635	13365	16405	[illegible]
160	1696	3064	4760	6794	9175	11900	14955	18355	[illegible]
180	1883	3400	5283	7540	10180	13250	16595	20370	[illegible]
200	2062	3724	5786	8258	11150	14165	18175	22310	[illegible]

If we take 30 lbs. of steam per hour, at 100 lbs. gauge-pressure = we have from the above table:

Diameter, inches...	1	1½	2	2½	3	3½	4	4½	5
Horse-power.......	38	69	107	153	206	277	336	412	496

A safety-valve should be capable of discharging a much greater of steam than that corresponding to the rated horse-power of a boil a boiler having ample grate surface and strong draught may genera than double the quantity of steam its rating calls for.

The Consolidated Safety-valve Co.'s circular gives the followi *capacity of its nickel-seat "pop" safety-valves:*

Size, in	1	1¼	1½	2	2½	3	3½	4	4½
Boiler H.P. from	8	10	20	35	60	75	100	125	150
to	10	15	30	50	75	100	125	150	175

The figures ...er line from 2 inch to 5 inch, inclusive, c *he formula* ...diameter − 1 inch).

THE INJECTOR.

Equation of the Injector.

the number of pounds of steam used;
number of pounds of water lifted and forced into the boiler;
height in feet of a column of water, equivalent to the absolute
ressure in the boiler;
height in feet the water is lifted to the injector;
temperature of the water before it enters the injector;
temperature of the water after leaving the injector;
total heat above 32° F. in one pound of steam in the boiler, in
eat-units;
lost work in friction and the equivalent lost work due to radia-
ion and lost heat;
e mechanical equivalent of heat.

$$[H - (t_2 - 32°)] = W(t_2 - t_1) + \frac{(W + S)h + Wh_0 + L}{778}.$$

ralent formula, neglecting $Wh_0 + L$ as small, is

$$S = \left[W(t_2 - t_1) + \frac{W + S}{d} \cdot p \cdot \frac{144}{778} \right] \frac{1}{H - (t_2 - 32°)},$$

$$\text{or} \quad S = \frac{W[(t_2 - t_1)d + .1851p]}{H - (t_2 - 32°)d - .1851p},$$

d = weight of 1 cu. ft. of water at temperature t_2; p = absolute
f steam, lbs. per sq. in.
e for finding the proper sectional area for the narrowest part of
s is given as follows by Rankine, S. E. p. 477:

$$\text{in square inches} = \frac{\text{cubic feet per hour gross feed-water}}{800 \sqrt{\text{pressure in atmospheres}}}.$$

rtant condition which must be fulfilled in order that the injector
is that the supply of water must be sufficient to condense the
s the temperature of the supply or feed-water is higher, the
f water required for condensing purposes will be greater.
le below gives the calculated value of the maximum ratio of water
am, and the values obtained on actual trial, also the highest admis-
perature of the feed-water as shown by theory and the highest
ound by trial with several injectors.

Maximum Ratio Water to Steam.				Gauge-pressure, pounds per sq. in.	Maximum Temperature of Feed-Water.					
Calculated from Theory.	Actual Experiment.				Theoretical.		Experi'tal Results.			
	H.	P.	M.		Temp. discharge 180°.	Temp. discharge 212°.	H.	P.	M.	S.
36.5	30.9	....	....	10	....	....	....	....	....	132°
25.6	22.5	19.9	21.5	20	142°	173°	135°	120°	130°	134
20.9	19.0	17.2	19.0	30	132	162	....	....	....	134
17.87	15.8	15.0	15.86	40	126	156	140	113	125	132
16.2	13.3	14.0	13.3	50	120	150	....	....	....	131
14.7	11.2	11.2	12.6	60	114	143	....	115	123	130
13.7	12.3	11.7	12.9	70	109	139	141*	....	123	130
12.9	11.4	11.2	....	80	105	134	141*	118	122	131
12.1	....	....	....	90	99	129	....	....	....	132*
11.5	....	....	....	100	95	125	....	....	....	132*
				120	87	117	....	....	....	134*
				150	77	107	....	....	....	121*

nperature of delivery above 212°. Waste-valve close

k Inspirator; P, Park injector; M, Metropolitan in
tor.

STEAM SEPARATORS.

If moist steam flowing at a high velocity in a pipe has its direc denly changed, the particles of water are by their momentum pro their original direction against the bend in the pipe or wall of the n which the change of direction takes place. By making proper p for drawing off the water thus separated the steam may be dr greater or less extent.

For long steam-pipes a large drum should be provided near the for trapping the water condensed in the pipe. A drum 3 feet dian feet high, has given good results in separating the water of conden a steam-pipe 10 inches diameter and 800 feet long.

Efficiency of Steam Separators.—Prof. R. C. Carpenter made a series of tests of six steam separators, furnishing them wit containing different percentages of moisture, and testing the qu steam before entering and after passing the separator. A condens of the principal results is given below.

Make of Separator.	Test with Steam of about 10% of Moisture.			Tests with Varying Mo	
	Quality of Steam before.	Quality of Steam after.	Efficiency per cent.	Quality of Steam before.	Quality of Steam after.
B	87.0%	98.8%	90.8	66.1 to 97.5%	97.8 to 99%
A	90.1	98.0	80.0	51.9 " 98	97.9 " 99.1
D	89.6	95.8	59.6	72.2 " 96.1	95.5 " 98.2
C	90.6	93.7	33.0	67.1 " 96.8	93.7 " 98.4
E	88.4	90.2	15.5	68.6 " 98.1	79.3 " 98.5
F	88.9	92.1	28.8	70.4 " 97.7	84.1 " 97.9

Conclusions from the tests were: 1. That no relation existed betw volume of the several separators and their efficiency.

2. No marked decrease in pressure was shown by any of the sepa the most being 1.7 lbs. in E.

3. Although changed direction, reduced velocity, and perhaps cent force are necessary for good separation, still some means must be pr to lead the water out of the current of the steam.

The high efficiency obtained from B and A was largely due to this f In B the interior surfaces are corrugated and thus catch the water out of the steam and readily lead it to the bottom.

In A, as soon as the water falls or is precipitated from the steam, it in contact with the perforated diaphragm through which it runs in space below, where it is not subjected to the action of the steam.

In D, the next in efficiency, this is accomplished by means of a > diaphragm which throws the water back into the corners out of the of steam.

DETERMINATION OF THE MOISTURE IN STE STEAM CALORIMETERS.

In all boiler-tests it is important to ascertain the quality of the i.e., 1st, whether the steam is "saturated" or contains the q of heat due to the pressure according to standard experiments; 2d, w the quantity of heat is deficient, so that the steam is wet; and 3d, w the heat is in excess and the steam superheated. The best method o taining the quality of the steam is undoubtedly that employed by mittee which tested the boilers at the American Institute Exhibi 1871-2, of which Prof. Thurston was chairman, i.e., condensing all the evaporated by the boiler by means of a surface condenser, weigh condensing water, and taking its temperature as it enters and as it the condenser; but this plan cannot always be adopted.

A substitute for this method is the barrel calorimeter, which with operation and fairly accurate instruments may generally be relied give results within two per cent of accuracy (that is, a sample of which gives the apparent result of 2% of moisture may contain any tween 0 and 4%). This calorimeter is described as follows: A samp steam is taken by inserting a perforated ½-inch pipe into and main pipe near [illegible] and led by a hose, thoroughly felted, siding prefe[illegible] of water, which is set upon a platfo ith a cock or valve for allowing the water to flow to waste, and ll propeller for stirring the water.

te the calorimeter the barrel is filled with water, the weight and re ascertained, steam blown through the hose outside the barrel ipe is thoroughly warmed, when the hose is suddenly thrust into and the propeller operated until the temperature of the water is to the desired point, say about 110° usually. The hose is then quickly, the temperature noted, and the weight again taken.

of 1/10 of a pound in weighing the condensed steam, or an error ee in the temperature, will cause an error of over 1% in the calcu- entage of moisture. See Trans. A. S. M. E., vi. 293.

ulation of the percentage of moisture is made as below:

$$Q = \frac{1}{H - T}\left[\frac{W}{w}(h_1 - h) - (T - h_1)\right].$$

lity of the steam, dry saturated steam being unity.

al heat of 1 lb. of steam at the observed pressure.

" " " " water at the temperature of steam of the observed pressure.

" " " " condensing water, original.

" " " " " " final.

ight of condensing water, corrected for water-equivalent of the pparatus.

ight of the steam condensed.

age of moisture = $1 - Q$.

reater than unity, the steam is superheated, and the degrees of ting = $2.0833\,(H - T)\,(Q - 1)$.

ulty of Obtaining a Correct Sample.—Recent experiments D. S. Jacobus, Trans. A. S. M. E., xvi. 1017, show that it is practi- possible to obtain a true average sample of the steam flowing in a or accurate determinations all the steam made by the boiler should d through a separator, the water separated should be weighed, and neter test made of the steam just after it has passed the separator.

Calorimeters.—Instead of the open barrel in which the steam nsed, a coil acting as a surface-condenser may be used, which is n the barrel, the water in coil and barrel being weighed separately. ription of an apparatus of this kind designed by the author, which ound to give results with a probable error not exceeding ½ per cent ure, see Trans. A. S. M. E., vi. 294. This calorimeter may be used ously, if desired, instead of intermittently. In this case a continu- of condensing water into and out of the barrel must be established, temperature of inflow and outflow and of the condensed steam short intervals of time.

ttling Calorimeter.—For percentages of moisture not ex- 3 per cent the throttling calorimeter is most useful and convenient narkably accurate. In this instrument the steam which reaches it nch pipe is throttled by an orifice 1/16 inch diameter, opening into a r which has an outlet to the atmosphere. The steam in this cham- its pressure reduced nearly or quite to the pressure of the atmos- but the total heat in the steam before throttling causes the steam in mber to be superheated more or less according to whether the efore throttling was dry or contained moisture. The only observa- quired are those of the temperature and pressure of the steam on le of the orifice.

uthor's formula for reducing the observations of the throttling eter is as follows (Experiments on Throttling Calorimeters, *Am.* Aug. 4, 1892): $w = 100 \times \frac{H - h - K(T - t)}{L}$, in which w = percent- moisture in the steam; H = total heat, and L = latent heat of steam ain pipe; h = total heat due the pressure in the discharge side of rimeter, = 1146.6 at atmospheric pressure; K = specific heat of su ed steam; T = temperature of the throttled and superheated steam calorimeter; t = temperature due the pressure in the calorimeter *atmospheric pressure.*

K at 0.48 and the pressure in the discharge side of the calorimeter *heric pressure, the formula becomes*

$$w = 100 \times \frac{H - 1146.6 - 0.48(T - 212°)}{L}.$$

ormula the following table is calculated :

MOISTURE IN STEAM—DETERMINATIONS BY THROTTLING CALORIMET

Degree of Super-heating $T - 212°$.	Gauge-pressures.										
	5	10	20	30	40	50	60	70	75	80	85
	Per Cent of Moisture in Steam.										
0°	0.51	0.90	1.54	2.06	2.50	2.90	3.24	3.56	3.71	3.86	3.9
10°	0.01	0.39	1.02	1.54	1.97	2.36	2.71	3.02	3.17	3.32	3.4
20°	.....	.....	.51	1.02	1.45	1.83	2.17	2.48	2.63	2.77	2.9
30°	.....	.....	.00	.50	.92	1.30	1.64	1.94	2.09	2.23	2.3
40°	.....	.....	.....	.....	.39	.77	1.10	1.40	1.55	1.69	1.8
50°	.....	.....	.....	.....	.....	.24	.57	.87	1.01	1.15	1.2
60°	.....	.....	.....	.....	.....	.....	.03	.33	.47	.60	.7
70°	.....	.....	.....	.....	.....	.....	.....	.....	.....	.06	.1
Dif. p. deg	.0503	.0507	.0515	.0521	.0526	.0531	.0535	.0539	.0541	.0542	.054

Degree of Super-heating $T - 212°$.	Gauge-pressures.										
	100	110	120	130	140	150	160	170	180	190	200
	Per Cent of Moisture in Steam.										
0°	4.39	4.63	4.85	5.08	5.29	5.49	5.68	5.87	6.05	6.22	6.3
10°	3.84	4.08	4.29	4.52	4.73	4.93	5.12	5.30	5.48	5.65	5.8
20°	3.29	3.52	3.74	3.96	4.17	4.37	4.56	4.74	4.91	5.08	5.2
30°	2.74	2.97	3.18	3.41	3.61	3.80	3.99	4.17	4.34	4.51	4.6
40°	2.19	2.42	2.63	2.85	3.05	3.24	3.43	3.61	3.78	3.94	4.1
50°	1.64	1.87	2.08	2.29	2.49	2.68	2.87	3.04	3.21	3.37	3.5
60°	1.09	1.32	1.52	1.74	1.93	2.12	2.30	2.48	2.64	2.80	2.9
70°	.55	.77	.97	1.18	1.38	1.56	1.74	1.91	2.07	2.23	2.3
80°	.00	.22	.42	.63	.82	1.00	1.18	1.34	1.50	1.66	1.8
90°	.....	.....	.....	.07	.26	.44	.61	.78	.94	1.09	1.2
100°	.....	.....	.....	.....	.....	.....	.05	.21	.37	.52	[illegible]
110°	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	[illegible]
Dif. p. deg	.0549	.0551	.0554	.0556	.0559	.0561	.0564	.0566	.0568	.0570	.057

Separating Calorimeters.—For percentages of moisture the range of the throttling calorimeter the separating calorimeter which is simply a steam separator on a small scale. An improved this calorimeter is described by Prof. Carpenter in *Power*, Feb. 189
For fuller information on various kinds of calorimeters, see pa Prof. Peabody, Prof. Carpenter, and Mr. Barrus in Trans. A. S. M. x, xi, xii, 1889 to 1891; Appendix to Report of Com. on Boile A. S. M. E., vol. vi, 1884; Circular of Schaeffer & Budenberg, N. Y. rimeters, Throttling and Separating," 1894.

Identification of Dry Steam by Appearance of a Prof. Denton (Trans. A. S. M. E., vol. x.) found that jets of steam mistakable change of appearance to the eye when steam varies less from the condition of saturation either in the direction of wetness o heating.

If a jet of steam flow from a boiler into the atmosphere under circum ch that very little loss of heat occurs through radiation, etc., an transparent close to the orifice, or be even a grayish-white c am may be assumed to be so nearly dry that no portable cor orimeter will be capable of measuring the amount of water in th be strongly white, the amount of water may be roughly , but beyond this a calorimeter only can determine oisture.

ommon brass pet-cock may be used as an orifice, but it should, if possi-e set into the steam-drum of the boiler and never be placed further from the latter than 4 feet, and then only when the intermediate reser-r pipe is well covered.

ual Amount of Moisture in Steam Escaping from a ler.—In the common forms of horizontal tubular land boilers and -tube boilers with ample horizontal drums, and supplied with water from substances likely to cause foaming, the moisture in the steam not generally exceed 2% unless the boiler is overdriven or the water- is carried too high.

CHIMNEYS.

imney Draught Theory.—The commonly accepted theory of ney draught, based on Peclet's and Rankine's hypotheses (see Rankine,), is discussed by Prof. De Volson Wood in Trans. A. S. M. E., vol. xi. clet represented the law of draught by the formula

$$h = \frac{u^2}{2g}\left(1 + G + \frac{fl}{m}\right),$$

ich h is the "head," defined as such a height of hot gases as, if added to the column of gases in the chimney, would produce the same pressure at the furnace as a column of outside air, of the same area of base, and a height equal to that of the chimney;

- u is the required velocity of gases in the chimney;
- G a constant to represent the resistance to the passage of air through the coal;
- l the length of the flues and chimney;
- m the mean hydraulic depth or the area of a cross-section divided by the perimeter;
- f a constant depending upon the nature of the surfaces over which the gases pass, whether smooth, or sooty and rough.

nkine's formula (Steam Engine, p. 288), derived by giving certain values e constants (so-called) in Peclet's formula, is

$$h = \frac{\frac{\tau_0}{\tau_2}(0.0807)}{\frac{\tau_0}{\tau_1}(0.084)}H - H = \left(0.96\frac{\tau_1}{\tau_2} - 1\right)H;$$

ich H = the height of the chimney in feet;
τ_0 = 493° F., absolute (temperature of melting ice);
τ_1 = absolute temperature of the gases in the chimney;
τ_2 = absolute temperature of the external air.

f. Wood derives from this a still more complex formula which gives eight of chimney required for burning a given quantity of coal per d, and from it he calculates the following table, showing the height of ney required to burn respectively 24, 20, and 16 lbs. of coal per square of grate per hour, for the several temperatures of the chimney gases .

ide Air. τ_2	Chimney Gas.		Coal per sq. ft. of grate per hour, lbs.		
	τ_1 Absolute.	Temp. Fahr.	24	20	16
			Height H, feet.		
520° lute or ° F.	700	239	250.9	157.6	67.8
	800	339	172.4	115.8	55.7
	1000	539	149.1	100.0	48.7
	1100	639	148.8	98.9	48.2
	1200	739	152.0	100.9	49.1
	1400	939	159.9	105.7	51.2
	1600	1139	168.8	111.0	[illegible]
	2000	1539	206.5	132.2	[illegible]

Rankine's formula gives a maximum draught when $\tau = 2\ 1/12\tau_0$, or 6 when the outside temperature is 60°. Prof. Wood says: "This result i a fixed value, but departures from theory in practice do not affect the r largely. There is, then, in a properly constructed chimney, properly ing, a temperature giving a maximum draught,* and that temperature far from the value given by Rankine, although in special cases it may or 75° more or less."

All attempts to base a practical formula for chimneys upon the the ical formula of Peclet and Rankine have failed on account of the in sibility of assigning correct values to the so-called "constants" G a (See Trans. A. S. M. E., xi. 984.)

Force or Intensity of Draught.—The force of the draught is to the difference between the weight of the column of hot gases inside chimney and the weight of a column of the external air of the same h It is measured by a draught-gauge, usually a U-tube partly filled with one leg connected by a pipe to the interior of the flue, and the other op the external air.

If D is the density of the air outside, d the density of the hot gas i in lbs. per cubic foot, h the height of the chimney in feet, and .192 the f for converting pressure in lbs. per sq. ft. into inches of water column, the formula for the force of draught expressed in inches of water is,

$$F = .192h(D - d).$$

The density varies with the absolute temperature (see Rankine).

$$d = \frac{\tau_0}{\tau_1} 0.084; \quad D = 0.0807 \frac{\tau_0}{\tau_2},$$

where τ_0 is the absolute temperature at 32° F., = 493., τ_1 the absolute perature of the chimney gases and τ_2 that of the external air. Substi these values the formula for force of draught becomes

$$F = .192h\left(\frac{39.79}{\tau_2} - \frac{41.41}{\tau_1}\right) = h\left(\frac{7.64}{\tau_2} - \frac{7.95}{\tau_1}\right).$$

To find the maximum intensity of draught for any given chimney heated column being 600° F., and the external air 60°, multiply the h above grate in feet by .0073, and the product is the draught in inches of w

Height of Water Column Due to Unbalanced Pressur Chimney 100 Feet High. (*The Locomotive*, 1884.)

Temp. in the Chimney.	Temperature of the External Air—Barometer, 14.7 lbs. per sq									
	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°
200	.453	.419	.384	.353	.321	.292	.263	.234	.209	.182
220	.488	.453	.419	.388	.355	.326	.298	.269	.244	.217
240	.520	.488	.451	.421	.388	.359	.330	.301	.276	.250
260	.555	.528	.484	.453	.420	.392	.363	.334	.309	.284
280	.584	.549	.515	.482	.451	.422	.394	.365	.340	.313
300	.611	.576	.541	.511	.478	.449	.420	.392	.367	.340
320	.637	.603	.568	.538	.505	.476	.447	.419	.394	.367
340	.662	.638	.593	.563	.530	.501	.472	.443	.419	[illegible]
360	.687	.653	.618	.588	.555	.526	.497	.468	.444	.417
380	.710	.676	.641	.611	.578	.549	.520	.492	.467	.440
400	.732	.697	.662	.632	.598	.570	.541	.513	.488	.461
420	.753	.718	.684	.653	.620	.591	.563	.534	.509	.482
440	.774	.739	.705	.674	.641	.612	.584	.555	.530	.503
460	.793	.758	.724	.694	.660	.632	.603	.574	.549	.522
480	.810	.776	.741	.710	.678	.649	.620	.591	.566	.542
500	.829	.791	.760	.730	.697	.669	.639	.610	.586	.558

* Much confusion to students of the theory of chimneys has resulted their understanding the words maximum draught to mean maximum sity or pressure of draught, as measured by a draught-gauge. It here maximum quantity or weight of gases passed up the chimney. The mum intensity is found only with maximum temperature, but w temperature reaches about 622° F. the density of the gas decreas rapidly than its velocity increases, so that the weight is a constant shown by Rankine.—W. K.

any other height of chimney than 100 ft. the height of water-column nd by simple proportion, the height of water column being directly rtioned to the height of chimney.

calculations have been made for a chimney 100 ft. high, with various ratures outside and inside of the flue, and on the supposition that the rature of the chimney is uniform from top to bottom. This is the on which all calculations respecting the draught-power of chimneys been made by Rankine and other writers, but it is very far from the in most cases. The difference will be shown by comparing the read- the draught-gauge with the table given. In one case a chimney 122 ft. howed a temperature at the base of 320°, and at the top of 230°. , in his "Treatise on Heat," gives the following table:

GHT POWERS OF CHIMNEYS, ETC., WITH THE INTERNAL AIR AT 552°, AND IE EXTERNAL AIR AT 62°, AND WITH THE DAMPER NEARLY CLOSED.

Draught Power in ins. of water.	Theoretical Velocity in feet per second.		Height of Chimney in feet.	Draught Power in ins. of water.	Theoretical Velocity in feet per second.	
	Cold Air Entering.	Hot Air at Exit.			Cold Air Entering.	Hot Air at Exit.
.073	17.8	35.6	80	.585	50.6	101.2
.146	25.3	50.6	90	.657	53.7	107.4
.219	31.0	62.0	100	.730	56.5	113.0
.292	35.7	71.4	120	.876	62.0	124.0
.365	40.0	80.0	150	1.095	69.3	138.6
.438	43.8	87.6	175	1.277	74.3	149.6
.511	47.3	94.6	200	1.460	80.0	160.0

te of Combustion Due to Height of Chimney.—ridge's "Heat and Heat Engines" gives the following table showing ights of chimney for producing certain rates of combustion per sq. ection of the chimney. It may be approximately true for anthracite erate and large sizes, but greater heights than are given in the table eeded to secure the given rates of combustion with small sizes of cite, and for bituminous coal smaller heights will suffice if the coal onably free from ash—5% or less.

hts .	Lbs. of Coal Burned per Hour per Sq. Ft. of Section of Chimney.	Lbs. of Coal Burned per Sq. Ft. of Grate, the Ratio of Grate to Section of Chimney being 8 to 1.	Heights in feet.	Lbs. of Coal Burned per Hour per Sq. Ft. of Section of Chimney.	Lbs. of Coal Burned per Sq. Ft. of Grate, the Ratio of Grate to Section of Chimney being 8 to 1.
	60	7.5	70	126	15.8
	68	8.5	75	131	16.4
	76	9.5	80	135	16.9
	84	10.5	85	139	17.4
	93	11.6	90	144	18.0
	99	12.4	95	148	18.5
	105	13.1	100	152	19 0
	111	13.8	105	156	19.5
	116	14.5	110	160	20 0
	121	15.1			

rston's rule for rate of combustion effected by a given height of chim- rans. A. S. M. E., xi. 991) is: Subtract 1 from twice the square root of ight, and the result is the rate of combustion in pounds per square foot *te per hour, for anthracite. Or rate* $= 2\sqrt{h} - 1$, in which h is the *in feet. This rule gives the following:*

=	*50*	*60*	*70*	*80*	*90*	*100*	110	125	150	175	200
=	*13.14*	*14.49*	*15.73*	*16.89*	17.97	19	19.97	21.36	23.49	25.45	27.2

lts agree closely with Trowbridge's table given above. In tice the high rates of combustion for high chimneys given by the formula are not generally obtained, for the reason that with high chimneys there are usually long horizontal flues, serving many boilers, and the friction and the interference of currents from the several boilers are apt to cause the intensity of draught in the branch flues leading to each boiler to be much less than that at the base of the chimney. The draught of each boiler is also usually restricted by a damper and by bends in the gas-passages. In a battery of several boilers connected to a chimney 150 ft. high, the author found a draught of ¾-inch water-column at the boiler nearest the chimney, and only ¼-inch at the boiler farthest away. The first boiler was wasting fuel from too high temperature of the chimney-gases, 900°, having too large a grate-surface for the draught, and the last boiler was working below its rated capacity and with poor economy, on account of insufficient draught.

The effect of changing the length of the flue leading into a chimney 60 ft. high and 2 ft. 9 in. square is given in the following table, from Box on "Heat":

Length of Flue in feet.	Horse-power.	Length of Flue in feet.	Horse-power.
50	107.6	800	56.1
100	100.0	1,000	51.4
200	85.3	1,500	43.3
400	70.8	2,000	38.2
600	62.5	3,000	31.7

The temperature of the gases in this chimney was assumed to be 552° F., and that of the atmosphere 62°.

High Chimneys not Necessary.—Chimneys above 150 ft. in height are very costly, and their increased cost is rarely justified by increased efficiency. In recent practice it has become somewhat common to build two or more smaller chimneys instead of one large one. A notable example is the Spreckels Sugar Refinery in Philadelphia, where three separate chimneys are used for one boiler-plant of 7500 H.P. The three chimneys are said to have cost several thousand dollars less than a single chimney of their combined capacity would have cost. Very tall chimneys have been characterized by one writer as "monuments to the folly of their builders."

Heights of Chimney required for Different Fuels.—The minimum height necessary varies with the fuel, wood requiring the least, then good bituminous coal, and fine sizes of anthracite the greatest. It also varies with the character of the boiler—the smaller and more circuitous the gas-passages the higher the stack required; also with the number of boilers, a single boiler requiring less height than several that discharge into a horizontal flue. No general rule can be given.

SIZE OF CHIMNEYS.

The formula given below, and the table calculated therefrom for chimneys up to 96 in. diameter and 200 ft. high, were first published by the author in 1884 (Trans. A. S. M. E. vi., 81). They have met with much approval since that date by engineers who have used them, and have been frequently published in boiler-makers' catalogues and elsewhere. The table is now extended to cover chimneys up to 12 ft. diameter and 300 ft. high. The sizes corresponding to the given commercial horse-powers are believed to be ample for all cases in which the draught areas through the boiler-flues and connections are sufficient, say not less than 20% greater than the area of the chimney, and in which the draught between the boilers and chimney is not checked by long horizontal passages and right-angled bends.

Note that the figures in the table correspond to a coal consumption of 5 lbs. of coal per horse-power per hour. This liberal allowance is made to cover the contingencies of poor coal being used, and of the boilers being driven beyond their rated capacity. In large plants, with economical boilers and engines, good fuel and other favorable conditions, which will reduce the maximum rate of coal consumption at any one time to less than 5 lbs. per H. P. per hour, the figures in the table may be multiplied by the ratio of 5 to the maximum expected coal consumption per H.P. per hour. Thus, with conditions which make the maximum coal consumption only 2.5 lbs. per hour, the chimney 300 ft. high × 12 ft. diameter should be sufficient for 6155 × 2 = 12,310 horse-power. The formula is based on the following data:

Diam. Inches.	Area A. sq. ft.	Effective Area. $E = A - 0.6\sqrt{A}$. sq. ft.	Height of Chimney. 50 ft.	60 ft.	70 ft.	80 ft.	90 ft.	100 ft.	110 ft.	125 ft.	150 ft.	175 ft.	200 ft.	225 ft.	250 ft.	300 ft.	Equivalent Square Chimney. Side of Square $\sqrt{E} + 4$ inches.
			Commercial Horse-power of Boiler.														
18	1.77	.97	23	25	27	29	......	......	......	......	......	......	......	......	......	......	16
21	2.41	1.47	35	38	41	44	......	......	......	......	......	......	......	......	......	......	19
24	3.14	2.08	49	54	58	62	66	......	......	......	......	......	......	......	......	......	22
27	3.98	2.78	65	72	78	83	88	......	......	......	......	......	......	......	......	......	24
30	4.91	3.58	84	92	100	107	113	119	......	......	......	......	......	......	......	......	27
33	5.94	4.48	......	115	125	133	141	149	156	......	......	......	......	......	......	......	30
36	7.07	5.47	......	141	152	163	173	182	191	204	......	......	......	......	......	......	32
39	8.30	6.57	......	......	183	196	208	219	229	245	......	......	......	......	......	......	35
42	9.62	7.76	......	......	216	231	245	258	271	289	316	......	......	......	......	......	38
48	12.57	10.44	......	......	......	311	330	348	365	389	426	......	......	......	......	......	43
54	15.90	13.51	......	......	......	......	427	449	472	503	551	595	......	......	......	......	48
60	19.64	16.98	......	......	......	......	536	565	593	632	692	748	......	......	......	......	54
66	23.76	20.83	......	......	......	......	......	694	728	776	849	918	981	......	......	......	59
72	28.27	25.08	......	......	......	......	......	835	876	934	1023	1105	1181	1253	......	......	64
78	33.18	29.73	......	......	......	......	......	......	1038	1107	1212	1310	1400	1485	1565	......	70
84	38.48	34.76	......	......	......	......	......	......	1214	1294	1418	1531	1637	1736	1830	2005	75
90	44.18	40.19	......	......	......	......	......	......	......	1496	1639	1770	1893	2008	2116	2318	80
96	50.27	46.01	......	......	......	......	......	......	......	1712	1876	2027	2167	2298	2423	2654	86
102	56.75	52.23	......	......	......	......	......	......	......	1944	2130	2300	2459	2609	2750	3012	91
108	63.62	58.83	......	......	......	......	......	......	......	2090	2399	2592	2771	2939	3098	3393	96
	70.88	65.83	......	......	......	......	......	......	......	......	2685	2900	3100	3288	3466	3797	101
	78.54	73.22	......	......	......	......	......	......	......	......	2986	3226	3448	3657	3855	4223	107
	95.03	89.18	......	......	......	......	......	......	......	......	3637	3929	4200	4455	4696	5144	117
	113.10	106.72	......	......	......	......	......	......	......	......	4352	4701	5026	5331	5618	6155	128

For pounds of coal burned per hour for any given size of chimney, multiply the figures in the table by 5.

1. The draught power of the chimney varies as the square ro[…] height.

2. The retarding of the ascending gases by friction may be cons[…] equivalent to a diminution of the area of the chimney, or to a lini[…] chimney by a layer of gas which has no velocity. The thickne[…] lining is assumed to be 2 inches for all chimneys, or the diminutio[…] equal to the perimeter × 2 inches (neglecting the overlapping of th[…] of the lining). Let D = diameter in feet, A = area, and E = effec[…] in square feet.

$$\text{For square chimneys, } E = D^2 - \frac{8D}{12} = A - \frac{2}{3}\sqrt{A}.$$

$$\text{For round chimeys, } E = \frac{\pi}{4}\left(D^2 - \frac{8D}{12}\right) = A - 0.591\sqrt{A}.$$

For simplifying calculations, the coefficient of $\sqrt{A}$ may be tak[…] for both square and round chimneys, and the formula becomes

$$E = A - 0.6\sqrt{A}.$$

3. The power varies directly as this effective area E.

4. A chimney should be proportioned so as to be capable of giving [...] draught to cause the boiler to develop much more than its rated p[…] case of emergencies, or to cause the combustion of 5 lbs. of fuel p[…] horse-power of boiler per hour.

5. The power of the chimney varying directly as the effective area [...] as the square root of the height, H, the formula for horse-power of b[…] a given size of chimney will take the form H P. = $CE\sqrt{H}$, in whic[…] constant, the average value of which, obtained by plotting the[…] obtained from numerous examples in practice, the author finds to b[…]

The formula for horse-power then is

$$\text{H.P.} = 3.33E\sqrt{H}, \quad \text{or} \quad \text{H.P.} = 3.33(A - .6\sqrt{A})\sqrt{H}.$$

If the horse-power of boiler is given, to find the size of chimney, th[…] being assumed,

$$E = \frac{0.3 \text{ H. P.}}{\sqrt{H}}; = A - 0.6\sqrt{A}.$$

For round chimneys, diameter of chimney = diam. of $E + 4''$.

For square chimneys, side of chimney = $\sqrt{E} + 4''$.

If effective area E is taken in square feet, the diameter in inche[…] $13.54\sqrt{E} + 4''$, and the side of a square chimney in inches is $s = 12\sqrt{}$[…]

If horse-power is given and area assumed, the height $H = \left(\frac{0.3 \text{ H}}{E}\right.$[…]

In proportioning chimneys the height is generally first assumed, [...] consideration to the heights of surrounding buildings or hills nea[…] proposed chimney, the length of horizontal flues, the character of co[…] used, etc., and then the diameter required for the assumed heig[…] horse-power is calculated by the formula or taken from the table.

The Protection of Tall Chimney-shafts from Light[…] —C. Molyneux and J. M. Wood (*Industries*, March 28, 1890) recomm[…] tall chimneys the use of a coronal or heavy band at the top of the c[…] with copper points 1 ft. in height at intervals of 2 ft. throughout the[…] ference. The points should be gilded to prevent oxidation. The m[…] proved form of conductor is a copper tape about ¾ in. by ⅛ in[…] weighing 6 ozs. per ft. If iron is used it should weigh not less than[…] per ft. There must be no insulation, and the copper tape should be f[…] to the chimney with holdfasts of the same material, to prevent[…] action. An allowance for expansion and contraction should be made[…] in. in 40 ft. Slight bends in the tape, not too abrupt, answer the p[…] For an earth terminal a plate of metal at least 3 ft. sq. and 1/16 i[…] should be buried as deep as possible in a damp spot. The plate sho[…] the same metal as the conductor, to which it should be soldered. […] earth terminal is water, and when a deep well or other large bo[…] is at hand, the conductor should be carried down into it. […] [illegible] conductor should be avoided. No bend in it shou[…]

Some Tall Brick Chimneys.

	Height.	Internal Diam.	Outside Diameter.		Capacity by the Author's Formula.	
			Base.	Top.	H. P.	Pounds Coal per hour.
Hütte, Sax.	460	15.7′	33′	16′	13,221	66,105
lasgow.. ..	454	....	32			
sgow........	435	13′ 6″	40		9,795	48,975
low, Bolton, ...	367½	13′ 2″	33′10″		8,245	41,225
Co., Boston	350	11	30	21	5,558	27,790
Co., Newark, ...	335	11	28′ 6″	14	5,435	27,175
s, Low'l, Mass	282′9″	12			5,980	29,900
Mills, Law-	250	10			3,839	19,195
ls, Manches-	250	10			3,839	19,195
E. L. Co., R. I.........	238	14			7,515	37,575
Mills, Law-	214	8			2,248	11,240
Works, Pas-	200	9			2,771	13,855
lyn, Two c'h	150	50″ × 120″		each	1,541	7,705

ABOVE CHIMNEYS.—1. This chimney is situated near right bank of the Mulde, at an elevation of 219 feet above ry works, so that its total height above the sea will be 711¾ are situated on the bank of the river, and the furnace-ed across the river to the chimney on a bridge, through a ength. It is built throughout of brick, and will cost about *d Bldr.*

he fact that it was struck by lightning, and somewhat ecautionary measure a copper extension subsequently was ing its entire height 488 feet.

were built of these great heights to remove deleterious eighborhood, as well as for draught for boilers.

re rests on a solid granite foundation, 55 × 30 feet, and its construction there were used 1,700,000 bricks, 2000 tons rrels of mortar, 1000 loads of sand, 1000 barrels of Portland estimated cost is $40,000. It is arranged for two flues, 9 6 feet, connecting with 40 boilers, which are to be run in four triple-expansion engines of 1350 horse-power each.

iform batter of 2.85 inches to every 10 feet. Designed f 200 H. P. each. It is surmounted by a cast-iron cop-hs six tons, and is composed of thirty-two sections, d together by inside flanges, so as to present a smooth foundation is in concrete, composed of crushed lime-and 3 parts, and Portland cement 1 part. It is 40 feet et deep. Two qualities of brick were used; the outer the first quality North River, and the backing up was of w Jersey brick. Every twenty feet in vertical measurement nches wide and ¾ to ½ inch thick, placed edgewise, was alls about 8 inches from the outer circle. As the chimney base it is double. The *outer wall is 5 feet 2 inches in thick-of this is a second wall 20 inches thick and spaced off about ain wall. From the interior surface of the main wall eight arried, nearly touching this inner or main flue wall in n line should it tend to sag.* The interior wall, starting *described, is gradually reduced* until a height of a *hen it is diminished to 8 inches.* At 165 feet it c

and the rest of the chimney is without lining. The total weight of th
ney and foundation is 5000 tons. It was completed in September, 18
7. Connected to 12 boilers, with 1200 square feet of grate-surface. D
gauge 1 9/16 inches.
8. Connected to 8 boilers, 6′ 8″ diameter × 18 feet. Grate-surfa
square feet.
9. Connected to 64 Manning vertical boilers, total grate surface
Designed to burn 18,000 lbs. anthracite per hour.
10. Designed for 12,000 H.P. of engines; (compound condensing)
11. Grate-surface 434 square feet; H.P. of boilers (Galloway) about
13. Eight boilers (water-tube) each 450 H.P.; 12 engines, each 300 H.P.
designed for 36,000 incandescent lights. For the first 60 feet the
wall is 28 inches thick, then 24 inches for 20 feet, 20 inches for 20
inches for 20 feet, and 12 inches for 20 feet. The interior wall is
thick of fire-brick for 50 feet, and then 8 inches thick of red brick
next 30 feet. Illustrated in *Iron Age*, January 2, 1890.

A number of the above chimneys are illustrated in *Power*, Dec., 18
Chimney at Knoxville, Tenn., illustrated in *Eng'g News*, Nov. 2,
6 feet diameter, 120 feet high, double wall:

Exterior wall,	height	20 feet,	30 feet,	30 feet,	40 feet;
" "	thickness	21½ in.,	17 in.,	13 in.,	8½ in.;
Interior wall,	height	85 ft.,	35 ft.,	29 ft.,	21 ft.;
" "	thickness	13½ in.,	8½ in.,	4 in.,	0.

Exterior diameter, 15′ 6″ at bottom; batter, 7/16 inch in 12 inches fr
tom to 8 feet from top. Interior diameter of inside wall, 6 feet unif
top of interior wall. Space between walls, 16 inches at bottom, dim
to 0 at top of interior wall. The interior wall is of red brick except
of 4 inches of fire-brick for 20 feet from bottom.

Stability of Chimneys.—Chimneys must be designed to re
maximum force of the wind in the locality in which they are bu
Weak Chimneys, below). A general rule for diameter of base, of
chimneys, approved by many years of practice in England and the
States, is to make the diameter of the base one tenth of the height.
chimney is square or rectangular, make the diameter of the inscribed
of the base one tenth of the height. The "batter" or taper of a ch
should be from 1/16 to ¼ inch to the foot on each side. The bri
should be one brick (8 or 9 inches) thick for the first 25 feet from the t
creasing ½ brick (4 or 4½ inches) for each 25 feet from the top down
If the inside diameter exceed 5 feet, the top length should be 1½ brick
if under 3 feet, it may be ½ brick for ten feet.

(From *The Locomotive*, 1884 and 1886.) For chimneys of four feet
eter and one hundred feet high, and upwards, the best form is circul
a straight batter on the outside. A circular chimney of this size, in ad
to being cheaper than any other form, is lighter, stronger, and looks
better and more shapely.

Chimneys of any considerable height are not built up of uniform thi
from top to bottom, nor with a uniformly varying thickness of wall,
wall, heaviest of course at the base, is reduced by a series of steps.

Where practicable the load on a chimney foundation should not exce
tons per square foot in compact sand, gravel, or loam. Where a soli
bottom is available for foundation, the load may be greatly increase
the rock is sloping, all unsound portions should be removed, and th
dressed to a series of horizontal steps, so that there shall be no tende
slide after the structure is finished.

All boiler-chimneys of any considerable size should consist of an
stack of sufficient strength to give stability to the structure, and an
stack or core independent of the outer one. This core is by many e
extended up to a height of but 50 or 60 feet from the base of the ch
but the better practice is to run it up the whole height of the chim
may be stopped off, say, a couple feet below the top, and the outer sh
tracted to the area of the core but the better way is to run it up to
or 12 inches of the top and not contract the outer shell. But under
cumstances should the core at its upper end be built into or connected
the outer stack. This has been done in several instances by brickla
the result has been the expansion of the inner core which lifted t
the outer stack squarely up and cracked the brickwork.

For a height of 100 feet we would make the outer shell in thre
first 30 feet high, 16 inches thick, the second 30 feet high, 12 inch

50 feet high and 8 inches thick. These are the minimum thicknesses ...ible for chimneys of this height, and the batter should be not less 1 in 36 to give stability. The core should also be built in three steps of which may be about one third the height of the chimney, the lowest ...hes, the middle 8 inches, and the upper step 4 inches thick. This will ... a good sound core. The top of a chimney may be protected by a ...ron cap; or perhaps a cheaper and equally good plan is to lay the ...mental part in some good cement, and plaster the top with the same ...rial.

...eak **Chimneys.**—James B. Francis, in a report to the Lawrence Co. in 1873 (*Eng'g News*, Aug. 28, 1880), gives some calculations con...ng the probable effects of wind on that company's chimney as then ...ructed. Its outer shell is octagonal. The inner shell is cylindrical, ...an air-space between it and the outer shell; the two shells not being ...d together, except at the openings at the base, but with projections in ...rickwork, at intervals of about 20 ft. in height, to afford lateral sup... by contact of the two shells. The principal dimensions of the chimney ...s follows:

...t above the surface of the ground........................	211 ft.
...eter of the inscribed circle of the octagon near the ground.	15 "
...eter of the inscribed circle of the octagon near the top...	10 ft. 1½ in.
...ness of the outer shell near the base, 6 bricks, or.........	23½ in.
...ness of the outer shell near the top, 3 bricks, or...........	11½ "
...ness of the inner shell near the base. 4 bricks, or..........	15 "
...ness of the inner shell near the top, 1 brick, or	3¾ "

...e tenth of the height for the diameter of the base is the rule commonly ...ed. The diameter of the inscribed circle of the base of the Lawrence ...facturing Company's chimney being 15 ft., it is evidently much less ...is usual in a chimney of that height.

...n after the chimney was built, and before the mortar had hardened, it ...ound that the top had swayed over about 29 in. toward the east. This ...vidently due to a strong westerly wind which occurred at that time. ...s soon brought back to the perpendicular by sawing into some of the ..., and other means.

...e stability of the chimney to resist the force of the wind depends mainly ...e weight of its outer shell, and the width of its base. The cohesion of ...ortar may add considerably to its strength; but it is too uncertain to ...lied upon. The inner shell will add a little to the stability, but it may ...acked by the heat, and its beneficial effect, if any, is too uncertain to ...ken into account.

...e effect of the joint action of the vertical pressure due to the weight of ...chimney, and the horizontal pressure due to the force of the wind is to ... the centre of pressure at the base of the chimney, from the axis to... one side, the extent of the shifting depending on the relative magni... of the two forces. If the centre of pressure is brought too near the ...of the chimney, it will crush the brickwork on that side, and the chim...will fall. A line drawn through the centre of pressure, perpendicular to ...irection of the wind, must leave an area of brickwork between it and ...ide of the chimney, sufficient to support half the weight of the chim... the other half of the weight being supported by the brickwork on the ...ward side of the line.

...fferent experimenters on the strength of brickwork give very different ...lts. Kirkaldy found the weights which caused several kinds of bricks, ...in hydraulic lime mortar and in Roman and Portland cements, to fail ...lly, to vary from 19 to 60 tons (of 2000 lbs.) per sq. ft. If we take in this ... 25 tons per sq. ft., as the weight that would cause it to begin to fail, we ...l not err greatly. To support half the weight of the outer shell of the ...mney, or 322 tons, at this rate, requires an area of 12.88 sq. ft. of brick...k. From these data and the drawings of the chimney, Mr. Francis cal...tes that the area of 12.88 sq. ft. is contained in a portion of the chimney ...nding 2.428 ft. from one of its octagonal sides, and that the limit to ...ch the centre of pressure may be shifted is therefore 5.072 ft. from the .... If shifted beyond this, he says, on the assumption of the strength ...e brickwork, it will crush and the chimney will fall.

...culating that the wind-pressure can affect only the upper 141 ft. of th... ...ney, the lower 70 ft. being protected by buildings, he calculates th... ...pressure of 44.02 lbs. per sq. ft. would blow the chimney down. ...kine, in a paper printed in the transactions of the Institution ...

neers, in Scotland, for 1867–68, says: "It had previously been asce by observation of the success and failure of actual chimneys, and esp of those which respectively stood and fell during the violent storms that, in order that a round chimney may be sufficiently stable, its should be such that a pressure of wind, of about 55 lbs. per sq. ft. of surface, directly facing the wind, or 27½ lbs. per sq. ft. of the plane tion of a cylindrical surface, . . . shall not cause the resultant p at any bed-joint to deviate from the axis of the chimney by more th quarter of the outside diameter at that joint."

According to Rankine's rule, the Lawrence Mfg. Co.'s chimney is a to a maximum pressure of wind on a plane acting on the whole hei 18.80 lbs. per sq. ft., or of a pressure of 21.70 lbs. per sq. ft. acting uppermost 141 ft. of the chimney.

Steel Chimneys are largely coming into use, especially for tal neys of iron-works, from 150 to 300 feet in height. The advantages c are: greater strength and safety; smaller space required; smaller c 30 to 50 per cent, as compared with brick chimneys; avoidance of i tion of air and consequent checking of the draught, common in brick neys. They are usually made cylindrical in shape, with a wide curve for 10 to 25 feet at the bottom. A heavy cast-iron base-plate is provi which the chimney is riveted, and the plate is secured to a massive f tion by holding-down bolts. No guys are used. F. W. Gordon, of the Engineering Works, gives the following method of calculating their ance to wind pressure (*Power*, Oct. 1893):

In tests by Sir William Fairbairn we find four experiments to dete the strength of thin hollow tubes. In the table will be found their ele with their breaking strain. These tubes were placed upon hollow b and the weights suspended at the centre from a block fitted to the ins the tube.

	Clear Span, ft. in.	Thickness Iron, in.	Outside Diameter, in.	Sectional Area, in.	Breaking Weight, lbs.	Breaking lbs., by C Formu Constan
I.	17	.037	12	1.3901	2,704	[illegible]
II.	15 7½	.113	12.4	4.3669	11,440	[illegible]
III.	23 5	.0631	17.68	3.487	6,400	[illegible]
IV.	23 5	.119	18.18	6.74	14,240	[illegible]

Edwin Clarke has formulated a rule from experiments conducted b during his investigations into the use of iron and steel for hollow bridges, which is as follows:

$$\left.\begin{array}{l}\text{Center break-}\\ \text{ing load, in tons.}\end{array}\right\} = \frac{\text{Area of material in sq.in.} \times \text{Mean depth in in.} \times \text{Con}}{\text{Clear span in feet.}}$$

When the constant used is 1.2, the calculation for the tubes experim upon by Mr. Fairbairn are given in the last column of the table. Clark's "Rules, Tables, and Data," page 513, gives a rule for hollow as follows: $W = 3.14D^2TS \div L$. W = breaking weight in pounds in D = extreme diameter in inches; T = thickness in inches; L = lengt tween supports in inches; S = ultimate tensile strength in pounds per

Taking S, the strength of a square inch of a riveted joint, at 35,00 per. sq. in., this rule figures as follows for the different examples ex mented upon by Mr. Fairbairn: I, 2870; II, 10,190; III, 7700; IV, 13,5

This shows a close approximation to the breaking weight obtain experiments and that derived from Edwin Clarke's and D. K. Clark's We therefore assume that this system of calculation is practically co and that it is eminently safe when a large factor of safety is provide from the fact that a chimney may be standing for many years wi receiving anything like the strain taken as the basis of the calculatio fifty pounds per square foot. Wind pressure at fifty pounds per squar may be assumed to be travelling in a horizontal direction, and be o same velocity from the top to the bottom of the stack. This is the w assumption. If, however, the chimney is round, its effective area w one-half of its diameter plane. We assume that the entire forc ated in the centre of the height of the section of the sideration.

n example a 125-foot iron chimney at Poughkeepsie, N. Y., the eter of which is 90 inches, the effective surface in square feet ie force of the wind may play will therefore be $7\frac{1}{2}$ times 125 which multiplied by 50 gives a total wind force of 23,437 resistance of the chimney to breaking across the top of the ould be 3.14×168^2 (that is, diameter of base) $\times .25 \times 35,000 \div$,486, or 10.6 times the entire force of the wind. We multiply it above the joint in inches, 750, by 4, because the chimney is ixed beam with a load suspended on one end. In calculating alf way up, we have a beam of the same character. It is a a line half way up the chimney, where it is 90 inches in diam- inch thick. Taking the diametrical section above this line, as concentrated in the centre of it, or half way up from the onsideration, its breaking strength is: $3.14 \times 90^2 \times .187 \times 35,000$ 09,220; and the force of the wind to tear it apart through its $7\frac{1}{2} \times 62\frac{1}{2} \times 50 \div 2 = 11,352$, or a little more than one tenth of f the stack.

k & Wilcox Co.'s book "Steam" illustrates a steel chimney f the Maryland Steel Co., Sparrow's Point, Md. It is 225 ft. ve the base, with internal brick lining 13′ 9″ uniform inside e shell is 25 ft. diam. at the base, tapering in a curve to 17 ft. e base, thence tapering almost imperceptibly to 14′ 8″ at the er 40 feet is of ¼-inch plates, the next four sections of 40 ft. ectively 9/32, 5/16, 11/32, and ⅜ inch.

es of Foundations for Steel Chimneys.

lected from circular of Phila. Engineering Works.)

HALF-LINED CHIMNEYS.

r, feet.........	3	4	5	6	7	9	11
....	100	100	150	150	150	150	150
r foundation..	15′9″	16′4″	20′4″	21′10″	22′7″	23′8″	24′8″
oundation......	6′	6′	9′	8′	9′	10′	10′
....	....	125	200	200	250	275	300
r foundation...	....	18′5″	23′8″	25′	29′8″	33′6″	36′
oundation......	....	7′	10′	10′	12′	12′	14′

ght of Sheet-iron Smoke-stacks per Foot.

(Porter Mfg. Co.)

k- s .	Weight per ft.	Diam., inches.	Thick-ness W. G.	Weight per ft.	Diam. inches.	Thick-ness W. G.	Weight per ft.
6	7.20	26	No. 16	17.50	20	No. 14	18.33
	8.66	28	"	18.75	22	"	20.00
	9.58	30	"	20.00	24	"	21.66
	11.68	10	No. 14	9.40	26	"	23.33
	13.75	12	"	11.11	28	"	25.00
	15.00	14	"	13.69	30	"	26.66
	16.25	16	"	15.00			

eet-iron Chimneys. (Columbus Machine Co.)

ngth nney, eet.	Thick-ness Iron, B. W. G.	Weight, lbs.	Diameter Chimney, inches.	Length Chimney, feet.	Thick-ness Iron, B. W. G	Weight, lbs.
20	No. 16	160	30	40	No. 15	960
20	" 16	240	32	40	" 15	1,020
20	" 16	320	34	40	" 14	1,170
[illegible]	" 16	350	36	40	" 14	1,240
	" 16	760	38	40	" 12	1,800
	" 16	826	40	40	" 12	1,890
	" 15	900				

THE STEAM-ENGINE.

Expansion of Steam. Isothermal and Adiabatic. ing to Mariotte's law, the volume of a perfect gas, the temperat kept constant, varies inversely as its pressure, or $p \propto \frac{1}{v}$; $pv = a$ The curve constructed from this formula is called the *isothermal* curve of equal temperatures, and is a common or rectangular h The relation of the pressure and volume of saturated steam, as from Regnault's experiments, and as given in Steam tables, is mately, according to Rankine (S. E., p. 403), for pressures not exce lbs., $p \propto \frac{1}{v^{\frac{17}{16}}}$, or $p \propto v^{-\frac{17}{16}}$, or $pv^{\frac{17}{16}} = pv^{1.0625}$ = a constant. Ze found that the exponent 1.0646 gives a closer approximation.

When steam expands in a closed cylinder, as in an engine, acc Rankine (S. E., p. 385), the approximate law of the expansion is p

$p \propto v^{-\frac{10}{9}}$, or $pv^{1.111}$ = a constant. The curve constructed from mula is called the *adiabatic* curve, or curve of no transmission of

Peabody Therm., p. 112) says: "It is probable that this equ obtained by comparing the expansion lines on a large number of diagrams. . . . There does not appear to be any good reason for exponential equation in this connection, . . . and the action of a lagg engine cylinder is far from being adiabatic. . . . For general pur hyperbola is the best curve for comparison with the expansion cu indicator-card. . . ." Wolff and Denton, Trans. A. S. M. E., ii. "From a number of cards examined from a variety of steam-engin rent use, we find that the actual expansion line varies betwee adiabatic curve and the Mariotte curve."

Prof. Thurston (A. S. M. E., ii. 203), says he doubts if the expo becomes the same in any two engines, or even in the same engin ferent times of the day and under varying conditions of the day.

Expansion of Steam according to Mariotte's La to the Adiabatic Law. (Trans. A. S. M. E., ii. 156.)—Mario $pv = p_1v_1$; values calculated from formula $\frac{P_m}{p_1} = \frac{1}{R}(1 + \text{hyp log } R$ $R = v_2 \div v_1$, p_1 = absolute initial pressure, P_m = absolute mean v_1 = initial volume of steam in cylinder at pressure p_1, v_2 = final steam at final pressure. Adiabatic law: $pv^{\frac{10}{9}} = p_1v_1^{\frac{10}{9}}$; values c from formula $\frac{P_m}{p_1} = 10R^{-1} - 9R^{-\frac{10}{9}}$.

Ratio of Expansion R.	Ratio of Mean to Initial Pressure.		Ratio of Expansion R.	Ratio of Mean to Initial Pressure.		Ratio of Expansion R.	Ratio of Mean to Initial Pressure.
	Mar.	Adiab.		Mar.	Adiab.		Mar.
1.00	1.000	1.000	3.7	.624	.600	6.	.465
1.25	.978	.976	3.8	.614	.590	6.25	.453
1.50	.937	.931	3.9	.605	.580	6.5	.442
1.75	.891	.881	4.	.597	.571	6.75	.431
2.	.847	.834	4.1	.588	.562	7.	.421
2.2	.813	.798	4.2	.580	.554	7.25	.411
2.4	.781	.765	4.3	.572	.546	7.5	.402
2.5	.766	.748	4.4	.564	.538	7.75	[illegible]
2.6	.752	.733	4.5	.556	.530	8.	[illegible]
2.8	.725	.704	4.6	.549	.523	8.25	[illegible]
3.	.700	.678	4.7	.542	.516	8.5	[illegible]
3.1	.688	.666	4.8	.535	.509	8.75	[illegible]
3.2	.676	.654	4.9	.528	.502	9.	[illegible]
3.3	.665	.642	5.05	.522	.495	9.25	[illegible]
3.4	.654	.630	5.2	.506	.479	9.5	[illegible]
3.5	.644	.620	5.5	.492	.464	9.75	[illegible]
[illegible]	.634	.610	5.75	.478	.450	10.	[illegible]

sure of Expanded Steam.—For calculations of
erally assumed that steam expands according to Mariotte's
f the expansion line being a hyperbola. The mean pressure,
vacuum, is then obtained from the formula

$$P_m = p_1 \frac{1 + \text{hyp} \log R}{R},$$

he absolute mean pressure, p_1 the absolute initial pressure
up to the point of cut-off, and R the ratio of expansion. If
oke to the cut-off, L = total stroke,

$$\frac{p_1 l \text{ hyp} \log \frac{L}{l}}{L}; \quad \text{and if } R = \frac{L}{l}, \quad P_m = p_1 \frac{1 + \text{hyp} \log R}{R}.$$

Terminal Absolute Pressures.—**Mariotte's**
lues in the following table are based on Mariotte's law,
s last column, which give the mean pressure of superheated
ccording to Rankine, expands in a cylinder according to
$-\frac{17}{16}$. These latter values are calculated from the formula
$\frac{1}{16}$. $R^{-\frac{1}{16}}$ may be found by extracting the square root of $\frac{1}{R}$
m the mean absolute pressures given deduct the mean back
te) to obtain the mean effective pressure.

Ratio of Mean to Initial Pressure.	Ratio of Mean to Terminal Pressure.	Ratio of Terminal to Mean Pressure.	Ratio of Initial to Mean Pressure.	Ratio of Mean to Initial Dry Steam.
0.1467	4.40	0.227	6.82	0.136
0.1547	4.33	0.231	6.46	....
0.1638	4.26	0.235	6.11	....
0.1741	4.18	0.239	5.75	....
0.1860	4.09	0.244	5.38	....
0.1998	4.30	0.250	5.00	0.186
0.2161	3.89	0.256	4.63	....
0.2358	3.77	0.265	4.24	....
0.2472	3.71	0.269	4.05	....
0.2599	3.64	0.275	3.85	....
0.2690	3.59	0.279	3.72	0.254
0.2742	3.56	0.280	3.65	....
0.2904	3.48	0.287	3.44	....
0.3089	3.40	0.294	3.24	....
0.3303	3.30	0.303	3.03	0.314
0.3552	3.20	0.312	2.81	....
0.3849	3.08	0.321	2.60	0.370
0.4210	2.95	0.339	2.37	....
0.4347	2.90	0.345	2.30	0.417
0.4653	2.79	0.360	2.15	....
0.4807	2.74	0.364	2.08	....
0.5218	2.61	0.383	1.92	0.506
0.5608	2.50	0.400	1.78	....
0.5965	2.39	0.419	1.68	0.582
0.6308	2.29	0.437	1.58	....
0.6615	2.20	0.454	1.51	0.648
0.6995	2.10	0.476	1.43	....
0.7171	2.05	0.488	1.39	0.707
0.7440	1.98	0.505	1.34	....
0.7664	1.91	0.523	1.31	0.756
0.8095	1.80	0.556	1.24	0.800
0.8465	1.69	0.591	1.18	0.840
0.8786	1.60	0.626	1.14	8.874
0.9066	1.51	0.662	1.10	0.900
0.9187	1.47	0.680	1.09	....
0.9292	1.43	0.699	1.07	0.926
0.9405	1.39	0.718	1.06	....

Calculation of Mean Effective Pressure, Clearan Compression Considered.—In the above tables no account of clearance, which in steam-engines modifies t of expansion and the me sure; nor of compress back-pressure, which the mean effective pres the following calculatio elements are considered.

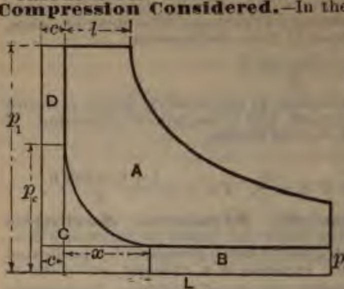


FIG. 137.

L = length of stroke, l before cut-off, x = length pression part of stroke, c ance, p_1 = initial pressu back pressure, p_c = pre clearance steam at end pression. All pressures a lute, that is, measured perfect vacuum.

$$\text{Area of ABCD} = p_1(l+c)\left(1 + \text{hyp log}\,\frac{L+c}{l+c}\right);$$

$$\text{B} = p_b(L-x);$$

$$\text{C} = p_c c\left(1 + \text{hyp log}\,\frac{x+c}{c}\right) = p_b(x+c)\left(1 + \text{hyp log}\right.$$

$$\text{D} = (p_1 - p_c)c = p_1 c - p_b(x+c).$$

$$\text{Area of A} = \text{ABCD} - (\text{B} + \text{C} + \text{D})$$

$$= p_1(l+c)\left(1 + \text{hyp log}\,\frac{L+c}{l+c}\right)$$

$$- \left[p_b(L-x) + p_b(x+c)\left(1 + \text{hyp log}\,\frac{x+c}{c}\right) + p_1 c - p_b(x+\right.$$

$$= p_1(l+c)\left(1 + \text{hyp log}\,\frac{L+c}{l+c}\right)$$

$$- p_b\left[(L-x) + (x+c)\,\text{hyp log}\,\frac{x+c}{c}\right]$$

$$\text{Mean effective pressure} = \frac{\text{area of A}}{L}.$$

EXAMPLE.—Let $L = 1$, $l = 0.25$, $x = 0.25$, $c = 0.1$, $p_1 = 60$ lbs., p_b

$$\text{Area A} = 60(.25 + .1)\left(1 + \text{hyp log}\,\frac{1.1}{.35}\right)$$

$$- 2\left[(1 - .25) + .35\ \text{hyp log}\,\frac{.35}{.1}\right] - 6$$

$$= 21(1 + 1.145) - 2[.75 + 35 \times 1.253] - 6$$

$$= 45.045 - 2.377 - 6 = 36.068 = \text{mean effective pres}$$

The actual indicator-diagram generally shows a mean pressure ably less than that due to the initial pressure and the rate of expans causes of loss of pressure are: 1. Friction in the stop-valves an pipes. 2. Friction or wire-drawing of the steam during admission off, due chiefly to defective valve-gear and contracted steam 3. Liquefaction during expansion. 4. Exhausting before the st completed its stroke. 5. Compression due to early closure of 6. Friction in the exhaust-ports, passages, and pipes.

Re-evaporation during expansion of the steam condensed sion, and valve-leakage after cut-off, tend to elevate the expans the diagram and increase the mean pressure.

If the theoretical mean pressure be calculated from the ini of expansion on the supposition that the expans riotte's law, pv = a constant, and the necessary corrections are clearance and compression, the expected mean pressure in practice found by multiplying the calculated results by the factor in the g table, according to Seaton.

Particulars of Engine.	Factor.
ansive engine, special valve-gear, or with a separate t-off valve, cylinder jacketed	0.94
ansive engine having large ports, etc., and good or-nary valves, cylinders jacketed	0.9 to 0.92
ansive engines with the ordinary valves and gear as general practice, and unjacketed	0.8 to 0.85
pound engines, with expansion valve to h.p. cylin-r; cylinders jacketed, and with large ports, etc	0.9 to 0.92
npound engines, with ordinary slide-valves, cylinders cketed, and good ports, etc	0.8 to 0.85
pound engines as in general practice in the merchant rvice, with early cut-off in both cylinders, without ckets and expansion-valves	0.7 to 0.8
t-running engines of the type and design usually fitted war-ships	0.6 to 0.8

correction be made for clearance and compression, and the engine cordance with general modern practice, the theoretical mean pres-ay be multiplied by 0.96, and the product by the proper factor in the o obtain the expected mean pressure.

the Initial Pressure and the Average Pressure, to nd the Ratio of Expansion and the Period of Admis-n.

P = initial absolute pressure in lbs. per sq. in.;
p = average total pressure during stroke in lbs. per sq. in.;
L = length of stroke in inches;
l = period of admission measured from beginning of stroke;
c = clearance in inches;

$$R = \text{actual ratio of expansion} = \frac{L+c}{l+c} \quad \ldots \ldots \quad (1)$$

$$p = \frac{P(1 + \text{hyp log } R)}{R}.$$

d average pressure p, taking account of clearance,

$$p = \frac{P(l+c) + P(l+c)\text{ hyp log } R - Pc}{L}, \quad \ldots \ldots \quad (2)$$

$$pL + Pc = P(l+c)(1 + \text{hyp log } R);$$

$$\text{hyp log } R = \frac{pL + Pc}{Pl + Pc} - 1 = \frac{\frac{p}{P}L + c}{l+c} - 1. \quad \ldots \quad (3)$$

p and P, to find R and l (by trial and error).—There being two un-quantities R and l, assume one of them, viz., the period of admission itute it in equation (3) and solve for R. Substitute this value of R in nula (1), or $l = \frac{L+c}{R} - c$, obtained from formula (1), and find l. If lt is greated than the assumed value of l, then the assumed value of od of admission is too long; if less, the assumed value is too short. a new value of l, substitute it in formula (3) as before, and continue method of trial and error till the required values of R and l are d.

PLE.—$P = 70$, $p = 42.78$, $L = 60''$, $c = 3''$, to find l. Assume $l = 21$ in.

$$\log R = \frac{\frac{p}{P}L + c}{l+c} - 1 = \frac{\frac{42.78}{70} \times 60 + 3}{21 + 3} - 1 = 1.653 - 1 = .653;$$

$g\ R = .653$, whence $R = 1.92$,

$$l = \frac{L+c}{R} - c = \frac{63}{192} - 3 = 29.8,$$

which is greater than the assumed value, 21 inches.

Now assume $l = 15$ inches:

$$\text{hyp log } R = \frac{\frac{42.78}{70} \times 60 + 3}{15 + 3} - 1 = 1.204, \text{ whence } R = 3.5;$$

$$l = \frac{L+c}{R} - c = \frac{63}{3.5} - 3 = 18 - 3 = 15 \text{ inches, the value assumed.}$$

Therefore $R = 3.5$, and $l = 15$ inches.

Period of Admission Required for a Given Actual Ratio of Expan

$$l = \frac{L+c}{R} - c, \text{ in inches} \quad . \quad . \quad . \quad . \quad . \quad . \quad .$$

$$\text{In percentage of stroke, } l = \frac{100 + \text{p.ct. clearance}}{R} - \text{p. ct. clearance}$$

$$\textit{Terminal pressure} = \frac{P(l+c)}{L+c} = \frac{P}{R}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad .$$

Pressure at any other Point of the Expansion.—Let L_1 = length of st up to the given point.

$$\text{Pressure at the given point} = \frac{P(l+c)}{L_1 + c}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad .$$

WORK OF STEAM IN A SINGLE CYLINDER.

To facilitate calculations of steam expanded in cylinders the table on the next page is abridged from Clark on the Steam-engine. The actual ratios of expansion, column 1, range from 1.0 to 8.0, for which the hyperbolic logarithms are given in column 2. The 3d column contains the periods of admission relative to the actual ratios of expansion, as percentages of the stroke, calculated by formula (5) above. The 4th column gives the ratios of the mean pressures relative to the initial pressures, the latter being taken as 1, calculated by formula (2). In the calculation of columns 3 and 4, clearance is taken into account, and its amount is assumed at 7% of the stroke. The final pressures, in the 5th column, are such as would be arrived at by the continued expansion of the whole of the steam to the end of the stroke, the initial pressure being equal to 1. They are the reciprocals of the ratios of expansion, column 1. The 6th column contains the relative total performances of equal weights of steam worked with the several actual ratios of expansion; the total performance, when steam is admitted for the whole of the stroke, without expansion, being equal to 1. They are obtained by dividing the figures in column 4 by those in column 5.

The pressures have been calculated on the supposition that the pressure of steam, during its admission into the cylinder, is uniform up to the point of cutting off, and that the expansion is continued regularly to the end of the stroke. The relative performances have been calculated without any allowance for the effect of compressive action.

The calculations have been made for periods of admission ranging from 100%, or the whole of the stroke, to 6.4%, or 1/16 of the stroke. And though, nominally, the expansion is 16 times in the last instance, it is actually only 8 times, as given in the first column. The great difference between the nominal and the actual ratios of expansion is caused by the clearance, which is equal to 7% of the stroke, and causes the nominal volume of steam admitted, namely, 6.4%, to be augmented to 6.4 + 7 = 13.4% of the stroke, or say, double, for expansion. When the steam is cut off at 1/8, the actual expansion is only 6 times; when cut off at 1/5, the expansion is 4 times; when cut off at 1/3, the expansion is 2⅔ times; and to effect an actual expansion to twice the initial volume, the steam is cut off at 46½% of the stroke, not at half-stroke.

**Vorking of Steam—Actual Ratios of Expan-
the Relative Periods of Admission, Press-
Performance.**

100 lbs. absolute. Clearance at each end of the cylinder 7%

(SINGLE CYLINDER.)

3	4	5	6	7	8	9
Period of Admission or Cut-off, 7% Clearance.	Average Total Pressure. Initial Pressure = 1.	Total Final Pressure. Initial Pressure = 1.	Ratio of Total Performance of Equal Weights of Steam. (Col. 4 + Col 5.)	Actual Work done by 1 lb. of 100 lbs. Steam. Ft.-lbs.	Quantity of Steam Consumed per H.P. of Actual Work done per hour	Net Capacity of Cylinder per lb. of 100 lbs. Steam admitted in 1 stroke. Cubic feet.
100	1.000	1.000	1.000	58,273	34.0	4.05
90.3	.996	.909	1.096	63,850	31.0	4.45
83.3	.986	.847	1.164	67,836	29.2	4.78
80	.980	.813	1.206	70,246	28.2	4.98
75.3	.969	.769	1.261	73,513	26.9	5.26
70	.953	.719	1.325	77,242	25.6	5.63
66.8	.942	.690	1.365	79,555	24.9	5.87
62.5	.925	.649	1.425	83,055	23.8	6.23
59.9	.913	.625	1.461	85,125	23.3	6.47
54.1	.883	.571	1.546	90,115	22.0	7.08
50	.860	.532	1.616	94,200	21.0	7.61
46.5	.836	.5	1.672	97,432	20.3	8.09
40	.787	.439	1.793	104,466	19.0	9.23
37.6	.766	.417	1.837	107,050	18.5	9.71
33.3	.726	.377	1.925	112,270	17.7	10.72
29.9	.692	.345	2.006	116,88	16.9	11.74
26.4	.652	.313	2.083	121,386	16.3	12.95
25	.637	.298	2.129	124,066	16.0	13.56
22.7	.608	.278	2.187	127,450	15.5	14.57
21.2	.589	.263	2.240	130,533	15.2	15.38
19.7	.569	.250	2.278	132,770	14.9	16.19
18.5	.551	.238	2.315	134,900	14.7	17.00
16.8	.526	.222	2.370	138,130	14.34	18.21
15.3	.503	.208	2.418	140,920	14.05	19.43
14.4	.488	.200	2.440	142,180	13.92	20.23
13.6	.476	.193	2.466	143,720	13.78	21.04
12.5	.457	.182	2.511	146,325	13.53	22.25
11.4	.438	.172	2.547	148,390	13.34	23.47
11.1	.432	.169	2.556	148,940	13.29	23.87
10.3	.419	.161	2.585	150,630	13.14	25.09
10	.413	.159	2.597	151,370	13.08	25.49
9.2	.398	.152	2.619	152,595	12.98	26.71
8.3	.381	.143	2.664	155,200	12.75	28.33
7.7	.369	.137	2.693	156,960	12.61	29.54
7.1	.357	.132	2.711	157,975	12.53	30.76
6.7	.348	.128	2.719	158,414	12.50	31.57
6.4	.342	.125	2.736	159,433	11.83	32.38

OF THE TABLE.—That the initial pressure is uniform; that complete to the end of the stroke; that the pressure in expansion varies inversely as the volume; that there is no back-pressure of compression, and that clearance is 7% of the stroke at each end of the cylinder. *No allowance has been made for loss of steam by cylinder condensation or leakage.*

steam of 100 lbs. pressure per sq. in., or 14,400 .... 4.33 cu. ft.
ressure and volume..62,352 f'

× 26.36 ft. = 55,788 ft.-lbs. The heat equivalent of this work is (55,788 ÷ =) 71.7 units. This is the work of 1 lb. of steam of one atmosphere on a piston without expansion.

The gross work thus done on a piston by 1 lb. of steam generated at pressures varying from 15 lbs. to 100 lbs. per sq. in. varies in round num from 56,000 to 62,000 ft.-lbs., equivalent to from 72 to 80 units of heat.

This work of 1 lb. of steam without expansion is reduced by clear according to the proportion it bears to the net capacity of the cylinder. the clearance be 7% of the stroke, the work of a given weight of steam out expansion, admitted for the whole of the stroke, is reduced in the r of 107 to 100.

Having determined by this ratio the quantity of work of 1 lb. of steam out expansion, as reduced by clearance, the work of the same weight of for various ratios of expansion may be found by multiplying it by the rela performance of equal weights of steam, given in the 6th column of the ta

Quantity of Steam Consumed per Horse-power of Tot Work per Hour. (Column 8 of table.)—The measure of a horse-po is the performance of 33,000 ft.-lbs. per minute, or 1,980,000 ft.-lbs. per h This work, divided by the work of 1 lb of steam, gives the weight of ste required per horse-power per hour. For example, the total actual w done in the cylinder by 1 lb. of 100 lbs. steam, without expansion and 7% of clearance, is 58,273 ft.-lbs.; and $\frac{1,980,000}{58,273} = 34$ lbs. of steam, is the weig of steam consumed for the total work done in the cylinder per horse-pow per hour. For any shorter period of admission with expansion the weig of steam per horse-power is less, as the total work of 1 lb. of steam is m and may be found by dividing 1,980,000 ft.-lbs. by the respective total w done; or by dividing 34 lbs. by the ratio of performance, column 6 in t table.

ACTUAL EXPANSIONS.

With Different Clearances and Cut-offs.

Computed by A. F. Nagle.

Cut-off.	Per Cent of Clearance.									
	0	1	2	3	4	5	6	7	8	9
.01	100.00	50.5	34.0	25.75	20.8	17.5	15.14	13.38	12.00	10.9
.02	50.00	33.67	25.50	20.60	17.53	15.00	13.25	11.89	10.80	9.91
.03	33.33	25.25	20.40	17.16	14.86	13.12	11.78	10.70	9.82	[illegible]
.04	25.00	20.20	17.00	14.71	13.00	11.66	10.60	9.73	9.00	[illegible]
.05	20.00	16.83	14.57	12.87	11.55	10.50	9.64	8.92	8.31	7.79
.06	16.67	14.43	12.75	11.44	10.40	9.55	8.83	8.23	7.71	[illegible]
.07	14.28	12.62	11.33	10.30	9.46	8.75	8.15	7.64	7.20	[illegible]
.08	12.50	11.22	10.2	9.36	8.67	8.08	7.57	7.13	6.75	6.41
.09	11.11	10.10	9.27	8.58	8.00	7.50	7.07	6.69	6.35	[illegible]
.10	10.00	9.18	8.50	7.92	7.43	7.00	6.63	6.30	6.00	[illegible]
.11	9.09	8.42	7.84	7.36	6.93	6.56	6.24	5.94	5.68	5.45
.12	8.33	7.78	7.29	6.86	6.50	6.18	5.89	5.64	5.40	5.19
.14	7.14	6.73	6.37	6.06	5.78	5.53	5.30	5.10	4.91	[illegible]
.16	6.25	5.94	5.67	5.42	5.20	5.00	4.82	4.65	4.50	[illegible]
.20	5.00	4.81	4.64	4.48	4.33	4.20	4.08	3.96	3.86	[illegible]
.25	4.00	3.88	3.77	3.68	3.58	3.50	3.42	3.34	[illegible]	[illegible]
.30	3.33	3.26	3.19	3.12	3.06	3.00	2.94	2.90	2.84	[illegible]
.40	2.50	2.46	2.43	2.40	2.36	2.33	2.30	2.28	2.25	[illegible]
.50	2.00	1.98	1.96	1.94	1.92	1.90	1.89	1.88	1.86	1.85
.60	1.67	1.66	1.65	1.64	1.63	1.615	1.606	[illegible]	[illegible]	[illegible]
.70	1.43	1.42	1.42	1.41	1.41	1.400	[illegible]	[illegible]	[illegible]	[illegible]
.80	1.25	1.25	1.244	1.241	1.238	1.235	[illegible]	[illegible]	[illegible]	[illegible]
.90	1.111	1.11	1.109	1.108	1.106	1.105	1.104	[illegible]	[illegible]	[illegible]
1.00	1.00	1.00	1.000	1.000	1.000	1.000	1.000	[illegible]	[illegible]	[illegible]

ciency of 1 lb. of Steam with and without
k pressure and compression not considered.

$$\text{ressure} = p = \frac{P(l+c) + P(l+c)\text{ hyp. log. } R - Pc}{L}.$$

00; $l = 25$; $c = 7$.

$$\frac{\text{2 hyp. log. } \frac{107}{32} - 7}{100} = \frac{32 + 32 \times 1.209 - 7}{100} = .637.$$

be added to the stroke, so that clearance becomes zero, of steam being used, admission l being then $= l + c =$ $c = 107$.

$$\frac{+ 32 \text{ hyp. log.} \frac{107}{32} - 0}{107} = \frac{32 + 32 \times 1.209}{107} = .707.$$

arance be reduced to 0, the amount of the clearance 7 th the admission and the stroke, the same quantity of e work than when the clearance is 7 in the ratio 707 : 637,

ure Considered.—If back pressure $= .10$ of P, this subtracted from p and p_1 giving $p = .537$, $p_1 = .607$, the quantity of steam used without clearance being greater ce is 7 per cent in the ratio of 607 : 537, or 13% more.

npression.—By early closure of the exhaust, so that a ain-t-steam is compressed into the clearance-space, much clearance may be avoided. If expansion is continued pressure, if the back pressure is uniform throughout the d if compression begins at such point that the exhaust- n the cylinder is compressed to the initial pressure at the roke, then the work of compression of the exhaust-steam one during expansion by the clearance-steam. The clear- illed by the exhaust steam thus compressed, no new steam he clearance-space for the next forward stroke, and the y of the steam used in the cylinder are just the same as if arance and no compression. When, however, there is a rom the final pressure of the expansion, or the terminal haust or back pressure (the usual case), the work of com- tial pressure is greater than the work done by the expan- nce-steam, so that a loss of efficiency results. In this ciency can be attained by inclosing for compression a less than that needed to fill the clearance-space with steam of . (See Clark, S. E., p. 399, *et seq.*; also F. H. Ball, Trans. 67.) It is shown by Clark that a somewhat greater effi- ined whether or not the pressure of the steam be carried n to the back exhaust-pressure. As a result of calcula- e the most efficient periods of compression for various k pressure, and for various periods of admission, he gives xt page:

n Low- and High-speed Engines. (Harris ., Sept. 17, 1891.)—The construction of the high-speed h its relatively short stroke, that the clearance must be in the releasing-valve type. The short-stroke engine is, ngine with large clearance, which is aggravated when a ion is a feature. Conversely, the releasing-valve gear is, n engine of slow rotative speed, where great power is ng stroke, and small clearance is a feature in its construc- the clearance will vary from 8% to 12% of the piston-dis- the other from 2% to 3%. In the case of an engine with a g *10% of the piston-displacement* the waste room becomes *nsidered in connection* with an early cut-off. The system of *es the waste due* to clearance in proportion as the steam *ver pressure. The* farther expansion is carried through *the greater will* be the reduction of waste due to *from the fact* that the high-speed engine, e steam much less than the Corliss, will show a greater gain when cha[illegible] from simple to compound than its rival under similar conditions.

COMPRESSION OF STEAM IN THE CYLINDER.

Best Periods of Compression; Clearance 7 per cent.

Cut-off in Percentages of the Stroke.	Total Back Pressure, in percentages of the total initial pres[illegible]						
	2½	5	10	15	20	25	30
	Periods of Compression, in parts of the stroke.						
10%	65%	57%	44%	32%	....	....	....
15	58	52	40	29	23%	.....	....
20	52	47	37	27	22	......	....
25	47	42	34	26	21	17%	....
30	42	39	32	25	20	16	14%
35	39	35	29	23	19	15	13
40	36	32	27	21	18	14	13
45	33	30	25	20	17	14	12
50	30	27	23	18	16	13	12
55	27	24	21	17	15	13	11
60	24	22	19	15	14	12	11
65	22	20	17	15	14	12	10
70	19	17	16	14	14	12	10
75	17	16	14	13	12	11	9

NOTES TO TABLE.—1. For periods of admission, or percentages o[illegible] pressure, other than those given, the periods of compression may be [illegible] found by interpolation.

2. For any other clearance, the values of the tabulated periods [illegible] pression are to be altered in the ratio of 7 to the given percent[illegible] clearance.

Cylinder-condensation may have considerable effect upon the bes[illegible] of compression, but it has not yet (1893) been determined by experi[illegible] (Trans. A. S. M. E., xiv. 1078.)

Cylinder-condensation.—Rankine, S. E., p. 421, says: Con[illegible] of heat to and from the metal of the cylinder, or to and from liqui[illegible] contained in the cylinder, has the effect of lowering the pressure at [illegible] ginning and raising it at the end of the stroke, the lowering effect [illegible] the whole greater than the raising effect. In some experiments the q[illegible] of steam wasted through alternate liquefaction and evaporation [illegible] cylinder has been found to be greater than the quantity which per[illegible] the work.

Percentage of Loss by Cylinder-condensation, tak[illegible] Cut-off. (From circular of the Ashcroft Mfg. Co. on the Indicator, 1889.)

Percentage of Stroke completed at Cut-off.	Percent. of Feed-water accounted for by the Indicator diagram.			Percent. of Feed-water Co[illegible] tion due to Cylinder-cond[illegible]	
	Simple Engines.	Compound Engines, h.p. cyl.	Triple-expansion Engines, h.p. cyl.	Simple Engines.	Compound Engines, h.p. cyl.
5	58	....	....	42	....
10	66	74	....	34	26
15	71	76	78	29	24
20	74	78	80	26	22
30	78	82	84	22	18
40	82	85	87	18	15
[illegible]	86	[illegible]	90	14	13

heoretical Compared with Actual Water-consump-on, Single-cylinder Automatic Cut-off Engines. (From e catalogue of the Buckeye Engine Co.)—The following table has been epared on the basis of the pressures that result in practice with a con-ant boiler-pressure of 80 lbs. and different points of cut-off, with Buckeye gines and others with similar clearance. Fractions are omitted, except the percentage column, as the degree of accuracy their use would seem imply is not attained or aimed at.

ut-off Part of Stroke.	Mean Effective Pressure.	Total Terminal Pressure.	Indicated Rate, lbs. Water, per I.H.P. per hour.	Assumed.	
				Act'l Rate.	Per ct. Loss.
.10	18	11	20	32	58
.15	27	15	19	27	41
.20	35	20	19	25	31.5
.25	42	25	20	25	25
.30	48	30	20	24	21.8
.35	53	35	21	25	19
.40	57	38	22	26	16.7
.45	61	43	23	27	15
.50	64	48	24	27	13.6

t will be seen that while the best indicated economy is when the cut-off about at .15 or .20 of the stroke, giving about 30 lbs. M.E.P., and a termi-l 3 or 4 lbs. above atmosphere, when we come to add the percentages due e constant amount of unindicated loss, as per sixth column, the most eco-mical point of cut-off is found to be about .30 of the stroke, giving 48 lbs. E.P. and 30 lbs. terminal pressure. This showing agrees substantially th modern experience under automatic cut-off regulation.

Experiments on Cylinder-condensation.—Experiments by jor Thos. English (*Eng'g*, Oct. 7, 1887, p. 386) with an engine 10 × 14 in., keted in the sides but not on the ends, indicate that the net initial con-nsation (or excess of condensation over re-evaporation) by the clearance rface varies directly as the initial density of the steam, and inversely as e square root of the number of revolutions per unit of time. The mean sults gave for the net initial condensation by clearance-space per sq. ft. of rface at one rev. per second 6.06 thermal units in the engine when run n-condensing and 5.75 units when condensing.

G. R. Bodmer (*Eng'g*, March 4, 1892, p. 299) says: Within the ordinary nits of expansion desirable in one cylinder the expansion ratio has prac-ally no influence on the amount of condensation per stroke, which for mple engines can be expressed by the following formula for the weight water condensed [per minute, probably; the original does not state]:

$= C\frac{S(T-t)}{L\sqrt[3]{N^2}}$, where T denotes the mean admission temperature, t the ean exhaust temperature, S clearance-surface (square feet), N the num-r of revolutions per second, L latent heat of steam at the mean admission mperature, and C a constant for any given type of engine.

Mr. Bodmer found from experimental data that for high-pressure non-cketed engines C = about 0.11, for condensing non-jacketed engines 0.085 0.11, for condensing jacketed engines 0.085 to 0.053. The figures for jack-ed engines apply to those jacketed in the usual way, and not at the ends. C varies for different engines of the same class, but is practically con-ant for any given engine. For simple high-pressure non-jacketed engines was found to range from 0.1 to 0.112.

Applying Mr. Bodmer's formula to the case of a Corliss non-jacketed non-ndensing engine, 4-ft. stroke, 24 in. diam., 60 revs. per min., initial pres-re 90 lbs. gauge, exhaust pressure 2 lbs., we have $T - t = 112°$, $N = 1$, $= 880$, $S = 7$ sq. ft.; and, taking $C = .112$ and W = lbs. water condensed r minute, $W = \frac{.112 \times 112 \times 7}{1 \times 880} = .09$ lb. per minute, or 5.4 lbs. per hour. I

steam used per I.H.P. per hour according to the diagram is 20 lbs., t *al water consumption is 25.4 lbs.,* corresponding to a cylinder conde *of 27%.*

INDICATOR-DIAGRAM OF A SINGLE-CYLIN ENGINE.

Definitions.—*The Atmospheric Line, AB*, is a line drawn by th of the indicator when the connections with the engine are closed a sides of the piston are open to the atmosphere.

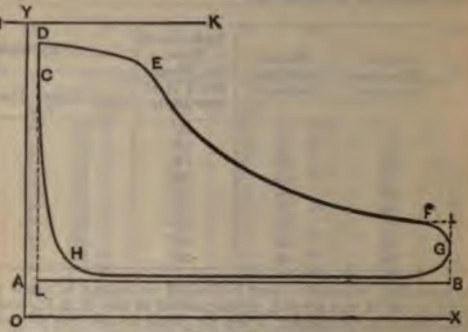


Fig. 138.

The Vacuum Line, OX, is a reference line usually drawn about pounds by scale below the atmospheric line.

The Clearance Line, OY, is a reference line drawn at a distance fr end of the diagram equal to the same per cent of its length as the cle and waste room is of the piston-displacement.

The Line of Boiler-pressure, JK, is drawn parallel to the atmos line, and at a distance from it by scale equal to the boiler-pressure by the gauge.

The Admission Line, CD, shows the rise of pressure due to the adm of steam to the cylinder by opening the steam-valve.

The Steam Line, DE, is drawn when the steam-valve is open and st being admitted to the cylinder.

The Point of Cut-off, E, is the point where the admission of st stopped by the closing of the valve. It is often difficult to determi exact point at which the cut-off takes place. It is usually located whe outline of the diagram changes its curvature from convex to concave

The Expansion Curve, EF, shows the fall in pressure as the steam cylinder expands doing work.

The Point of Release, F, shows when the exhaust-valve opens.

The Exhaust Line, FG, represents the change in pressure that place when the exhaust-valve opens.

The Back-pressure Line, GH, shows the pressure against which the acts during its return stroke.

The Point of Exhaust Closure, H, is the point where the exhaus closes. It cannot be located definitely, as the change in pressure is due to the gradual closing of the valve.

The Compression Curve, HC, shows the rise in pressure due to the pression of the steam remaining in the cylinder after the exhaust-val closed.

The Mean Height of the Diagram equals its area divided by its len

The Mean Effective Pressure is the mean net pressure urging the forward = the mean height × the scale of the indicator-spring.

To find the Mean Effective Pressure from the Diagram.—Divi length, *LB*, into a number, say 10, equal parts, setting off half a par half a part at *B*, and nine other parts between; erect ordinates perpe lar to the atmospheric line at the points of division of *LB*, cutting th gram; add together the lengths of these ordinates intercepted betwe upper and lower lines of the diagram and divide by their number.

height, which multiplied by the scale of the indicator-spring .P. Or find the area by a planimeter, or other means (see . 55), and divide by the length LB to obtain the mean height. *ressure* is the pressure acting on the piston at the beginning

l Pressure is the pressure above the line of perfect vacuum st at the end of the stroke if the steam had not been released ound by continuing the expansion-curve to the end of the

D HORSE-POWER OF ENGINES, SINGLE-CYLINDER.

$$\text{Indicated Horse-power I.H.P.} = \frac{PLan}{33,000},$$

ean effective pressure in lbs. per sq. in.; L = length of stroke ea of piston in square inches. For accuracy, one half of the of the piston-rod must be subtracted from the area of the d passes through one head, or the whole area of the rod if it both heads; n = No. of single strokes per min. = 2 × No. of

$= \frac{PaS}{33,000}$, in which S = piston speed in feet per minute.

$$= \frac{PLd^2n}{42,017} = \frac{Pd^2S}{42,017} = .0000238PLd^2n = .0000238Pd^2S,$$

diam. of cyl. in inches. (The figures 238 are exact, since 8 exactly.) If product of piston-speed × mean effective 017, then the horse-power would equal the square of the hes.

ule for Estimating the Horse-power of a nder Engine.—Square the diameter and divide by 2. This is ver the product of the mean effective pressure and the piston- 42,017, or, say, 21,000, viz., when M.E.P. = 30 and $S = 700$; 35 and $S = 600$; when M.E.P. = 38.2 and $S = 550$; and when d $S = 500$. These conditions correspond to those of ordinary oth Corliss engines and shaft-governor high-speed engines.

orse-power, Mean Effective Pressure, and d, to find Size of Cylinder.—

$$\frac{33,000 \times \text{I.H.P.}}{PLn}. \qquad \text{Diameter} = 205\sqrt{\frac{\text{I.H.P.}}{PS}}. \text{ (Exact.)}$$

rse-power is the actual horse-power of the engine as e fly-wheel by a friction-brake or dynamometer. It is the -power minus the friction of the engine.

Roughly Approximating the Horse-power of nd Engine from the Diameter of its Low-ylinder.—The indicated horse-power of an engine being

n P = mean effective pressure per sq. in., s = piston-speed in

and d = diam. of cylinder in inches; if s = 600 ft. per min., ximately the speed of modern stationary engines, and $P = 35$ an approximately average figure for the M.E.P. of single- es, and of compound engines referred to the low-pressure I.H.P. = $\frac{1}{2}d^2$; hence the rough-and-ready rule for horse-power quare the diameter in inches and divide by 2. This applies to druple expansion engines as well as to single cylinder and or most economical loading, the M.E.P. referred to the low- der of compound engines is usually not greater than that of ; for the greater economy is obtained by a greater number of steam of higher pressures, and the greater the number of a given initial pressure the lower the mean effective pressure. able gives *approximately* the figures of mean total and effective pressures for the different types of engines, together with the factor which the square of the diameter is to be multiplied to obtain the hor power at most economical loading, for a piston-speed of 600 ft. per minu

Type of Engine.	Initial Absolute Steam-pressure.	Number of Expansions.	Terminal Absolute Press., lbs.	Ratio Mean Total to Initial Pressure.	Mean Total Pressure, lbs.	Total Back Pressure, Mean, lbs.	Mean Effective Pressure, lbs.	Piston-speed, ft. per min.	Horse-
Non-condensing.									
Single Cylinder.	100	5.	20	.522	52.2	15.5	36.7	600	[illegible]
Compound	120	7.5	16	.402	48.2	15.5	32.7	"	[illegible]
Triple...........	160	10.	16	.330	52.8	15.5	37.3	"	[illegible]
Quadruple......	200	12.5	16	.282	56.4	15.5	40.9	"	[illegible]
Condensing Engines.									
Single Cylinder.	100	10.	10	.330	33.0	2	31.0	600	[illegible]
Compound.....	120	15.	8	.247	29.6	2	27.6	"	[illegible]
Triple.	160	20.	8	.200	32.0	2	30.0	"	[illegible]
Quadruple......	200	25.	8	.169	33.8	2	31.8	"	[illegible]

For any other piston-speed than 600 ft. per min., multiply the figure the last column by the ratio of the piston-speed to 600 ft.

Nominal Horse-power.—The term "nominal horse-power" o nated in the time of Watt, and was used to express approximately the po of an engine as calculated from its diameter, estimating the mean pres in the cylinder at 7 lbs. above the atmosphere. It has long been obsolet America, and is nearly obsolete in England.

Horse-power Constant of a given Engine for a Fix Speed = product of its area of piston in square inches, length of strok feet, and number of single strokes per minute divided by 33,000, or $\frac{L}{33}$ $= C$. The product of the mean effective pressure as found by the diag and this constant is the indicated horse-power.

Horse-power Constant of a given Engine for Varyi Speeds = product of its area of piston and length of stroke divided 33,000. This multiplied by the mean effective pressure and by the num of single strokes per minute is the indicated horse-power.

Horse-power Constant of any Engine of a given Dia eter of Cylinder, whatever the length of stroke = area of piston ÷ 3 = square of the diameter of piston in inches × .0000238. A table of const derived from this formula is given below.

The constant multiplied by the piston-speed in feet per minute and the M.E.P. gives the I.H.P.

Errors of Indicators.—The most common error is that of the spr which may vary from its normal rating; the error may be determined proper testing apparatus and allowed for. But after making this correc even with the best work, the results are liable to variable errors which amount to 2 or 3 per cent. See Barrus, Trans. A. S. M. E., v. 310; Den A. S. M. E., xi. 329; David Smith, U. S. N., Proc. Eng'g Congress, Marine Division.

Indicator "Rigs," or Reducing-motions; Interpretation of Diagrams Errors of Steam-distribution, etc. For these see circulars of manufactu of Indicators; also works on the Indicator.

Table of Engine Constants for Use in Figuring Hor power.—"Horse-power constant" for cylinders from 1 inch to 60 inche diameter, advancing by 8ths, for one foot of piston-speed per minute and pound of M.E.P. Find the diameter of the cylinder in the column at side. If the diameter contains no fraction the constant will be found in *column* headed Even Inches. If the diameter is not in even inches, fo *the line* horizontally to the column corresponding to the required fract constants multiplied by the piston-speed and by the M.E.P. give the -power.

eter f der.	Even Inches.	+ ⅛ or .125.	+ ¼ or .25.	+ ⅜ or .375.	+ ½ or .5.	+ ⅝ or .625.	+ ¾ or .75.	+ ⅞ or .875.
1	.0000238	.0000301	.0000372	.0000450	.0000535	.0000628	.0000729	.0000837
2	.0000952	.0001074	.0001205	.0001342	.0001487	.0001640	.0001800	.0001967
3	.0002142	.0002324	.0002514	.0002711	.0002915	.0003127	.0003347	.0003574
4	.0003808	.0004050	.0004299	.0004554	.0004819	.0005091	.0005370	.0005656
5	.0005950	.0006251	.0006560	.0006876	.0007199	.0007530	.0007869	.0008215
6	.0008568	.0008929	.0009297	.0009672	.0010055	.0010445	.0010844	.0011249
7	.0011662	.0012082	.0012510	.0012944	.0013387	.0013837	.0014295	.0014759
8	.0015232	.0015711	.0016198	.0016693	.0017195	.0017705	.0018222	.0018746
9	.0019278	.0019817	.0020363	.0020916	.0021479	.0022048	.0022625	.0023209
0	.0023800	.0024398	.0025004	.0025618	.0026239	.0026867	.0027502	.0028147
1	.0028798	.0029456	.0030121	.0030794	.0031475	.0032163	.0032859	.0033561
2	.0034272	.0034990	.0035714	.0036447	.0037187	.0037934	.0038690	.0039452
3	.0040222	.0040999	.0041783	.0042576	.0043375	.0044182	.0044997	.0045819
4	.0046648	.0047484	.0048328	.0049181	.0050039	.0050906	.0051780	.0052661
5	.0053550	.0054446	.0055349	.0056261	.0057179	.0058105	.0059039	.0059979
6	.0060928	.0061884	.0062847	.0063817	.0064795	.0065780	.0066774	.0067774
7	.0068782	.0069797	.0070819	.0071850	.0072887	.0073932	.0074985	.0076044
8	.0077112	.0078187	.0079268	.0080360	.0081452	.0082560	.0083672	.0084791
9	.0085918	.0087052	.0088198	.0089343	.0090499	.0091663	.0092835	.0094013
0	.0095200	.0096393	.0097594	.0098803	.0100019	.0101243	.0102474	.0103712
1	.0104958	.0106211	.0107472	.0108739	.0110015	.0111299	.0112589	.0113886
2	.0115192	.0116505	.0117825	.0119152	.0120487	.0121830	.0123179	.0124537
3	.0125902	.0127274	.0128654	.0130040	.0131435	.0132837	.0134247	.0135664
4	.0137088	.0138519	.0139959	.0141405	.0142859	.0144321	.0145789	.0147266
5	.0148750	.0150241	.0151739	.0153246	.0154759	.0156280	.0157809	.0159345
6	.0160888	.0162439	.0163997	.0165563	.0167135	.0168716	.0170304	.0171899
7	.0173502	.0175112	.0176729	.0178355	.0179988	.0181627	.0183275	.0184929
8	.0186592	.0188262	.0189939	.0191624	.0193316	.0195015	.0196722	.0198436
9	.0200158	.0201887	.0203674	.0205368	.0207119	.0208879	.0210645	.0212418
0	.0214200	.0215988	.0217785	.0219588	.0221399	.0223218	.0225044	.0226877
1	.0228718	.0230566	.0232422	.0234285	.0236155	.0238033	.0239919	.0241812
2	.0243712	.0245619	.0247535	.0249457	.0251387	.0253325	.0255269	.0257222
3	.0259182	.0261149	.0263124	.0265106	.0267095	.0269092	.0271097	.0273109
4	.0275128	.0277155	.0279189	.0281231	.0283279	.0285336	.0287399	.0289471
5	.0291550	.0293636	.0295729	.0297831	.0299939	.0302056	.0304179	.0306309
6	.0308448	.0310594	.0312747	.0314908	.0317075	.0319251	.0321434	.0323624
7	.0325822	.0328027	.0330239	.0332460	.0334687	.0336922	.0339165	.0341415
8	.0343672	.0345937	.0348209	.0350489	.0352775	.0355070	.0357372	.0359681
9	.0361998	.0364322	.0366654	.0368993	.0371339	.0373694	.0376055	.0378424
0	.0380800	.0383184	.0385575	.0387973	.0390379	.0392793	.0395214	.0397642
1	.0400078	.0402521	.0404972	.0407430	.0409895	.0412368	.0414849	.0417337
2	.0419832	.0422335	.0424845	.0427362	.0429887	.0432420	.0434959	.0437507
3	.0440062	.0442624	.0445194	.0447771	.0450355	.0452947	.0455547	.0458154
4	.0460768	.0463389	.0466019	.0468655	.0471299	.0473951	.0476609	.0479276
5	.0481950	.0484631	.0487320	.0490016	.0492719	.0495430	.0498149	.0500875
6	.0503608	.0506349	.0509097	.0511853	.0514615	.0517386	.0520164	.0522949
7	.0525742	.0528542	.0531349	.0534165	.0536988	.0539818	.0542655	.0545499
8	.0548352	.0551212	.0554079	.0556953	.0559835	.0562725	.0565622	.0568526
9	.0571438	.0574357	.0577284	.0580218	.0583159	.0586109	.0589065	.0592029
0	.0595000	.0597979	.0600965	.0603959	.0606959	.0609969	.0612984	.0616007
1	.0619038	.0622076	.0625122	.0628175	.0632235	.0634304	.0637379	.0640462
2	.0643552	.0646649	.0649753	.0652867	.0655987	.0659115	.0662250	.0665392
3	.0668542	.0671699	.0674864	.0678036	.0681215	.0684402	.0687597	.0690799
4	.0694008	.0697225	.0700449	.0703681	.0705293	.0710166	.0713419	.0716681
5	.0719950	.0724226	.0726510	.0729801	.0733099	.0736406	.0739719	.0743039
6	.0746368	.0749704	.0753047	.0756398	.0759755	.0763120	.0766494	.0769874
7	.0773262	.0776657	.0780060	.0783476	.0786887	.0790312	.0793745	.0797185
8	.0800632	.0804087	.0807549	.0811019	.0814495	.0817980	.0821472	.0824971
9	.0828478	.0831992	.0835514	.0839043	.0842579	.0846123	.0849675	.0853234
0	.0856800	.0860374	.0863955	.0867543	.0871139	.0874743	.0878354	.0881973

INDICATOR-DIAGRAM OF A SINGLE-CYLINDE ENGINE.

Definitions.—*The Atmospheric Line, AB*, is a line drawn by the p of the indicator when the connections with the engine are closed and sides of the piston are open to the atmosphere.

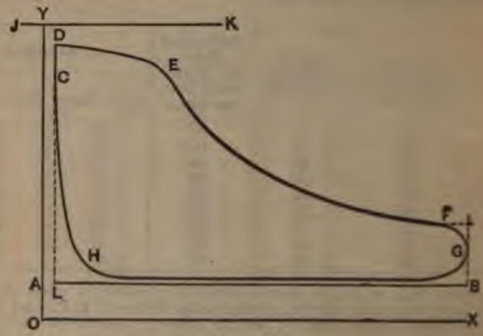


FIG. 138.

The Vacuum Line, OX, is a reference line usually drawn about 14 pounds by scale below the atmospheric line.

The Clearance Line, OY, is a reference line drawn at a distance from end of the diagram equal to the same per cent of its length as the clear and waste room is of the piston-displacement.

The Line of Boiler-pressure, JK, is drawn parallel to the atmosph line, and at a distance from it by scale equal to the boiler-pressure sh by the gauge.

The Admission Line, CD, shows the rise of pressure due to the admi of steam to the cylinder by opening the steam-valve.

The Steam Line, DE, is drawn when the steam-valve is open and stea being admitted to the cylinder.

The Point of Cut-off, E, is the point where the admission of stea stopped by the closing of the valve. It is often difficult to determine exact point at which the cut-off takes place. It is usually located where outline of the diagram changes its curvature from convex to concave.

The Expansion Curve, EF, shows the fall in pressure as the steam in cylinder expands doing work.

The Point of Release, F, shows when the exhaust-valve opens.

The Exhaust Line, FG, represents the change in pressure that place when the exhaust-valve opens.

The Back-pressure Line, GH, shows the pressure against which the p acts during its return stroke.

The Point of Exhaust Closure, H, is the point where the exhaust-v closes. It cannot be located definitely, as the change in pressure is al due to the gradual closing of the valve.

The Compression Curve, HC, shows the rise in pressure due to the pression of the steam remaining in the cylinder after the exhaust-valve is closed.

The Mean Height of the Diagram equals its area divided by its length.

The Mean Effective Pressure is the mean net pressure urging the pi forward = the mean height × the scale of the indicator-spring.

To find the Mean Effective Pressure from the Diagram.—Divide length, *LB*, into a number, say 10, equal parts, setting off half a part half a part at *B*, and nine other parts between; erect ordinates perpe to the atmospheric line at the points of division of LB, cutting add the lengths of these ordinates intercepted betw of the diagram and divide by their numb ean height, which multiplied by the scale of the indicator-spring M.E.P. Or find the area by a planimeter, or other means (see n, p. 55), and divide by the length LB to obtain the mean height. *il Pressure* is the pressure acting on the piston at the beginning ke.

inal Pressure is the pressure above the line of perfect vacuum exist at the end of the stroke if the steam had not been released is found by continuing the expansion-curve to the end of the

TED HORSE-POWER OF ENGINES, SINGLE-CYLINDER.

$$\text{Indicated Horse-power I.H.P.} = \frac{PLan}{33,000},$$

= mean effective pressure in lbs. per sq. in.; L = length of stroke = area of piston in square inches. For accuracy, one half of the rea of the piston-rod must be subtracted from the area of the ie rod passes through one head, or the whole area of the rod if it ough both heads; n = No. of single strokes per min. = $2 \times$ No. of .

$$\text{.P.} = \frac{PaS}{33,000}, \text{ in which } S = \text{piston speed in feet per minute.}$$

$$\text{.P.} = \frac{PLd^2n}{42,017} = \frac{Pd^2S}{42,017} = .0000238PLd^2n = .0000238Pd^2S,$$

d = diam. of cyl. in inches. (The figures 238 are exact, since = 23.8 exactly.) If product of piston-speed $\times$ mean effective 42,017, then the horse-power would equal the square of the n inches.

Rule for Estimating the Horse-power of a ylinder Engine.—Square the diameter and divide by 2. This is enever the product of the mean effective pressure and the piston- of 42,017, or, say, 21,000, viz., when M.E.P. = 30 and $S = 700$; P. = 35 and $S = 600$; when M.E.P. = 38.2 and $S = 550$; and when 2 and $S = 500$. These conditions correspond to those of ordinary ith both Corliss engines and shaft-governor high-speed engines.

Horse-power, Mean Effective Pressure, and peed, to find Size of Cylinder.—

$$\text{ea} = \frac{33,000 \times \text{I.H.P.}}{PLn}. \qquad \text{Diameter} = 205\sqrt{\frac{\text{I.H.P.}}{PS}}. \text{ (Exact.)}$$

Horse-power is the actual horse-power of the engine as at the fly-wheel by a friction-brake or dynamometer. It is the horse-power minus the friction of the engine.

for Roughly Approximating the Horse-power of ound Engine from the Diameter of its Low- Cylinder.—The indicated horse-power of an engine being

which P = mean effective pressure per sq. in., s = piston-speed in

n., and d = diam. of cylinder in inches; if $s = 600$ ft. per min., pproximately the speed of modern stationary engines, and $P = 35$ is an approximately average figure for the M.E.P. of single- ngines, and of compound engines referred to the low-pressure hen I.H.P. = $\frac{1}{2}d^2$; hence the rough-and-ready rule for horse-power e: Square the diameter in inches and divide by 2. This applies to quadruple expansion engines as well as to single cylinder and . For most economical loading, the M.E.P. referred to the low- *ylinder of compound engines is usually not greater than that of ines; for the greater economy is obtained by a greater number of of steam of higher pressures, and the greater the number of or a given initial pressure the lower the mean effective pressure. table gives approximately the figures of mean total and effec* tive pressures for the different types of engines, together with the factor which the square of the diameter is to be multiplied to obtain the horse-power at most economical loading, for a piston-speed of 600 ft. per min.

Type of Engine.	Initial Absolute Steam-pressure.	Number of Expansions.	Terminal Absolute Press., lbs.	Ratio Mean Total to Initial Pressure.	Mean Total Pressure, lbs.	Total Back Pressure, Mean, lbs.	Mean Effective Pressure, lbs.	Piston-speed, ft. per min.
Non-condensing.								
Single Cylinder.	100	5.	20	.522	52.2	15.5	36.7	600
Compound	120	7.5	16	.402	48.2	15.5	32.7	"
Triple...........	160	10.	16	.330	52.8	15.5	37.3	"
Quadruple......	200	12.5	16	.282	56.4	15.5	40.9	"
Condensing Engines.								
Single Cylinder.	100	10.	10	.330	33.0	2	31.0	600
Compound......	120	15.	8	.247	29.6	2	27.6	"
Triple....... ...	160	20.	8	.200	32.0	2	30.0	"
Quadruple......	200	25.	8	.169	33.8	2	31.8	"

For any other piston-speed than 600 ft. per min., multiply the figures in the last column by the ratio of the piston-speed to 600 ft.

Nominal Horse-power.—The term "nominal horse-power" originated in the time of Watt, and was used to express approximately the power of an engine as calculated from its diameter, estimating the mean pressure in the cylinder at 7 lbs. above the atmosphere. It has long been obsolete in America, and is nearly obsolete in England.

Horse-power Constant of a given Engine for a Fixed Speed = product of its area of piston in square inches, length of stroke in feet, and number of single strokes per minute divided by 33,000, or $\frac{aln}{33000}$ = C. The product of the mean effective pressure as found by the diagram and this constant is the indicated horse-power.

Horse-power Constant of a given Engine for Various Speeds = product of its area of piston and length of stroke divided by 33,000. This multiplied by the mean effective pressure and by the number of single strokes per minute is the indicated horse-power.

Horse-power Constant of any Engine of a given Diameter of Cylinder, whatever the length of stroke = area of piston ÷ 33,000 = square of the diameter of piston in inches × .0000238. A table of constants derived from this formula is given below.

The constant multiplied by the piston-speed in feet per minute and by the M.E.P. gives the I.H.P.

Errors of Indicators.—The most common error is that of the spring, which may vary from its normal rating; the error may be determined by proper testing apparatus and allowed for. But after making this correction, even with the best work, the results are liable to variable errors which may amount to 2 or 3 per cent. See Barrus, Trans. A. S. M. E., v. 310; Trans. A. S. M. E., xi. 329; David Smith, U. S. N., Proc. Eng'g Congress, Marine Division.

Indicator "Rigs," or Reducing-motions; Interpretation of Diagrams, Errors of Steam-distribution, etc. For these see circulars of manufacturers of Indicators; also works on the Indicator.

Table of Engine Constants for Use in Figuring Horse-power.—"Horse-power constant" for cylinders from 1 inch to 60 in. diameter, advancing by 8ths, for one foot of piston-speed per minute and one pound of M.E.P. Find the diameter of the cylinder in the column at the side. If the diameter contains no fraction the constant will be found in the column headed Even Inches. If the diameter is not in even inches, follow the line horizontally to the column corresponding to the required fraction.

constants multiplied by the piston-speed and by the M.E.P. give the
e-power.

neter of nder.	Even Inches.	+ ⅛ or .125.	+ ¼ or .25.	+ ⅜ or .375.	+ ½ or .5.	+ ⅝ or .625.	+ ¾ or .75.	+ ⅞ or .875.
1	.0000238	.0000301	.0000372	.0000450	.0000535	.0000628	.0000729	.0000837
2	.0000952	.0001074	.0001205	.0001342	.0001487	.0001640	.0001800	.0001967
3	.0002142	.0002324	.0002514	.0002711	.0002915	.0003127	.0003347	.0003574
4	.0003808	.0004050	.0004299	.0004554	.0004819	.0005091	.0005370	.0005656
5	.0005950	.0006251	.0006560	.0006876	.0007199	.0007530	.0007869	.0008215
6	.0008568	.0008929	.0009297	.0009672	.0010055	.0010445	.0010844	.0011249
7	.0011662	.0012082	.0012510	.0012944	.0013387	.0013837	.0014295	.0014759
8	.0015232	.0015711	.0016198	.0016693	.0017195	.0017705	.0018222	.0018746
9	.0019278	.0019817	.0020363	.0020916	.0021479	.0022048	.0022625	.0023209
0	.0023800	.0024398	.0025004	.0025618	.0026239	.0026867	.0027502	.0028147
1	.0028798	.0029456	.0030121	.0030794	.0031475	.0032163	.0032859	.0033561
2	.0034272	.0034990	.0035714	.0036447	.0037187	.0037934	.0038690	.0039452
3	.0040222	.0040999	.0041783	.0042576	.0043375	.0044182	.0044997	.0045819
4	.0046648	.0047484	.0048328	.0049181	.0050039	.0050906	.0051780	.0052661
5	.0053550	.0054446	.0055349	.0056261	.0057179	.0058105	.0059039	.0059979
6	.0060928	.0061884	.0062847	.0063817	.0064795	.0065780	.0066774	.0067774
7	.0068782	.0069797	.0070819	.0071850	.0072887	.0073932	.0074985	.0076044
8	.0077112	.0078187	.0079268	.0080360	.0081452	.0082560	.0083672	.0084791
9	.0085918	.0087052	.0088193	.0089343	.0090499	.0091663	.0092835	.0094013
0	.0095200	.0096395	.0097594	.0098803	.0100019	.0101243	.0102474	.0103712
1	.0104958	.0106211	.0107472	.0108739	.0110015	.0111299	.0112589	.0113886
2	.0115192	.0116505	.0117825	.0119152	.0120487	.0121830	.0123179	.0124537
3	.0125902	.0127274	.0128654	.0130040	.0131435	.0132837	.0134247	.0135664
4	.0137088	.0138519	.0139959	.0141405	.0142859	.0144321	.0145789	.0147266
5	.0148750	.0150241	.0151739	.0153246	.0154759	.0156280	.0157809	.0159345
6	.0160888	.0162439	.0163997	.0165563	.0167135	.0168716	.0170304	.0171899
7	.0173502	.0175112	.0176729	.0178355	.0179988	.0181627	.0183275	.0184929
8	.0186592	.0188262	.0189939	.0191624	.0193316	.0195015	.0196722	.0198436
9	.0200158	.0201887	.0203674	.0205368	.0207119	.0208879	.0210645	.0212418
0	.0214200	.0215988	.0217785	.0219588	.0221399	.0223218	.0225044	.0226877
1	.0228718	.0230566	.0232422	.0234285	.0236155	.0238033	.0239919	.0241812
2	.0243712	.0245619	.0247535	.0249457	.0251387	.0253325	.0255269	.0257222
3	.0259182	.0261149	.0263124	.0265106	.0267095	.0269092	.0271097	.0273109
4	.0275128	.0277155	.0279189	.0281231	.0283279	.0285336	.0287399	.0289471
5	.0291550	.0293636	.0295729	.0297831	.0299939	.0302056	.0304179	.0306309
6	.0308448	.0310594	.0312747	.0314908	.0317075	.0319251	.0321434	.0323624
7	.0325822	.0328027	.0330239	.0332460	.0334687	.0336922	.0339165	.0341415
8	.0343672	.0345937	.0348209	.0350489	.0352775	.0355070	.0357372	.0359681
9	.0361998	.0364322	.0366654	.0368993	.0371339	.0373694	.0376055	.0378424
0	.0380800	.0383184	.0385575	.0387973	.0390379	.0392793	.0395214	.0397642
1	.0400078	.0402521	.0404972	.0407430	.0409895	.0412368	.0414849	.0417337
2	.0419832	.0422335	.0424845	.0427362	.0429887	.0432420	.0434959	.0437507
3	.0440062	.0442624	.0445194	.0447771	.0450355	.0452947	.0455547	.0458154
4	.0460768	.0463389	.0466019	.0468655	.0471299	.0473951	.0476609	.0479276
5	.0481950	.0484631	.0487320	.0490016	.0492719	.0495430	.0498149	.0500875
6	.0503608	.0506349	.0509097	.0511853	.0514615	.0517386	.0520164	.0522949
7	.0525742	.0528542	.0531349	.0534165	.0536988	.0539818	.0542655	.0545499
8	.0548352	.0551212	.0554079	.0556953	.0559835	.0562725	.0565622	.0568526
9	.0571438	.0574357	.0577284	.0580218	.0583159	.0586109	.0589065	.0592029
0	.0595000	.0597979	.0600965	.0603959	.0606959	.0609969	.0612984	.0616007
1	.0619088	.0622076	.0625122	.0628175	.0632235	.0634304	.0637379	.064046[illegible]
2	.0643552	.0646649	.0649753	.0652867	.0655987	.0659115	.0662250	.066539[illegible]
3	.0668542	.0671699	.0674864	.0678036	.0681215	.0684402	.0687597	.069079[illegible]
4	.0694008	.0697225	.0700449	.0703681	.0705298	.0710166	.0713419	.0716681
5	.0719950	.0724226	.0726510	.0729801	.0733099	.0736406	.0739719	[illegible]
6	.0746368	.0749704	.0753047	.0756398	.0759755	.0763120	.0766494	[illegible]
	.0773262	.0776657	.0780060	.0783476	.0786887	.0790312	.0793745	[illegible]
	.0800682	.0804087	.0807549	.0811019	.0814495	.0817980	.082147[illegible]	[illegible]
	.0828478	.0831992	.0835514	.0839043	.0842579	.0846123	.08496[illegible]	[illegible]
	.0856800	.0860374	.0863955	.0867543	.0871139	.0874743	.0878[illegible]	[illegible]

Horse-power per Pound Mean Effective Press

Formula, $\frac{\text{Area in sq. in.} \times \text{piston-speed}}{33.000}$.

Diam. of Cylinder, inches.	Speed of Piston in feet per minute.						
	100	200	300	400	500	600	700	800
4	.0381	.0762	.1142	.1523	.1904	.2285	.2666	[illegible]
4½	.0482	.0964	.1446	.1928	.2410	.2892	.3374	[illegible]
5	.0595	.1190	.1785	.2380	.2975	.3570	.4165	[illegible]
5½	.0720	.1440	.2160	.2880	.3600	.4320	.5040	[illegible]
6	.0857	.1714	.2570	.3427	.4284	.5141	.5998	[illegible]
6½	.1006	.2011	.3017	.4022	.5028	.6033	.7039	[illegible]
7	.1166	.2332	.3499	.4665	.5831	.6997	.8163	[illegible]
7½	.1339	.2678	.4016	.5355	.6694	.8033	.9371	[illegible]
8	.1523	.3046	.4570	.6093	.7616	.9139	1.0662	[illegible]
8½	.1720	.3439	.5159	.6878	.8598	1.0317	1.2037	[illegible]
9	.1928	.3856	.5783	.7711	.9639	1.1567	1.3495	[illegible]
9½	.2148	.4296	.6444	.8592	1.0740	1.2888	1.5036	[illegible]
10	.2380	.4760	.7140	.9520	1.1900	1.4280	1.6660	[illegible]
11	.2880	.5760	.8639	1.1519	1.4399	1.7279	2.0159	[illegible]
12	.3427	.6854	1.0282	1.3709	1.7136	2.0563	2.3990	[illegible]
13	.4022	.8044	1.2067	1.6089	2.0111	2.4133	2.8155	[illegible]
14	.4665	.9330	1.3994	1.8659	2.3324	2.7989	3.2654	[illegible]
15	.5355	1.0710	1.6065	2.1420	2.6775	3.2130	3.7485	[illegible]
16	.6093	1.2186	1.8278	2.4371	3.0464	3.6557	4.2650	[illegible]
17	.6878	1.2756	1.9635	2.6513	3.3391	4.0269	4.6147	[illegible]
18	.7711	1.5422	2.3134	3.0845	3.8556	4.6267	5.3978	[illegible]
19	.8592	1.7184	2.5775	3.4367	4.2959	5.1551	6.0143	[illegible]
20	.9520	1.9040	2.8560	3.8080	4.7600	5.7120	6.6640	[illegible]
21	1.0496	2.0992	3.1488	4.1983	5.2479	6.2975	7.3471	[illegible]
22	1.1519	2.3038	3.4558	4.6077	5.7596	6.9115	8.0634	[illegible]
23	1.2590	2.5180	3.7771	5.0361	6.2951	7.5541	8.8131	[illegible]
24	1.3709	2.7418	4.1126	5.4835	6.8544	8.2253	9.5962	[illegible]
25	1.4875	2.9750	4.4625	5.9500	7.4375	8.9250	10.413	[illegible]
26	1.6089	3.2178	4.8266	6.4355	8.0444	9.6534	11.262	[illegible]
27	1.7350	3.4700	5.2051	6.9401	8.6751	10.410	12.145	[illegible]
28	1.8659	3.7318	5.5978	7.4637	9.3296	11.196	13.061	[illegible]
29	2.0016	4.0032	6.0047	8.0063	10.008	12.009	14.011	[illegible]
30	2.1420	4.2840	6.4260	8.5680	10.710	12.852	14.994	[illegible]
31	2.2872	4.5744	6.8615	9.1487	11.436	13.723	16.010	[illegible]
32	2.4371	4.8742	7.3114	9.7485	12.186	14.623	17.060	[illegible]
33	2.5918	5.1836	7.7755	10.367	12.959	15.551	18.143	[illegible]
34	2.7513	5.5026	8.2538	11.005	13.756	16.508	19.259	[illegible]
35	2.9155	5.8310	8.7465	11.662	14.578	17.493	20.409	[illegible]
36	3.0845	6.1690	9.2534	12.338	15.422	18.507	21.591	[illegible]
37	3.2582	6.5164	9.7747	13.033	16.291	19.549	22.808	[illegible]
38	3.4367	6.8734	10.310	13.747	17.184	20.620	24.057	[illegible]
39	3.6200	7.2400	10.860	14.480	18.100	21.720	25.340	[illegible]
40	3.8080	7.6160	11.424	15.232	19.040	22.848	26.656	[illegible]
41	4.0008	8.0016	12.002	16.003	20.004	24.005	28.005	[illegible]
42	4.1983	8.3966	12.585	16.783	20.982	25.180	29.378	[illegible]
43	4.4006	8.8012	13.202	17.602	22.003	26.404	30.804	[illegible]
44	4.6077	9.2154	13.823	18.431	23.038	27.646	32.254	[illegible]
45	4.8195	9.6390	14.459	19.278	24.098	28.917	33.737	[illegible]
46	5.0361	10.072	15.108	20.144	25.180	30.216	35.253	[illegible]
47	5.2574	10.515	15.772	21.030	26.287	31.545	36.802	[illegible]
48	5.4835	10.967	16.451	21.934	27.418	32.901	38.385	[illegible]
49	5.7144	11.429	17.143	22.858	28.572	34.286	40.001	[illegible]
50	5.9500	11.900	17.850	23.800	29.750	35.700	41.650	[illegible]
51	6.1904	12.381	18.571	24.762	30.952	37.142	43.333	[illegible]
52	6.4355	12.871	19.307	25.742	32.178	38.613	45.049	[illegible]
53	6.6854	13.371	20.056	26.742	33.427	40.113	46.798	[illegible]
54	6.9401	13.880	20.820	27.760	34.700	41.640	48.581	55.521
55	7.1995	14.399	21.599	28.798	35.998	43.197	[illegible]	[illegible]
56	7.4637	14.927	22.391	29.855	37.318	44.782	[illegible]	[illegible]
57	7.7326	15.465	23.198	30.930	38.663	46.396	54.128	[illegible]
58	[illegible]	[illegible]	24.019	32.025	40.032	48.038	56.044	[illegible]
59	[illegible]	[illegible]	24.854	33.139	41.424	49.709	57.993	[illegible]
60	[illegible]	[illegible]	25.704	34.272	42.840	51.408	59.976	[illegible]

w the Clearance-line on the Indicator-diagram, learance not being known.—The clearance-line may be obtained ely by drawing a straight line, *cbad*, across the compression having drawn *OX* parallel to the atmospheric line and 14.7 lbs. asure from *a* the distance *ad*, equal to *cb*, and draw *YO* perpen-*X* through *d*; then will *TB* divided by *AT* be the percentage of

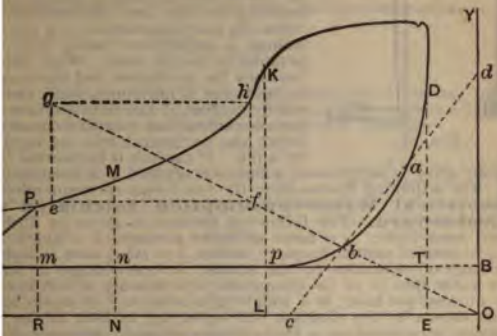


Fig. 139.

The clearance may also be found from the expansion-line by g a rectangle *efhg*, and drawing a diagonal *gf* to intersect the his will give the point *O*, and by erecting a perpendicular to *XO* clearance-line *OY*.

e methods for finding the clearance require that the expansion ession curves be hyperbolas. Prof. Carpenter (*Power*, Sept. nat with good diagrams the methods are usually very accurate, sults which check substantially.

eye Engine Co., however, say that, as the results obtained are ect, being sometimes too little, but more frequently too much, ndications from the two curves seldom agree, the operation has cal value, though when a clearly defined and apparently undis-pression curve exists of sufficient extent to admit of the applica- process, it may be relied on to give much more correct results pansion curve.

w the Hyperbolic Curve on the Indicator-dia- elect any point *I* in the actual curve, and from this point draw a dicular to the line *JB*, meet-er in the point *J*. The line the line of boiler-pressure, ot material; it may be drawn enient height near the top of d parallel to the atmospheric *J* draw a diagonal to *K*, the t being the intersection of and clearance lines; from *I* rallel with the atmospheric *L*, the point of intersection onal *JK* and the horizontal w the vertical line *LM*. The

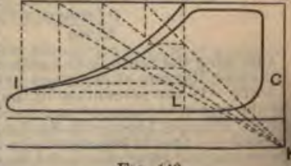


Fig. 140.

the theoretical point of cut-off, and *LM* the cut-off line. Fix *umber of points 1, 2, 3, etc., on the line JB*, and from these points *als to K. From the intersection of these diago*... *tal lines, and from* 1, 2, 3, etc., vertical lines. Wh... *points in the hyperbolic curve.*

n Indicator Rig.—*Power* (Feb. 1893) giv... *of the errors in indicator-diagrams*, caused b...

correct form of the pendulum rigging. It is shown that the "br pulley on the pendulum, to which the cord is attached, does not ally give as good a reduction as a sim attachment. When the end of the pen slotted, working in a pin on the crossh error is apt to be considerable at both the card. With a vertical slot in a pl to the crosshead, and a pin on the p working in this slot, the reduction is when the cord is attached to a pin on dulum, a slight error being introduce brumbo pulley is used. With the co between the pendulum and the crossh by means of a horizontal link, the re nearly perfect, if the construction is the connecting link vibrates equally a below the horizontal, and the cord is by a pin. If the link is horizontal at m a serious error is introduced, which is magnified if a brumbo pul used. The adjoining figures show the two forms recommended.

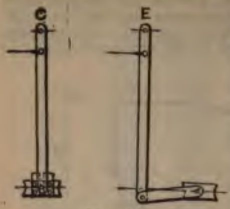


FIG. 141.

Theoretical Water-consumption calculated fr Indicator-card.—The following method is given by Prof. (*Power*, Sept. 1893): p = mean effective pressure, l = length of feet, a = area of piston in square inches, $a \div 144$ = area in squar percentage of clearance to the stroke, b = percentage of strok where water rate is to be computed, n = number of strokes p $60n$ = number per hour, w = weight of a cubic foot of steam havi sure as shown by the diagram corresponding to that at the p water rate is required, w' = that corresponding to pressure at e pression.

Number of cubic feet per stroke $= l\left(\frac{b+c}{100}\right)\frac{a}{144}$.

Corresponding weight of steam per stroke in lbs. $= l\left(\frac{b+c}{100}\right)$

Volume of clearance $= \frac{lca}{14,400}$.

Weight of steam in clearance $= \frac{lcaw'}{14,400}$.

Total weight of steam per stroke $= l\left(\frac{b+c}{100}\right)\frac{wa}{144} - \frac{lcaw'}{14,400} = \frac{la}{14,400}\left[(b+c)w\right.$

Total weight of steam from diagram per hour $= \frac{60nla}{14,400}\left[(b+c)w - cw'\right]$.

The indicated horse-power is $p\,l\,a\,n \div 33,000$. Hence the steam tion per indicated horse-power is

$$= \frac{\frac{60nla}{14,400}\left[(b+c)w - cw'\right]}{\frac{p\,l\,a\,n}{33,000}} = \frac{137.50}{p}\left[(b+c)w - cw'\right].$$

Changing the formula to a rule, we have: To find the water rat indicator diagram at any point in the stroke.

RULE.—To the percentage of the entire stroke which has been by the piston at the point under consideration add the percentag ance. Multiply this result by the weight of a cubic foot of steam pressure of that at the required point. Subtract from this the percentage of clearance multiplied by weight of a cubic foot of st ing a pressure equal to that at the end of the compression. Mu result by 137.50 divided by the mean effective pressure.*

NOTE.—This method only applies to points in the expansion cu een cut-off and release.

* or triple-expansion engines read: divided b pressure, on the supposition that all work

beneficial effect of compression in reducing the water-consumption of ine is clearly shown by the formula. If the compression is carried to point that it produces a pressure equal to that at the point under eration, the weight of steam per cubic foot is equal, and $w = w'$. In ase the effect of clearance entirely disappears, and the formula es $\frac{137.5}{p}(bw)$.

se of no compression, w' becomes zero, and the water-rate =

$$\frac{137.5}{p}[(b + c)w].$$

Denton (Trans. A. S. M. E., xiv. 1363) gives the following table of ical water-consumption for a perfect Mariotte expansion with steam bs. above atmosphere, and 2 lbs. absolute back pressure:

of Expansion, r.	M.E.P., lbs. per sq. in.	Lbs. of Water per hour per horse-power, W.
10	52.4	9.68
15	38.7	8.74
20	30.9	8.20
25	25.9	7.84
30	22.2	7.63
35	19.5	7.45

ifference between the theoretical water-consumption found by the and the actual consumption as found by test represents "water not ed for by the indicator," due to cylinder condensation, leakage ports, radiation, etc.

age of Steam.—Leakage of steam, except in rare instances, has effect upon the lines of the diagram that it can scarcely be detected. y satisfactory way to determine the tightness of an engine is to take not in motion, apply a full boiler-pressure to the valve, placed in a osition, and to the piston as well, which is blocked for the purpose at int away from the end of the stroke, and see by the eye whether occurs. The indicator-cocks provide means for bringing into view hich leaks through the steam-valves, and in most cases that which y the piston, and an opening made in the exhaust-pipe or observa- the atmospheric escape-pipe, are generally sufficient to determine with regard to the exhaust-valves.

eam accounted for by the indicator should be computed for both off and the release points of the diagram. If the expansion-line de- uch from the hyperbolic curve a very different result is shown at it from that shown at the other. In such cases the extent of the asioned by cylinder condensation and leakage is indicated in a much uthful manner at the cut-off than at the release. (Tabor Indicator .)

COMPOUND ENGINES.

pound, Triple- and Quadruple-expansion Engines. pound engine is one having two or more cylinders, and in whi m after doing work in the first or high-pressure cylinder complet nsion in the other cylinder or cylinders.

erm "compound" is commonly restricted, however, to engines e expansion takes place in two stages only—high and low pressu s triple-expansion and quadruple-expansion engines being used wl ansion takes place respectively in three and four stages. The numl ders may be greater than the number of stages of expansion ctive reasons; thus in the compound or two-stage expansion pressure *stage may be effected in two cylinders so as t* *es of nearly equal sizes of* cylinders and of three cranks *triple-expansion engines* there are frequently two *one of them being placed* tandem with the high-pres *the intermediate cylinder*, as in mill engines with *triple-expansion engines of* the steamers *Campania*

Werner, $\sqrt{r}$; and Rankine, $\sqrt[3]{r^2}$, r being the ratio of expansion. Busley makes the ratio dependent on the boiler-pressure thus:

Lbs. per sq. in.	60	90	105	120
$V \div v$ =	3	4	4.5	5

(See Seaton's Manual, p. 95, etc., for analytical method; Sennett, p. 496, etc.; Clark's Steam-engine, p. 445, etc; Clark, Rules, Tables, Data, p. 849, etc.)

Mr. J. McFarlane Gray states that he finds the mean effective pressure in the compound engine reduced to the low-pressure cylinder to be approximately the square root of 6 times the boiler-pressure.

Approximate Horse-power of a Modern Compound Marine-engine. (Seaton.)—The following rule will give approximately the horse-power developed by a compound engine made in accordance with modern marine practice. Estimated H.P. $= \dfrac{D^2 \times \sqrt{p} \times R \times S}{8500}$.

D = diameter of l.p. cylinder; p = boiler-pressure by gauge;
R = revs. per min.; S = stroke of piston in feet.

Ratio of Cylinder Capacity in Compound Marine Engines. (Seaton.)—The low-pressure cylinder is the measure of the power of a compound engine, for so long as the initial steam-pressure and rate of expansion are the same, it signifies very little, so far as total power only is concerned, whether the ratio between the low and high-pressure cylinders is 3 or 4; but as the power developed should be nearly equally divided between the two cylinders, in order to get a good and steady working engine, there is a necessity for exercising a considerable amount of discretion in fixing on the ratio.

In choosing a particular ratio the objects are to divide the power evenly and to avoid as much as possible "drop" and high initial strain.

If increased economy is to be obtained by increased boiler pressures, the rate of expansion should vary with the initial pressure, so that the pressure at which the steam enters the condenser should remain constant. In this case, with the ratio of cylinders constant, the cut-off in the high-pressure cylinder will vary inversely as the initial pressure.

Let R be the ratio of the cylinders; r, the rate of expansion; p_1 the initial pressure; then cut-off in high-pressure cylinder $= R \div r$; r varies with p_1, so that the terminal pressure p_n is constant, and consequently $r = p_1 \div p_n$; therefore, cut-off in high-pressure cylinder $= R \times p_n \div p_1$.

Ratios of Cylinders as Found in Marine Practice.—The rate of expansion may be taken at one tenth of the boiler-pressure (or about one twelfth the absolute pressure), to work economically at full speed. Therefore, when the diameter of the low-pressure cylinder does not exceed 100 inches, and the boiler-pressure 70 lbs., the ratio of the low-pressure to the high-pressure cylinder should be 3.5; for a boiler-pressure of 80 lbs., 3.75; for 90 lbs., 4.0; for 100 lbs., 4.5. If these proportions are adhered to, there will be no need of an expansion-valve to either cylinder. If, however, to avoid "drop," the ratio be reduced, an expansion-valve should be fitted to the high-pressure cylinder.

Where economy of steam is not of first importance, but rather a large power, the ratio of cylinder capacities may with advantage be decreased, so that with a boiler-pressure of 100 lbs. it may be 3.75 to 4.

In tandem engines there is no necessity to divide the work equally. The ratio is generally 4, but when the steam-pressure exceeds 90 lbs. absolute 4.5 is better, and for 100 lbs. 5.0.

When the power requires that the l. p. cylinder shall be more than 100 in. diameter, it should be divided in two cylinders. In this case the ratio of the combined capacity of the two l. p. cylinders to that of the h. p. may be 3 for 80 lbs. absolute, 3.4 for 95 lbs., 3.7 for 105 lbs., and 4.0 for 115 lbs.

Receiver Space in Compound Engines should be from 1 to 1.5 times the capacity of the high-pressure cylinder, when the cranks are at an angle of from 90° to 120°. When the cranks are at 180° or nearly that, the space may be very much reduced. In the case of triple-compound engines, with cranks at 120°, and the intermediate cylinder leading the high pressure, a very small receiver will do. The pressure in the receiver should never exceed half the boiler-pressure. (Seaton.)

›erbolic curve of expansion in the first cylinder, and gh the con-
ısion-line of back pressure
stroke of the first piston,
e pressure for the steam-
cond piston. At the point
f the stroke of the second
am is exhausted into the
l the pressure falls to the
t vacuum, mn.
of the second cylinder,
ıracterized by the absence
period of admission; the
team-line gh being expan-
ted by the expansion of
y of steam contained in
er into the second. When
roke is completed, the
steam transferred from
ıt into the second cylin-
pressure and volume of
ıe second cylinder are the

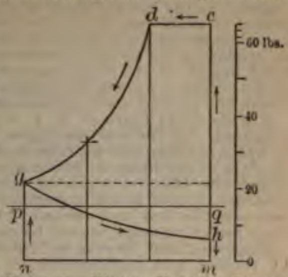
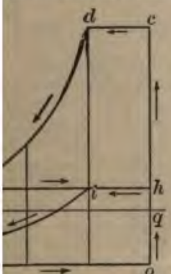


FIG. 142.—WOOLF ENGINE—IDEAL INDICATOR-DIAGRAMS.

whole of the initial steam had been admitted at once into the
r, and then expanded to the end of the stroke in the manner
nder engine.
of the steam is also the same, according to both distributions.
engine, without Clearance–Ideal Diagrams.—
eceiver-engine the pistons of the two cylinders are con-
ıks at right angles to each other on the same shaft. The
the steam exhausted from the first cylinder and supplies it to
which the steam is cut off and then expanded to the end of
n the assumption that the initial pressure in the second cylin-
the final pressure in the first, and of course equal to the pres-
eceiver, the volume cut off in the second cylinder must be
lume of the first cylinder, for the second cylinder must admit
at each stroke as is discharged from the first cylinder.
l is the line of admission and hg the exhaust-line for the first



ıIVER-ENGINE, IDEAL OR-DIAGRAMS.

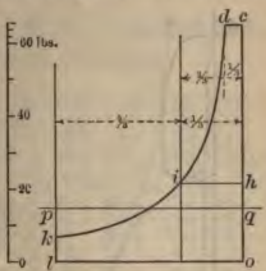


FIG. 144.—RECEIVER ENGINE, IDEAL DIAGRAMS REDUCED AND COMBINED.

dg is the expansion-curve and pq the atmospheric line. In
ıw the exhaust-line of the first cylinder, between it and the
vacuum, ol, the diagram of the second cylinder is formed; hi,
of admission, coincides with the exhaust-line hg of the first
ng in the ideal diagram no intermediate fall of pressure, and
sion-curve. The *arrows indicate the order in which t*
ed.
of the receiver-engine, the expansive working of
divided into two consecutive stages, is, as in
y continuous from the point of cut-off in the f
stroke of the second cylinder, where it is del
e first and second diagrams may be placed

combined to form a continuous diagram. For this purpose take the diagram as the basis of the combined diagram, namely, *hiklo*, Fig. 14 period of admission, *hi*, is one third of the stroke, and as the ratio cylinders are as 1 to 3, *hi* is also the proportional length of the first as applied to the second. Produce *oh* upwards, and set off *oc* equ total height of the first diagram above the vacuum-line; and, u shortened base *hi*, and the height *hc*, complete the first diagram w steam-line *cd*, and the expansion-line *di*.

It is shown by Clark (S. E., p. 432, *et seq.*) in a series of arithmeti culations, that the receiver-engine is an elastic system of compound in which considerable latitude is afforded for adapting the pressure receiver to the demands of the second cylinder, without considerabl ishing the effective work of the engine. In the Woolf engine, contrary, it is of much importance that the intermediate volume o between the first and second cylinders, which is the cause of an i diate fall of pressure, should be reduced to the lowest practicable a

Supposing that there is no loss of steam in passing through the by cooling and condensation, it is obvious that whatever steam through the first cylinder must also find its way through the secon der. By varying, therefore, in the receiver-engine, the period of adi in the second cylinder, and thus also the volume of steam admitted f stroke, the steam will be measured into it at a higher pressure and o bulk, or at a lower pressure and of a greater bulk; the pressure and naturally adjusting themselves to the volume that the steam from ceiver is permitted to occupy in the second cylinder. With a suff restricted admission, the pressure in the receiver may be maintaine pressure of the steam as exhausted from the first cylinder. On t trary, with a wider admission, the pressure in the receiver may "drop" to three fourths or even one half of the pressure of the e steam from the first cylinder.

(For a more complete discussion of the action of steam in the Wo receiver engines, see Clark on the Steam-engine.)

Combined Diagrams of Compound Engines.—The co of making a correct combined diagram from the indicator-diagrams several cylinders in a compound engine is to set off all the diagrams same horizontal scale of volumes, adding the clearances to the cylin

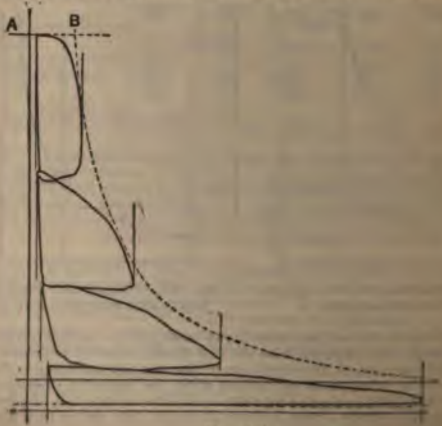


FIG. 145.

ities proper. When this is attended to, the successive diagra ly into their right places relatively to one another, and s operly with any theoretical expansion-curve. (Prof. N. B. W [illegible] M. E., Oct. 1886.)

of combining diagrams is commonly adopted, but there are its accuracy, since the whole quantity of steam consumed in der at the end of the stroke is not carried forward to the part of it is retained in the first cylinder for compression. For ombining diagrams in which compression is taken account of, s by Thomas Mudd and others, in Proc. Inst. M. E., Feb., he usual method of combining diagrams is also criticised by l as inaccurate and misleading (Am. Mach., April 12, 1894; E., xiv. 1465, and xv. 428).

hows a combined diagram of a quadruple-expansion engine, ng to the usual method, that is, the diagrams are first reduced lative scales that correspond with the relative piston-displace- ree cylinders. Then the diagrams are placed at such distances ance-line of the proposed combined diagram as to correctly clearance in each cylinder.

Expansions and Pressures in Two-cylinder d Engines. (James Tribe, *Am. Mach.*, Sept. & Oct. 1891.)

TWO-CYLINDER COMPOUND NON-CONDENSING.

re ½ lb. above atmosphere.

re-... ute	100	110	120	130	140	150	160	170	175
...	115	125	135	145	155	165	175	185	190
on.	7.39	7.84	8.41	9	9.61	10.24	10.89	11.56	11.9
in r..	2.7	2.8	2.9	3	3.10	3.2	3.3	3.4	3.45
1.	1.993	2.029	2.064	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	2.238
gh.	84.8	90.5	96	101.4	106.5	111.5	116.3	120.9	123.2
w..	31.3	32.3	33.1	33.7	34.3	34.8	35.2	35.6	35.7
gh.	42.5	44.6	46.5	48.3	50	51.5	53	54.4	55
w..	15.5	15.5	15.5	15.5	15.5	15.5	15.5	15.5	15.5
gh.	42.3	45.9	49.5	53.1	56.5	60	63.3	66.5	68.2
w..	15.8	16.8	17.6	18.2	18.8	19.3	19.7	20.1	20.2
er ...	2.67	2.73	2.81	2.91	3	3.11	3.21	3.31	3.37

TWO-CYLINDER COMPOUND CONDENSING.

re, 6.5 lbs. above vacuum.

pressures.......	90	100	110	120	130	140	150
e pressures.......	105	115	125	135	145	155	165
ent of loss.......	2.6	2.9	3.3	3.6	3.8	4.0	4.3
ns	15.7	17	18.5	20	21.5	22.7	24.2
cylinder, ...	3.96	4.13	4.3	4.47	4.64	4.77	4.92
1	2.376	2.418	2.458	2.497	2.534	2.562	2.593
High...........	62.9	67.3	71.4	75.4	79.3	83.2	87
Low	15.25	15.55	15.9	16.2	16.5	16.75	17.05
High	26.5	27.8	29	30.2	31.4	32.4	33.5
Low	4.3	4.3	4.3	4.3	4.3	4.3	4.3
gh..............	36.4	39.5	42.4	45.2	47.9	50.8	53.5
w	10.95	11.25	11.6	11.9	12.2	12.45	12.75
gh..............	26.5	27.8	29.0	30.2	31.4	32.4	33.5
w...............	6.4	6.45	6.45	6.5	6.55	6.55	6.6
in l. p. cyl.	25.3	26.6	27.8	29	30.2	31.4	32.4
er areas.......	3.32	3.51	3.66	3.8	3.92	4.08	4.19

percentage of loss, line 3, is thus explained: There is always lue to condensation, and which increases with the pressure of *xact percentage cannot be predetermined, as it depends e quality of the non-conducting covering used on the cylin- d pipes, etc., but will probably be about as shown.*

s of Cylinders in Compound Engines.—Author *the proportions by volume of the high and low pressure . Thus Grashof gives* $V + v = 0.85\sqrt{r}$; Hrabak, 0.90

Werner, $\sqrt{r}$; and Rankine, $\sqrt[3]{r^2}$, r being the ratio of expansi makes the ratio dependent on the boiler-pressure thus:

Lbs. per sq. in........................	60	90	105
$V \div v$.................................. =	3	4	4.5

(See Seaton's Manual, p. 95, etc., for analytical method; Seato etc.; Clark's Steam-engine, p. 445, etc; Clark, Rules, Tables, Data, Mr. J. McFarlane Gray states that he finds the mean effective p the compound engine reduced to the low-pressure cylinder to be mately the square root of 6 times the boiler-pressure.

Approximate Horse-power of a Modern Co Marine-engine. (Seaton.)—The following rule will give appr the horse-power developed by a compound engine made in accord modern marine practice. Estimated H.P. $= \frac{D^2 \times \sqrt{p} \times R \times S}{8500}$.

D = diameter of l.p. cylinder; p = boiler-pressure by gauge; R = revs. per min.; S = stroke of piston in feet.

Ratio of Cylinder Capacity in Compound Mari gines. (Seaton.)—The low-pressure cylinder is the measure of of a compound engine, for so long as the initial steam-pressure a expansion are the same. it signifies very little, so far as total pow concerned, whether the ratio between the low and high-pressure is 3 or 4; but as the power developed should be nearly equally di tween the two cylinders, in order to get a good and steady worki there is a necessity for exercising a considerable amount of dis fixing on the ratio.

In choosing a particular ratio the objects are to divide the pow and to avoid as much as possible "drop" and high initial strain.

If increased economy is to be obtained by increased boiler pres rate of expansion should vary with the initial pressure, so that th at which the steam enters the condenser should remain constan case, with the ratio of cylinders constant, the cut-off in the hig cylinder will vary inversely as the initial pressure.

Let R be the ratio of the cylinders; r, the rate of expansion; p_1 pressure: then cut-off in high-pressure cylinder $= R \div r$; r vari so that the terminal pressure p_n is constant and consequently r = therefore, cut-off in high-pressure cylinder $= R \times p_n \div p_1$.

Ratios of Cylinders as Found in Marine Practi rate of expansion may be taken at one tenth of the boiler-pressure one twelfth the absolute pressure), to work economically at f Therefore, when the diameter of the low-pressure cylinder does n 100 inches, and the boiler-pressure 70 lbs., the ratio of the low-p the high-pressure cylinder should be 3.5; for a boiler-pressure of 8 for 90 lbs., 4.0; for 100 lbs., 4.5. If these proportions are adhered will be no need of an expansion-valve to either cylinder. If, ho avoid "drop," the ratio be reduced, an expansion-valve should be the high-pressure cylinder.

Where economy of steam is not of first importance, but rath power, the ratio of cylinder capacities may with advantage be d so that with a boiler-pressure of 100 lbs. it may be 3.75 to 4.

In tandem engines there is no necessity to divide the work equ ratio is generally 4. but when the steam-pressure exceeds 90 lbs. a is better, and for 100 lbs. 5.0.

When the power requires that the l. p. cylinder shall be more th diameter, it should be divided in two cylinders. In this case the r combined capacity of the two l. p. cylinders to that of the h. p. s for 85 lbs. absolute, 3.4 for 95 lbs., 3.7 for 105 lbs., and 4.0 for 115 lb

Receiver Space in Compound Engines should be 1.5 times the capacity of the high-pressure cylinder, when the cra an angle of from 90° to 120°. When the cranks are at 180° or ne the space may be very much reduced. In the case of triple-comp gines, with cranks at 120°, and the intermediate cylinder leadin pressure, a very small receiver will do. The pressure in the rec never exceed half the boiler-pressure. (Seaton.)

or Calculating the Expansion and the Work of Steam in Compound Engines.

ondensed from Clark on the "Steam-engine.")

he first cylinder in square inches;
he second cylinder in square inches;
he capacity of the second cylinder to that of the first;
f stroke in feet, supposed to be the same for both cylinders;
f admission to the first cylinder in feet, excluding clearance;
e at each end of the cylinders, in parts of the stroke, in feet;
f the stroke plus the clearance, in feet;
f admission plus the clearance, in feet;
f a given part of the stroke of the second cylinder, in feet;
tial pressure in the first cylinder, in lbs. per square inch, sup-
to be uniform during admission;
ssure at the end of the given part of the stroke x;
total pressure for the whole stroke;
ratio of expansion in the first cylinder, or $L \div l$;
atio of expansion in the first cylinder, or $L' \div l'$;
ombined ratio of expansion, in the first and second cylinders
ler;
the final pressure in the first cylinder to any intermediate
pressure between the first and second cylinders;
the volume of the intermediate space in the Woolf engine,
ed up to, and including the clearance of, the second piston,
capacity of the first cylinder plus its clearance. The value
is correctly expressed by the actual ratio of the volumes as
, on the assumption that the intermediate space is a vacuum
it receives the exhaust-steam from the first cylinder. In point
t, there is a residuum of unexhausted steam in the interme-
space, at low pressure, and the value of N is thereby prac-
reduced below the ratio here stated. $N = \frac{n}{n-1} - 1$.

et work in one stroke, in foot-pounds.

ansion in the second cylinder:

In the Woolf engine, $\dfrac{\left(r\frac{L}{L'}\right) + N}{1 + N}$;

In the receiver-engine, $\dfrac{(n-1)r}{n}$.

ratio of expansion = product of the ratios of the three con-
sions, in the first cylinder, in the intermediate space, and
ylinder,

In the Woolf engine, $R'\left(r\frac{L}{L'} + N\right)$;

In the receiver-engine, $r\frac{L'}{l'}$, or rR'.

tio of expansion behind the pistons $= \frac{n-1}{n} rR' = R''$.

n the two cylinders for one stroke, with a given cut-off and a
d actual ratio of expansion:

Woolf engine, $w = aP[l'(1 + \text{hyp log } R'') - c]$;

engine, $w = aP\left[l'(1 + \text{hyp log } R'') - c\left(1 + \frac{r-1}{R'}\right)\right]$.

o intermediate fall of pressure.
an intermediate fall, when the pressure falls to ¾, ⅔, ½ of
e in the 1st cylinder, the reduction of work is 0.2%, 1.0%, 4.
re is no fall.

the mean unbalanced pressure on a square inch of the annular ring of [illegible] large piston obtained by subtracting the intermediate from the large [illegible] We thus see that the mean ordinates of the two lower cards act on [illegible] rings.

Let H = area of small piston in square inches;
I = " " intermediate piston in square inches;
L = " " large piston in square inches;
p_h = mean unbalanced pressure per square inch from card "C";
p_i = " " " " " " " "B";
p_l = " " " " " " " "A";
S = piston-speed in feet per minute;
(I.H.P.) = indicated horse-power of engine.

Then for equal work in each cylinder we have:

$$\text{Area of small piston} = H = 33{,}000 \times \frac{\text{I.H.P.}}{3} \div (p_h \times S); \quad \ldots$$

$$\left.\begin{array}{l}\text{Area of annular ring of}\\ \text{intermediate cylinder}\end{array}\right\} = 33{,}000 \times \frac{\text{I.H.P.}}{3} \div (p_i \times S);$$

$$\left.\begin{array}{l}\text{Area of interme-}\\ \text{diate piston}\end{array}\right\} = I = H + 33{,}000 \times \frac{\text{I.H.P.}}{3} \div (p_l \times S); \quad \ldots$$

$$\text{Area of annular ring of large piston} = 33{,}000 \times \frac{\text{I.H.P.}}{3} \div (p \times [\text{illegible}]$$

$$\text{Area of large piston} = L = I + 33{,}000 \times \frac{\text{I.H.P.}}{3} \div (p_l \times S); \ldots$$

This method is illustrated by the following example: Given I.H.P. = [illegible] piston-speed S = 900 ft. per min., ratio of expansion 10, initial steam-[illegible] sure at cylinder 127 lbs. absolute, and back-pressure in large cylinder [illegible] absolute. Find cylinder diameters for equal work in each.

The mean ordinate of "C" is found to be p_h = 37.414 lbs. per sq. in.
" " " " "B" " " " p_i = 15.782 " "
" " " " "A" " " " p_l = 11.730 " " "

Then by (1), (2), and (3) we have:

$$H = 33{,}000 \times \frac{3000}{3} \div 37.414 \times 900 = 980 \text{ sq. in., diam. } 35\tfrac{3}{8}'';$$

$$I = 980 + 33{,}000 \times \frac{3000}{3} \div 15.782 \times 900 = 3303 \text{ sq. in., diam. } 60'';$$

$$L = 3303 + 33{,}000 \times \frac{3000}{3} \div 11.730 \times 900 = 6432 \text{ sq. in., diam. } 90[\text{illegible}]$$

Mr. Whitham recommends the following cylinder ratios when the pis[illegible] speed is from 750 to 1000 ft. per min., the terminal pressure in the la[illegible] cylinder being about 10 lbs. absolute.

CYLINDER RATIOS RECOMMENDED FOR TRIPLE-EXPANSION ENGINES.

Boiler-pressure (Gauge).	Small.	Intermediate.	Large.
130	1	2.25	[illegible]
140	1	2.40	[illegible]
150	1	2.55	[illegible]
160	1	2.70	[illegible]
170 and upwards—quadruple-expansion engine to be used.			

He gives the following ratios from examination of a number of [illegible] engines:

No. of Engines Averaged.	Steam-boiler Pressure.	Cylinder Ratios. h. p.	int.	l. p.
9	130	1	2.10	[illegible]
8	135	1	2.07	[illegible]
11	140	1	2.40	[illegible]
2	145	1	2.35	[illegible]
28	150	1	2.54	[illegible]
27	160	1	2.66	[illegible]

mon Rule for Proportioning the Cylinders of mul-
sion engines is: for two-cylinder compound engines, the cylinder square root of the number of expansions, and for triple-expansion ratios of the high to the intermediate and of the intermediate are each equal to the cube root of the number of expansions, the high to the low being the product of the two ratios, that is, the he cube root of the number of expansions. Applying this rule to es above given, assuming a terminal pressure (absolute) of 10 lbs. espectively, we have, for triple-expansion engines:

Terminal Pressure, 10 lbs.		Terminal Pressure, 8 lbs.	
No. of Expansions.	Cylinder Ratios, areas.	No. of Expansions.	Cylinder Ratios, areas.
13	1 to 2.35 to 5.53	16¼	1 to 2.53 to 6.42
14	1 to 2.41 to 5.81	17½	1 to 2.60 to 6.74
15	1 to 2.47 to 6.08	18¾	1 to 2.66 to 7.06
16	1 to 2.52 to 6.35	20	1 to 2.71 to 7.37

of the diameters is the square root of the ratios of the areas, and f the diameters of the first and third cylinders is the same as the e areas of first and second.

n his Marine Engineering, says: When the pressure of steam em-eeds 115 lbs. absolute, it is advisable to employ three cylinders, ich of which the steam expands in turn. The ratio of the low-high-pressure cylinder in this system should be 5, when the sure is 125 lbs. absolute; when 135 lbs. absolute, 5.4; when 145 te, 5.8; when 155 lbs. absolute, 6.2; when 165 lbs. absolute, 6.6. f low-pressure to intermediate cylinder should be about one half en low-pressure and high-pressure, as given above. That is, if f l. p. to h. p. is 6, that of l. p. to int. should be about 3, and conse-at of int. to h. p. about 2. In practice the ratio of int. to h. p. is , so that the diameter of the int. cylinder is 1.5 that of the h. p. uction of the triple-compound engine has admitted of ships being at higher rates of speed than formerly obtained without exceeding mption of fuel of similar ships fitted with ordinary compound n such cases the higher power to obtain the speed has been devel-creasing the rate of expansion, the low-pressure cylinder being es the capacity of the high-pressure, with a working pressure of solute. It is now a very general practice to make the diameter of essure cylinder equal to the sum of the diameters of the h. p. and ers; hence,

iameter of int. cylinder = 1.5 diameter of h. p. cylinder;
iameter of l. p. cylinder = 2.5 diameter of h. p. cylinder.

ase the ratio of l. p. to h. p. is 6.25; the ratio of int. to h. p. is 2.25; of l. p. to int. is 2.78.

of Cylinders for Different Classes of Engines.
. M. E., Feb. 1887, p. 36.)—As to the best ratios for the cylinders engine there seems to be great difference of opinion. Considera-e, however, is due to the requirements of the case, inasmuch as ot be expected that the same ratio would be suitable for an eco-and engine, where the space occupied and the weight were of ortance, as in a war-ship, where the conditions were reversed. In engine, for example, a theoretical terminal pressure of about 7 absolute vacuum would probably be aimed at, which would give capacity of high pressure to low pressure of 1 to 8⅛ or 1 to n a war-ship a terminal pressure would be required of 12 to 13 lbs. ld need a ratio of capacity of 1 to 5; yet in both these instances ers were correctly proportioned and suitable to the requirements e. It is *obviously unwise, therefore*, to introduce any hard-and-

of Three-stage Expansion Engines.—1. Three cranks 2. *Two cranks with 1st and 2d cylinders tandem.* 3. *Two 1st and 3d cylinders tandem.* The most common type is *inders arranged in the sequence high, intermediate, low.*

Sequence of Cranks.—Mr. Wyllie (Proc. Inst. M. E., 1887) fa sequence high, low, intermediate, while Mr. Mudd favors high, inter low. The former sequence, high, low, intermediate, gave an approx horizontal exhaust-line, and thus minimizes the range of temperat the initial load; the latter sequence, high, intermediate, low, incre range and also the load.

Mr. Morrison, in discussing the question of sequence of cranks, p a diagram showing that with the cranks arranged in the sequen low, intermediate, the mean compression into the receiver was 19⅛ of the stroke; with the sequence high, intermediate, low, it was 57 In the former case the compression was just what was required the receiver-pressure practically uniform; in the latter case the com caused a variation in the receiver-pressure to the extent somew 22¼ lbs.

Velocity of Steam through Passages in Com Engines. (Proc. Inst. M. E., Feb. 1887.)—In the SS. *Para*, taking of the cylinder multiplied by the piston-speed in feet per sec dividing by the area of the port the velocity of the initial steam the high-pressure cylinder port would be about 100 feet per second; haust would be about 90. In the intermediate cylinder the initia had a velocity of about 180, and the exhaust of 120. In the low-p cylinder the initial steam entered through the port with a velocity and in the exhaust-port the velocity was about 140 feet per second.

QUADRUPLE-EXPANSION ENGINES.

H. H. Suplee (Trans. A. S. M. E., x. 583) states that a study of 14 quadruple-expansion engines, nearly all intended to be operated at sure of 180 lbs. per sq. in., gave average cylinder ratios of 1 to 2.3 7.70, or nearly in the proportions 1, 2, 4, 8.

If we take the ratio of areas of any two adjoining cylinders as th root of the number of expansions, the ratio of the 1st to the 4th w cube of the fourth root. On this basis the ratios of areas for differ sures and rates of expansion will be as follows:

Gauge-pressures.	Absolute Pressures.	Terminal Pressures.	Ratio of Expansion.	Ratios of of Cyli
160	175	12	14.6	1 : 1.90 : 3.
		10	17.5	1 : 2.05 : 4.
		8	21.9	1 : 2.16 : 4.
180	195	12	16.2	1 : 2.01 : 4.
		10	19.5	1 : 2.10 : 4.
		8	24.4	1 : 2.22 : 4.
200	215	12	17.9	1 : 2.06 : 4.
		10	21.5	1 : 2.15 : 4.
		8	26.9	1 : 2.28 : 5.
220	235	12	19.6	1 : 2.10 : 4.
		10	23.5	1 : 2.20 : 4.
		8	29.4	1 : 2.31 : 5.

Seaton says: When the pressure of steam employed exceeds 190 lb lute, four cylinders should be employed, with the steam expanding each successively; and the ratio of l. p. to h. p. should be at least if economy of fuel is of prime consideration it should be 8; then t of first intermediate to h. p. should be 1.8, that of second interme first int. 2, and that of l. p. to second int. 2.2.

In a paper read before the North East Coast Institution of Engine Shipbuilders, 1890, William Russell Cummins advocates the use of cylinder engine with four cranks as being more suitable for high than the three-cylinder three-crank engine. The cylinder ratios, he should be designed so as to obtain equal initial loads in each cylinde ratios determined for the triple engine are 1, 2.04, 6.54, and for the ple 1, 2.08, 4.46, 10.47. He advocates long stroke, high piston-spee olutions per minute, and 250 lbs. boiler-pressure, unjacketed cy separate steam and exhaust valves.

rs of Cylinders of Recent Triple-expansion Engines, Chiefly Marine.

Compiled from several sources, 1890-1893.

hes: H = high pressure, I = intermediate, L = low pressure.

L	*H*	*I*	*L*	*H*	*I*	*L*	*H*	*I*	*L*
8	16	25.6	41	22	36	40	36	58	94
13	16¼	23⅞	38.5			40	38	61.5	100
12	16.5	24.5	31	23	38	61	28	56	86
16.5			31	23.5	38	60	28		
12.5	17	27	44	24	37	56	39	61	97
18.9	17	26.5	42	25	40	64	40	59	88
19	17	28	45	26	42	69	40	67	106
16	18	27	40	26	42.5	70	40	66	100
22.5	18	29	48	28	44	72	41	66	101
25.6	18	305.	51	29⅜	44	70	41⅝	67	106¾
25	18.7	29.5	43.3	29.5	48	78	42	59	92
24	18¾	23.6	35.4	30	48	77	43	66	92
25	19.7	29.6	47.3	32	46	70	43	68	110
30	20	30	45	32	51	82	43⅝	67	106¼
28.5	20	32.5	36	32	54	82	45	71	113
30.5			36	33	58	88	32.5	68	85.7
30.7	20	33	52	33.9	55.1	84.6	32.5		85.7
33.5	21	32	48	34	54	85	47	75	81.5
36	21	36	51	34	50	90			81.5
39	21.7	33.5	49.2	34.5	51	85	37	79	98
39	21.9	34	57	34.5	57	92	37		98
38	22	34	51						

igures are bracketed there are two cylinders of a kind. Two
″, two 31″ = one 43.8″, two 32.5″ = one 46.0″, two 36″ = one
= one 52.3″, two 40″ = one 56.6″, two 81.5″ = one 115″, two
1″, two 98″ = one 140″. The average ratio of diameters of
ll the engines in the above table is nearly 1 to 1.60 to 2.56 and
eas nearly 1 to 2.56 to 6.55.

ress in Steam-engines between 1876 and 1893 is shown
g comparison of the Corliss engine at the Centennial Exhibi-
d the Allis-Corliss quadruple-expansion engine at the Chicago

	1893.	1876.
....	Quadruple-expansion.	Simple
nber........................	4	2
meter.......................	24, 40, 60, 70 in.	40 in.
oke.........................	72 in.	120 in.
ameter	30 ft.	30 ft.
dth of face.................	76 in.	24 in.
ight........................	136,000 lbs.	125,440 lbs.
er minute...................	60	36
nomical.....................	2000 H.P.	1400 H.P.
ximum.......................	3000 H.P.	2500 H.P.
....	650,000 lbs.	1,360,[illegible]88 lbs.

haft body or wheel-seat of the Allis engine has a diameter of
rnals 19 inches, and crank bearings 18 inches, with a total
et. The crank-disks are of cast iron and are 8 feet in diam-
nk-pins are 9 inches in diameter by 9 inches long.

-tandem Triple-expansion Engine, built by Watts,
o., Newark, N. J., is described in *Am. Mach.*, April 26, 1894.
-cylinder tandem engines coupled to one shaft, cranks at 90°,
2 and 48 by 60 in. stroke, 65 revolutions per minute, rated H.P.
28 feet diameter, 12 ft. face, weight 174,000 lbs.; main shaft
at the swell; main journals 19 × 38 in.; crank-pins 9½ × 10
tween centre lines of two engines 24 ft. 1½ in.; Corliss
rate eccentrics for the exhaust-valves of the l.p. cylinder

Principal Engines in the Power-Plant at the World's Columbian Exposition, 1893.

Name of Engine and where Built.	Type of Engine.	Horizontal or Vertical.	Cylinders, ins. Diameters and Stroke.	I.H.P. Maximum Economy.	I.H.P. Maximum Load.	Driv. Pulley, Diameter in. Face, in.	Revolutions per Minute.	Size of Steam-pipe.	Size of Exhaust-pipe.	Weight of Engine, lbs.
E. P. Allis Co., Milwaukee	Quad. exp. condensing	H.	24, 40, 60, 70 x 72	2,000	3,000	360 x 76	60	10	22	650,000
Fraser & Chalmers, Chicago	4-cyl. trip. exp. cond	"	20, 34, 2-34 x 60	1,000	1,250	336 x 68	64 1-2	7	16	260,000
McIntosh-Seymour, Auburn, N. Y	Double tand. comp. cond.	"	18, 32 x 36 double	1,000	1,500	192 x 78	112	7	14	250,000
Buckeye Engine Co., Salem, O	4-cyl. trip. exp. cond	"	20, 30, 2-36 x 48	1,000	1,300	240 x 75	85	8	12	120,000
Atlas, Indianapolis, Ind	Comp. condensing	"	14, 24 x 30	1,000	1,300	144 x 75	150	5	8	114,000
Westinghouse, Pittsburg, Pa.	Comp. cond., 6 engines	V.	21 1-2, 37 x 22	1,000	......	132 x 14	200	10	14	195,000*
" " "	" " 2 "	"	18, 30 x 16	350	......	96 x 26	250	8	10	......
Russell, Massillon, O.	Double tand. comp. cond.	H.	15, 24 x 24	506	897	120 x 60	125	7	12	83,298
Atlas, Indianapolis, Ind	Tandem comp. cond	"	14, 24 x 30	500	650	144 x 36	150	5	8	57,000
Bass, Ft. Wayne, Ind	Cross compound cond	"	16, 30 x 42	224	440	192 x 43	70	5	10	61,000
Lane & Bodley, Cincinnati, O.	" " "	"	16, 30 x 42	300	350	192 x 43	70	5	10	80,000
" " " "	Tandem comp. cond	"	16, 28 x 42	300	350	216 x 36	80	5	10	75,000
Watertown, Watertown, N. Y.	Double tand. comp. cond.	"	9, 16 x 14	250	......	60 x 12	250	4 1-2	10	21,200
Buckeye, Salem, O.	Cross compound cond	"	14, 28 x 24	300	360	120 x 33	50	......	......	33,000
Ball & Wood, Elizabeth, N. J.	Cross compound	"	14, 22 x 12	200	250	66	280	5	8	......
N. Y. Safety Steam-Power Co.	Simple	"	15 1-2 x 16	150	200	72 x 12	250	4 1-2	6	......
Schichau, Germany	Triple exp. condensing	V.	23, 37, 57 x 27	1,000	......	......	100	......	......	......
Stearns Mfg. Co., Erie, Pa	Tandem comp. cond	H.	19, 31 x 24	600	800	102 x 31	165	7	12	75,000
Phoenix Iron Works Co., Meadville, Pa.	4-cyl. trip. exp. cond	"	15, 24, 2-26 x 18	500	......	2-108 x 26 / 1-108 x 38	200	6	14	......
" " " "	Tandem compound cond.	"	13, 24 x 18	250	......	108 x 26	200	5	9	......
A. L. Ide & Son, Springfield, Ill.	" " "	"	13, 22 x 16	225	250	72 x 16 1-2	245	6	8	24,000
Allis, Milwaukee, Wis.	Cross compound cond	"	19, 36 x 48	500	800	192	90	7	12	85,000
General Electric Co., N. Y.	Triple exp. condensing	V.	......	1,000	......	......	......	8	18	......
Armington & Sims, Providence, R. I.	Simple	H.	18 x 21	......	......	84 x 25	225	7	10	......
Ball Engine Co., Erie, Pa	Cross compound	"	18, 36 x 18	480	570	86	225	7	18	65,000
McEwen, Ridgway, Pa.	Tandem compound cond.	"	14, 23 x 20	220	225	84 x 29	200	5	10	24,000
[illegible] Sioux City, Ia	Simple	"	24 x 48	330	425	192 x 44	......	8	9	49,000
[illegible] Iron Wks., San Fran'co, Cal	Tandem compound	"	12, 22 x 30	200	270	132 x 25	115	4	7 1-2	35,000
[illegible] Machine Co., Joliet, Ill	Simple	"	20 x 48	297	346	192 x 27	60	6	7	43,000
[illegible] Harrisburg, Pa.	Tandem compound	"	17, 28 x 18	300	370	[illegible]	176	6	11	[illegible]
[illegible] Steam-Engine Works, R. I.	"	"	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

CONOMIC PERFORMANCE OF STEAM-ENGINES.

conomy of Expansive Working under Various Conditions, Single Cylinder.

(Abridged from Clark on the Steam Engine.)

SINGLE CYLINDERS WITH SUPERHEATED STEAM, NONCONDENSING.—In- cylinder locomotive, cylinders and steam-pipes enveloped by the hot s in the smoke-box. Net boiler pressure 100 lbs.; net maximum press- in cylinders 80 lbs. per sq. in.

-off, per cent.........	20	25	30	35	40	50	60	70	80
nal ratio of expansion	3.91	3.31	2.87	2.53	2.26	1.86	1.59	1.39	1.23
ter per I.H.P. per hour, lbs....................	18.5	19.4	20	21.2	22.2	24.5	27	30	33

SINGLE CYLINDERS WITH SUPERHEATED STEAM, CONDENSING.—The best lts obtained by Hirn, with a cylinder 23¾ × 67 in. and steam super- ted 150° F., expansion ratio 3¾ to 4½, total maximum pressure in cylin- 63 to 69 lbs. were 15.63 and 15.69 lbs. of water per I.H.P. per hour.

SINGLE CYLINDERS OF SMALL SIZE, 8 OR 9 IN. DIAM., JACKETED, NON- DENSING.—The best results are obtained at a cut-off of 20 per cent, with s. maximum pressure in the cylinder; about 25 lbs. of water per I.H.P. hour.

SINGLE CYLINDERS, NOT STEAM-JACKETED, CONDENSING.—Best results.

Engine.	Cylinder, Diam. and Stroke.	Cut-off.	Actual Expansion Ratio.	Total Maximum Pressure in Cylinder per sq. in.	Water as Steam per I.H.P. per hour.
	ins.	per cent.	ratio.	lbs.	lbs.
iss and Wheelock...	18 × 48	12.5	6.95	104.4	19.58
n, No. 6...............	23¾ × 67	16.3	5.84	61.5	19.93
r, M.......	32 × 66	24.6	3.84	54.5	26.46
he....................	25 × 24	15.5	5.32	87.7	26.25
ter....................	26 × 36	18.3	4.46	80.4	23.86
as.....................	36 × 30	13.3	5.07	46.9	26.69
atin..................	30.1 × 30	15.0	4.94	81.7	21.89

SAME ENGINES, AVERAGE RESULTS.

Long Stroke.	Inches.	Cut-off, Per cent.	Lbs.	Lbs.
iss and Wheelock...	18 × 48	12.5	104.4	19.58
n	23¾ × 67	16.3	61.5	19.93
Short Stroke.				
he....................	25 × 24	15.5	87.7	26.25
ter, Nos. 20, 21, 22, 23	26 × 36	18.3 to 33.3 average 25	79.0	24.05
as, Nos. 27, 28, 29....	36 × 30	13.3 to 26.4 average 19.8	46.8	26.86
atin, Nos, 24, 25, 22, 26	30.1 × 30	12.3 to 18.5 average 15.8	78.2	23.50

eed-water Consumption of Different Types of Engines.

e following tables are taken from the circular of the *Tabor Indicator* ncroft Mfg. Co., 1889). In the first of the two columns under Feed-water uired, in the tables for simple engines, the figures are obtained by putation from nearly perfect indicator diagrams, with allowance for cyl- er condensation *according to the table* on page 752, but without allow- *e for leakage, with back-pressure* in the non-condensing table taken at 16 *above zero, and in the condensing* table at 3 lbs. above zero. The co- *ion curve is supposed* to be hyperbolic, and commences at 0.91 of *-stroke, with a clearance of 3%* of the piston-displacement. *No. 2 gives the feed-water* consumption for jacketed compoun densing engines of the best class. The water condensed in t included in the quantities given. The ratio of areas of the two as 1 to 4 for 120 lbs. pressure; the clearance of each cylinder i cut off in the two cylinders occurs at the same point of stroke pressure in the l. p. cylinder is 1 lb. per sq. in. below the back-pr h. p. cylinder. The average back pressure of the whole strok cylinder is 4.5 lbs. for 10% cut-off; 4.75 lbs. for 20% cut-off; and cut-off. The steam accounted for by the indicator at cut-off cylinder (allowing a small amount for leakage) is .74 at 10% c 20%, and .82 at 30% cut-off. The loss by condensation between t is such that the steam accounted for at cut-off in the l. p. c pressed in proportion of that shown at release in the h. p. cylin 10% cut-off, .87 at 20% cut-off, and .89 at 30% cut-off.

The data upon which table No. 3 is calculated are not given, b water consumption is somewhat lower than has yet been reache lowest steam consumption of a triple-exp. engine yet recorded be

TABLE No. 1.

FEED-WATER CONSUMPTION, SIMPLE ENGINES.

NON-CONDENSING ENGINES.

Per Cent Cut-off.	Initial Pressure above Atmosphere, lbs.	Mean Effective Pressure, lbs.	Feed-water Required per I.H.P. per Hour. Corresponding to Diagrams with no Leakage, lbs.	Corresponding to Actual Results Attained in Practice, assuming Slight Leakage.
10	60	8.70	37.26	40.95
	70	12.39	30.99	33.68
	80	16.07	27.61	29.88
	90	19.76	25.43	27.43
	100	23.45	23.90	25.73
20	60	21.12	27.55	29.43
	70	26.57	25.44	27.04
	80	32.02	21.04	25.68
	90	37.47	23.00	24.57
	100	42.92	22.25	23.77
30	60	30.47	27.24	29.10
	70	37.21	25.76	27.43
	30	43.97	24.71	26.29
	90	50.73	23.91	25.38
	100	57.49	23.27	24.68
40	60	37.75	27.92	29.63
	70	45.50	26.66	28.18
	80	53.25	25.76	27.17
	90	61.01	25.03	26.35
	100	68.76	24.47	25.73
50	60	43.42	28.94	30.66
	70	51.94	27.79	29.31
	80	60.44	26.99	28.38
	90	68.96	26.32	27.62
	100	77.48	25.78	26.99

CONDENSING ENGI

Per Cent Cut-off.	Initial Pressure above Atmosphere, lbs.	Mean Effective Pressure, lbs.	Feed-water Required per I.H.P. per Hour. Corresponding to Diagrams with no Leakage, lbs.
5	60	14.42	18.22
	70	16.96	17.96
	80	19.50	17.74
	90	22.04	17.57
	100	24.58	17.41
10	60	22.34	17.68
	70	26.03	17.47
	80	29.72	17.30
	90	33.41	17.15
	100	37.10	17.02
15	60	29.00	17.91
	70	33.65	17.73
	80	38.28	17.60
	90	42.92	17.45
	100	47.56	17.34
20	60	34.78	18.58
	70	40.18	18.40
	80	45.63	18.27
	90	51.08	18.14
	100	56.53	18.02
30	60	44.06	20.19
	70	50.81	20.04
	80	57.57	19.91
	90	64.32	19.78
	100	71.08	19.67
40	60	[illegible]	[illegible]
	70	[illegible]	[illegible]
	80	[illegible]	[illegible]
	90	[illegible]	[illegible]
	100	[illegible]	[illegible]

TABLE No. 2.

MPTION FOR COMPOUND CONDENSING ENGINES.

sure above here.	Mean Effective Pressure above Atmosphere.		Feed-water Required per I.H.P. per Hour, Lbs.
L.P. Cyl., lbs.	H.P. Cyl., lbs.	L.P. Cyl., lbs.	
4.0	11.67	2.65	16.92
7.3	15.33	3.87	15.00
11.0	18.54	5.23	13.86
4.3	26.73	5.48	14.60
8.1	33.13	7.56	13.67
12.1	39.29	9.74	13.09
4.6	37.61	7.48	14.99
8.5	46.41	10.10	14.21
11.7	56.00	12.26	13.87

TABLE No. 3.

TION FOR TRIPLE-EXPANSION CONDENSING ENGINES.

sure above phere.		Mean Effective Pressure.			Feed-water Required per I.H.P. per Hour, lbs.
yl., bs.	L.P. Cyl., lbs.	H.P. Cyl., lbs.	I. Cyl., lbs.	L.P. Cyl., lbs.	
7.8	1.3	38.5	17.1	6.5	12.05
3.8	2.8	46.5	18.6	7.1	11.4
49.3	3.8	55.0	20.0	8.0	10.75
38.8	2.8	51.5	22.8	8.6	11.65
45.8	3.9	59.5	23.7	9.1	11.4
51.3	5.3	70.0	25.5	10.0	10.85
39.8	3.7	60.5	26.7	10.1	12.2
46.8	4.8	70.5	28.0	10.8	11.6
52.8	6.3	82.5	30.0	11.8	11.15

mical Point of Cut-off in Steam-engines. ff and Denton, Trans. A. S. M. E., vol. ii. p. 147-281; also, n at Maximum Efficiency, R. H. Thurston, vol. ii. p. 128.) the best ratio of expansion is not one of economy of con- and economy of cost of boiler alone. The question of engine, depreciation of value of engine, repairs of engine, ll; for as we increase the rate of expansion, and thus, its fixed by the back-pressure and condensation of steam, unt of fuel required and cost of boiler per unit of work, se the dimensions of the cylinder and the size of the en- required power. We thus increase the cost of the engine, se the rate of expansion, while at the same time we de- consumption, the cost of boiler, etc. So that there is in e point of cut-off, determinable by calculation and graphi- which will secure the greatest efficiency for a given expen- taking into consideration the cost of fuel, wages of engineer rest on cost, depreciation of value, repairs to and insurance ne, and oil, waste, etc., used for engine. In case of freight- the value of the room occupied by fuel should be consid- g the cost of fuel.

Calculated Performances of Vertical High- s.—*The following tables are taken from a circular of the Co., New York, describing the engines made by the Lake Works, Buffalo, N. Y. The engines are fair representatives ning largely into use for driving dynamos directly with- bles were calculated by E. F. Williams, designer of the here somewhat abridged to save space:*

Simple Engines—Non-condensing.

Diam. of Cylinder, inches.	Stroke, inches.	Revs. per Minute.	H.P. when Cutting off at 1/5 stroke.			H.P. when Cutting off at ¼ stroke.			H.P. when Cutting off at ⅓ stroke.			Dimensions of Wheels. diam.	face	Steam-pipe, in.	Exhaust-pipe.
			70 lbs.	80 lbs.	90 lbs.	70 lbs.	80 lbs.	90 lbs.	70 lbs.	80 lbs.	90 lbs.	Ft.	In.		
7½	10	370	20	25	30	26	31	36	32	37	43	4	4	2½	3
8½	12	318	27	32	39	34	41	47	41	48	56	4½	5	2¾	3½
10½	14	277	41	49	60	52	62	71	63	74	85	5′9″	6½	3½	4
12	16	246	53	64	77	67	81	93	82	96	111	6′8″	9	4	4½
13½	18	222	66	80	96	84	100	116	102	120	138	7½	11	4	5
16	20	181	95	115	138	120	144	166	146	172	198	8′4″	15	4½	6
18	24	158	119	144	173	151	181	208	183	215	248	10	19	5	7
22	28	138	179	216	261	227	272	313	276	324	373	11′8″	28	6	8
24½	32	120	221	267	322	281	336	386	340	400	460	13′4″	34	7	9
27	34	112	269	325	392	342	409	470	414	487	560	14′2″	41	8	10
Mean eff. press. lb.			24	29	35	30.5	36.5	42	37	43.5	50				
Ratio of expans'n.			5			4			3						
Terminal pressure (about)......lbs.			17.9	20	22.3	22.4	25	27.6	29.8	33.3	36.8				
Cyl. condensat'n, %			26	26	26	24	24	24	21	21	21				
Steam per I.H.P. per hour......lbs.			32.9	30	27.4	31.2	29.0	27.9	32	31.4	30				

NOTE.—The nominal-power rating of the engines is at 80 lbs. gauge pressure, steam cut-off at ¼ stroke.

Compound Engines — Non-condensing — High-pressure Cylinder and Receiver Jacketed.

Diam. Cylinder, inches.			Stroke, inches.	Revolutions per Minute.	H.P. when cutting off at ¼ Stroke in h.p. Cylinder. Cyl. Ratio, 3⅛ : 1.		H.P. when cutting off at ¼ Stroke in h.p. Cylinder. Cyl. Ratio, 4½ : 1.		H.P. when cutting off at ⅓ Stroke in h.p. Cylinder. Cyl. Ratio, 3⅛ : 1.		H.P. when cutting off at ⅓ Stroke in h.p. Cylinder. Cyl. Ratio, 4½ : 1.		H.P. when cutting off at ½ Stroke in h.p. Cylinder. Cyl. Ratio, 3⅛ : 1.		H.P. when cutting off at ½ Stroke in h.p. Cylinder. Cyl. Ratio, 4½ : 1.	
H.P.	H.P.	L.P.			80 lbs.	90 lbs.	130 lbs.	150 lbs.	80 lbs.	90 lbs.	130 lbs.	150 lbs.	80 lbs.	90 lbs.	130 lbs.	150 lbs.
5¾	6½	12	10	370	7	15	19	32	23	31	35	46	44	55	64	[illegible]
6⅝	7½	13½	12	318	9	19	24	40	29	39	45	59	56	70	81	[illegible]
7¾	9	16½	14	277	14	28	36	60	43	58	67	87	83	104	121	[illegible]
9	10½	19	16	246	18	37	47	78	57	76	87	114	109	136	158	[illegible]
10½	12	22½	18	222	26	53	68	112	81	109	125	164	156	195	[illegible]	[illegible]
12	13½	25	20	185	32	65	84	139	100	135	154	202	192	241	[illegible]	[illegible]
13½	15½	28½	24	158	43	88	112	186	135	181	206	271	258	323	374	[illegible]
16	18½	33½	28	138	57	118	151	249	180	242	277	363	346	433	[illegible]	[illegible]
18	20½	38	32	120	74	152	194	321	232	312	357	468	446	558	647	[illegible]
20	22½	43	34	112	94	194	249	412	297	400	457	601	572	715	[illegible]	[illegible]
24½	28½	52	42	93	138	285	365	603	436	587	670	880	838	1048	[illegible]	[illegible]
28½	33	60	48	80	180	374	477	789	570	767	877	1151	1096	1370	[illegible]	[illegible]
Mean effec. press...lbs					3.3	6.8	8.7	14.4	10.4	14.0	16	21	20	25	[illegible]	[illegible]
Ratio of expansion....					13½		18¼		10¼		13¾		6¾		[illegible]	
Cyl. condensation, %...					14	14	16	16	12	12	13	13	10	10	11	[illegible]
Ter. press. (about). lbs.					7.3	7.7	7.9	9	9.2	10.4	10.5	12	14	15.5	[illegible]	[illegible]
Loss from expanding below atmosphere, %					34	15	17	3	5	0	0	0	0	0	0	[illegible]
St. per I.H.P. p. hr. lbs.					55	42	47	29	33.3	27.7	28.7	25.4	30	26.2	[illegible]	[illegible]

The original table contains figures of horse-power, etc., for 110 and 130 [illegible] cylinder ratio of 4 to 1; and 140 lbs., ratio 4½ to 1.

ınd-engines—Condensing—Steam-jacketed.

troke, inches.	Revolutions per Minute.	H.P. when cutting off at ¼ Stroke in h.p. Cylinder.				H.P. when cutting off at ⅓ Stroke in h.p. Cylinder.				H.P. when cutting off at ⅙ Stroke in h.p. Cylinder.			
		Cyl. Ratio, 3⅓ : 1.		Cyl. Ratio, 4 : 1.		Cyl. Ratio, 3⅓ : 1.		Cyl. Ratio, 4 : 1.		Cyl. Ratio, 3⅓ : 1.		Cyl. Ratio, 4 : 1.	
		80 lbs.	110 lbs.	115 lbs.	125 lbs.	80 lbs.	110 lbs.	115 lbs.	125 lbs.	80 lbs.	110 lbs.	115 lbs.	125 lbs.
0	370	44	59	53	62	55	70	68	75	70	97	95	106
2	318	56	76	67	78	70	90	87	95	90	123	120	134
4	277	83	112	100	116	104	133	129	141	133	183	179	200
6	246	109	147	131	152	136	174	169	185	174	239	234	261
8	222	156	210	187	218	195	250	242	265	250	343	335	374
0	185	192	260	231	269	241	308	298	327	308	423	414	462
4	158	258	348	310	361	323	413	400	439	413	568	555	619
8	138	346	467	415	484	433	554	536	588	554	761	744	830
2	120	446	602	535	624	558	714	691	758	714	981	959	1070
4	112	572	772	686	801	715	915	887	972	915	1258	1230	1373
2	93	838	1131	1006	1174	1048	1341	1299	1425	1341	1844	1801	2012
8	80	1096	1480	1316	1534	1370	1757	1699	1863	1757	2411	2356	2632
ess..lbs.		20	27	24	28	25	32	31	34	32	44	43	48
nsion...		13½		16¼		10		12¼		6¾		8¼	
ion, %...		18	18	20	20	15	15	18	18	12	12	14	14
. hr.lbs.		17.3	16.6	16.6	15.2	17.0	16.4	16.3	15.8	17.5	17.0	16.8	16.0

table contains figures for 95 lbs., cylinder ratio 3⅓ to 1; and to 1.

pansion Engines, Non-condensing.—Receiver only Jacketed.

. P.	Stroke, inches.	Revolutions per Minute.	Horse-power when Cutting off at 42 per cent of Stroke in First Cylinder.		Horse-power when Cutting off at 50 per cent of Stroke in First Cylinder.		Horse-power when Cutting off at 67 per cent of Stroke in First Cylinder.	
			180 lbs.	200 lbs.	180 lbs.	200 lbs.	180 lbs.	200 lbs
2	10	370	55	64	70	84	95	108
3½	12	318	70	81	90	106	120	137
6½	14	277	104	121	133	158	179	204
9	16	246	136	158	174	207	234	267
2½	18	222	195	226	250	296	335	382
5	20	185	241	279	308	366	414	471
8½	24	158	323	374	413	490	555	632
3½	28	138	433	502	554	657	744	848
8	32	120	558	647	714	847	959	1093
3	34	112	715	829	915	1089	1230	1401
2	42	93	1048	1215	1341	1592	1801	2053
0	48	80	1370	1589	1754	2082	2356	2685
press., lbs.			25	29	32	38	43	49
ns.......			16		13		10	
ndens....			14		12		10	
.hr., lbs.			20.76	19.36	19.25	17.00	17.89	
ap. lbs.			2.59	2.39	2.40	2.12	2.23	

Triple-expansion Engines—Condensing—St[...] Jacketed.

Diameter Cylinders, inches. H.P.	I.P.	L.P.	Stroke, inches.	Revolutions per Minute.	Horse-power when Cutting off at ¼ Stroke in First Cylinder. 120 lbs.	140 lbs.	160 lbs.	Horse-power when Cutting off at ⅓ Stroke in First Cylinder. 120 lbs.	140 lbs.	160 lbs.	Horse-power when Cutting off at ½ Stroke in First Cylinder. 120 lbs.	140 lbs.	160 lbs.
4¾	7½	12	10	370	35	42	48	44	53	59	57	72	84
5½	8½	13½	12	318	45	53	62	56	67	76	73	92	107
6½	10½	16½	14	277	67	79	92	83	100	112	108	137	139
7½	12	19	16	246	87	103	120	109	131	147	141	180	208
9	14½	22½	18	222	125	148	172	156	187	211	203	257	299
10	16	25	20	185	154	183	212	192	231	260	250	317	368
11½	18	28½	24	158	206	245	284	258	310	348	335	426	494
13	22	33½	28	138	277	329	381	346	415	467	450	571	663
15	24½	38	32	120	357	424	491	446	535	602	580	736	854
17	27	43	34	112	458	543	629	572	686	772	744	944	1095
20	33	52	42	93	670	796	922	838	1006	1131	1089	1383	1605
23½	38	60	48	80	877	1041	1206	1096	1316	1480	1424	1808	2099
Mean effec. press., lbs.					16	19	22	20	24	27	26	33	38.3
No. of expansions....					26.8			20.1			13.4		
Per cent cyl. condens.					19	19	19	16	16	16	12	12	12
St. p. I.H.P. p. hr., lbs.					14.7	13.9	13.3	14.3	13.98	13.2	14.3	13.6	13.0
Coal at 8 lb. evap., lbs.					1.8	1.73	1.66	1.78	1.74	1.65	1.78	1.70	1.62

Type of Engine to be used where Exhaust-st[...] needed for Heating.—In many factories more or less of t[...] exhausted from the engines is utilized for boiling, drying, hea[...] Where all the exhaust-steam is so used the question of economic[...] steam in the engine itself is eliminated, and the high-pressure simp[...] is entirely suitable. Where only part of the exhaust-steam is used[...] quantity so used varies at different times, the question of adopting[...] a condensing, or a compound engine becomes more complex. This[...] is treated by C. T. Main in Trans. A. S. M. E., vol. x. p. 48. He sh[...] the ratios of the volumes of the cylinders in compound engines sho[...] according to the amount of exhaust-steam that can be used for he[...] case is given in which three different pressures of steam are re[...] could be used, as in a worsted dye-house: the high or boiler pre[...] the engine, an intermediate pressure for crabbing, and low pre[...] boiling, drying, etc. If it did not make too much complication of[...] the engine, the boiler-pressure might be used in the high-pressure[...] exhausting into a receiver from which steam could be taken for[...] small engines and crabbing, the steam remaining in the receiver[...] into the intermediate cylinder and expanded there to from 5 to 10 l[...] the atmosphere and exhausted into a second receiver. From this[...] is drawn the low-pressure steam needed for drying, boiling, warm[...] etc., the steam remaining in receiver passing into the condensing c[...]

Comparison of the Economy of Compound and [...] cylinder Corliss Condensing Engines, each exp[...] about Sixteen Times. (D. S. Jacobus, Trans. A. S. M. E.[...]

The engines used in obtaining comparative results are located at[...] I. and II. of the Pawtucket Water Co.

The tests show that the compound engine is about 30% more e[...] than the single-cylinder engine. The dimensions of the two eng[...] follows: Single 20″ × 48″; compound 15″ and 30½″ × 30″[...] used per [...]-power per hour was: single 20.35 lbs., compoun[...]

B[...] engines are steam-jacketed, practically on the[...] [...]ll boiler-pressure, viz. single 106.3 lbs., comp[...]

am-pressure in the case of the compound engine is 127 lbs., or 21 r than for the single engine. If the steam-pressure be raised this the case of the single engine, and the indicator-cards be increased ly, the consumption for the single-cylinder engine would be 19.97 our per horse-power.

ylinder vs. Three-cylinder Compound Engine.—ock triple-expansion engine, built for the Merrick Thread Co., Mass., is constructed so that the intermediate cylinder may be cut circuit and the high-pressure and low-pressure cylinders run as a der compound, using the same conditions of initial steam-pressure The diameters of the cylinders are 12, 16, and $24\frac{11}{16}$ inches, the the first two being 36 in. and that of the low-pressure cylinder 48 esults of a test reported by S. M. Green and G. I. Rockwood, Trans. ., vol. xiii. 647, are as follows: In lbs. of dry steam used per I.H.P. 12 and $24\frac{11}{16}$ in. cylinders only used, two tests 13.06 and 12.76 lbs., 2.91. All three cylinders used, two tests 12.67 and 12.90 lbs., average difference is only 1%, and would indicate that more than two cylinunnecessary in a compound engine, but it is pointed out by Prof. that the conditions of the test were especially favorable for the der engine, and not relatively so favorable for the three cylinders. n-pressure was 142 lbs. and the number of expansions about 25. liscussion on the Rockwood type of engine, Trans. A. S. M. E., vol.

of Water contained in Steam on the Efficiency of am-engine. (From a lecture by Walter C. Kerr, before the Institute, 1891.)—Standard writers make little mention of the effect ed moisture on the expansive properties of steam, but by common ather than any demonstration they seem to agree that moisture an ill effect simply to the percentage amount of its presence. % moisture will increase the water rate of an engine 5%.
nents reported in 1893 by R. C. Carpenter and L. S. Marks, Trans. ., xv., in which water in varying quantity was introduced into the e, causing the quality of the steam to range from 99% to 58% dry, nat throughout the range of qualities used the consumption of dry indicated horse-power per hour remains practically constant, and that the water was an inert quantity, doing neither good nor harm. ars that the extra work done by the heat of the entrained water pansion is sensibly equal to the extra negative work which it does haust and compression, that the heat carried in by the entrained forms no useful function, and that a fair measure of the economy ine is the consumption of dry and saturated steam.

ive Commercial Economy of Best Modern Types of und and Triple-expansion Engines. (J. E. Denton, *Machinist*, Dec. 17, 1891.)—The following table and deductions relative commercial economy of the compound and triple type for stationary practice in steam plants of 500 indicated horse-power. is based on the tests of Prof. Schröter, of Munich, of engines built urg, and those of Geo. H. Barrus on the best plants of America, and d estimates of cost obtained from several first-class builders.

ion, or Corliss engines of u-compound-receiver con- type, expanding 16 times. ressure 120 lbs.	Lbs. water per hour per H.P., by measurement.	13.6	14.0
	Lbs. coal per hour per H.P., assuming 8.5 lbs. actual evaporation.	1.60	1.65
ion, or Corliss engines of ple-expansion four-cylin- eiver condensing type, ex- 22 times. Boiler pressure,	Lbs. water per hour per H.P., by measurement.	12.56	12.80
	Lbs. coal per hour per H.P., assuming 8.5 lbs. actual evaporation.	1.48	1.50

ures in the first column represent the best recorded performance d those in the second column the probable reliable performance.

cost of triple-expansion plant per horse-power, including *himney, heaters, foundations,* piping and erection...... ..

ing table shows the total annual cost of operation, wi *, the plant running* 300 days in the year, for 10 hou *lay:*

Hours running per day	10	[illegible]
	Per H.P.	Per H.
Expense for coal. Compound plant	$9.90	[illegible]
Expense for coal. Triple plant	9.00	[illegible]
Annual saving of triple plant in fuel	0.90	[illegible]
Annual interest at 5% on $4.50	$0.23	[illegible]
Annual depreciation at 5% on $4.50	0.23	[illegible]
Annual extra cost of oil, 1 gallon per 24-hour day, at $0.50, or 15% of extra fuel cost	0.15	[illegible]
Annual extra cost of repairs at 3% on $4.50 per 24 hours	0.06	[illegible]
	$0.67	[illegible]
Annual saving per H.P.	$0.23	[illegible]

The saving between the compound and triple types is much less th[an] involved in the step from the single-expansion condensing to the com[pound] engine. The increased cost per horse-power of the triple plant [over] compound is due almost entirely to the extra cost of the triple engi[ne and] its foundations, the boilers costing the same or slightly more, owing [to] extra strength. In the case of the single *versus* the compound, [how]ever about one third of the increased cost of the compound engine is offset [by] less cost of the latter's boilers.

Taking the total cost of the plants at $33.50, $36.50 and $41 per [horse-]power respectively, the figures in the table imply that the total annu[al sav]ing is as follows for coal at $4 per ton:

1. A compound 500 horse-power plant costs $18,250, and saves abou[t] for 10 hours' service, and $4885 for 24 hours' service, per year over a [single] plant costing $16,750. That is, the compound saves its extra cost in [10-hour] service in about one year, or in 24-hour service in four months.

2. A triple 500 horse-power plant costs $20,500, and saves about $[illegible] year in 10-hour service, or $826 in 24-hour service, over a compound, thereby saving its extra cost in 10-hour service in about 19¾ years, or [in 24-]hour service in about 2¾ years.

Triple-expansion Pumping-engine at Milwau[kee]. Highest Economy on Record, 1893. (See paper on "[illegible] Contemporary Economy of the Steam-engine," by R. H. Thurston, [Trans.] A. S. M. E., xv. 313.)—Cylinders 28, 48 and 74 in. by 60 in. stroke; ra[tio of] volumes 1 to 3 to 7; total number of expansions 19.55; clearance[s: h. p.] 1.4%; int. 1.5%; l. p. 0.77%; volume of receivers: 1st, 101.3 cu. ft.; 2d, [illegible] ft.; steam-pressure gauge during test, average 121.5 lbs.; vacuum [illegible] absolute; revolutions 20.3 per minute; indicated horse-power, h.p. [illegible] 169.6, l. p. 228.9; total, 573.9; total friction, horse-power 52.91 = [illegible] steam per I.H.P. per hour 11.678; B.T.U. per I.H.P. per min. [illegible] foot-pounds per 100 lbs. of coal, 143,306,000; per million B.T.U., [illegible]

Steam per I.H.P. per hour, from diagram, at cut-off	9.35	[illegible]
" " " " " " " " release	10.1	[illegible]
Steam accounted for by indicator at cut-off, per cent	87.1	[illegible]
" " " " " " release, "	94.0	[illegible]
Per cent of total steam used by jackets	9.29	

Highest Economy of the Two-cylinder Comp[ound] Pumping-engines.—Repeated tests of the Pawtucket-Corliss [engine,] 15 and 30½ by 30 in. stroke, gave a water consumption of [illegible] per I.H.P. per hour. Steam-pressure 123 lbs.; revolutions per [minute;] expansions about 16. Cylinders jacketed. The lowest water rate was [with] jackets in use; both jackets supplied with steam of boiler pressure[. The] average saving due to jackets was only about 2½ per cent. (Trans. [A. S.] M. E., xi. 328 and 1038; xiii. 176.)

This record was beaten in 1894 by a Leavitt pumping-engine at L[ouisville,] Ky. (Trans. A. S. M. E. xvi.) Cylinders 27.21 and 54.13 in. [illegible] stroke; revolutions per min. 18.57; piston speed 371.5 ft.; expa[nsions] [illegible] Cylinders and receiver jacke[ted]; steam-[pressu]re, gauge, 140 lbs.

er I.H.P. per hour, 12.223 lbs. Duty per million B.T.U. = 138,126,000

t of a Triple-expansion Pumping-engine with and out Jackets, at Laketon, Ind., by Prof. J. E. Denton (Trans. A. E., xiv. 1340).—Cylinders 24, 34 and 54 in. by 36 in. stroke; 28 revs. per H.P. developed about 320; boiler-pressure 150 lbs. Tests made on eight nt days with different sets of conditions in jackets. At 150 lbs. boiler-re, and about 20 expansions, with any pressure above 43 lbs. in all of kets and reheaters, or with no pressure in the high jacket, the per-nce was as follows: With 2.5% of moisture in the steam entering the , the jackets used 16% of the total feed-water. About 20% of the latter ndensed during admission to the high cylinder, and about 13.85 lbs. -water was consumed per hour per indicated horse-power. With no s or reheaters in action the feed-water consumption was 14.99 lbs., or ore than with jackets and reheaters. The consumption of lubricating two thirds of a gallon of machine oil and one and three quarter gal- cylinder oil per 24 hours. The friction of the engine in eight tests on nt days varied from 5.1% to 8.7%.

e regard the measurements of indicated horse-power and water as to an error of one per cent, which is probably a minimum allowance most careful determinations, the steam economy is the same for the ing conditions:

ny pressure from 43 to 131 in the intermediate and low jackets and ers.

ny pressure from 0 to 151 in the jacket of high cylinder.

ny cut-off from 21% to 23% in high cylinder, from 39% to 43% in inter-te cylinder, from 40% to 53% in low cylinder.

er Consumption of Three Types of Sulzer Engines.

(B. Donkin, Jr., *Eng'g*, Jan. 15, 1892, p. 77.)

RY AND AVERAGES OF TWENTY-ONE PUBLISHED EXPERIMENTS OF THE SULZER TYPE OF STEAM-ENGINE. ALL HORIZONTAL CONDENSING AND STEAM-JACKETED. From 1872 to 1891.

	Steam-pressure above Atmosphere.	Piston-speed.	Indicated Horse-power.	Steam Consumption, pounds per I.H.P. per hour, including Steam-pipe water and Jacket Water.	Steam Consumption, pounds per I.H.P. per hour, exclud'g Steam-pipe water, but including Jacket Water.	Remarks.
	lbs.	ft. per min		lbs.	lbs.	
e	72 to 95	272 to 433	157 to 400	18.7 to 19.8 Mean 19.4	17.9 to 19.2 Mean 18.95	5 exp. 1872–78
d.	84 to 104	384 to 689	133 to 524	13.35 to 16.0 Mean 14.44	13.4 to 15.5 Mean 14.3	10 exp. 1888–91
..	104 to 156	444 to 607	198 to 615	11.85 to 12.86 Mean 12.36	11.7 to 12.7 Mean 12.18	6 exp. 1888–89

le-expansion Corliss engine at Narragansett E. L. Co., Providence, R. lt by E. P. Allis Co. Cylinder 14, 25 and 33 in. by 48 in. stroke tested at s. per min.; 125 lbs. steam-pressure; steam per I.H.P. per hour 12.91 I.P. 516. A full account of this engine, with records of tests is given by Henthorn, in Trans. A. S. M. E., xii. 643.

keye-cross compound engine, tested at Chicago Exposition, by Geo. rrus (*Eng'g Record*, Feb. 17, 1894). Cylinder 14 and 28 by 24 in. stroke; at 165 r. p. m.; 120 lbs. steam-pressure. I.H.P. in four tests condens-nd one non-*condensing*............ 295 224 123 277 267
per horse-power per hour........ 16.07 15.71 17.22 16.07 23.24

tive Economy of Compound Non-condensing En- under Variable Loads.—F. M. Rites, in a paper on the Steam *tion in a Form of Single-acting Engine* (Trans. A. S. M. E. xiii. 537 an engine designed to meet the following problem: Given extreme range of conditions as to load or steam-pressure, either or fluctuate together or apart, violently or with easy gradations, to an engine whose economical performance should be as good as the engine were specially designed for a momentary condition—the adj to be complete and automatic. In the ordinary non-condensing co engine with light loads the high-pressure cylinder is frequently supply all the power and in addition drag along with it the low piston, whose cylinder indicates negative work. Mr. Rites sh peculiar value of a receiver of predetermined volume which acts as ance chamber for compression in the high-pressure cylinder. The house compound single-acting engine is designed upon this princip following results of tests of one of these engines rated at 175 H.P. economical load are given :

WATER RATES UNDER VARYING LOADS, LBS. PER H.P. PER HO

Horse-power	210	170	140	115	100	80
Non-condensing	22.6	21.9	22.2	22.2	22.4	21.6
Condensing	18.4	18.1	18.2	18.2	18.3	18.1

Efficiency of Non-condensing Compound Engine Lee Church, *Am. Mach.*, Nov. 19, 1891.)—The compound engine, densing, at its best performance will exhaust from the low-press der at a pressure 2 to 6 pounds above atmosphere. Such an engine limited in its economy to a very short range of power, for the rea its valve-motion will not permit of any great increase beyond power, and any material decrease below its rated power at once br expansion curve in the low-pressure cylinder below atmosphere. words, decrease of load tells upon the compound engine somewhat and much more severely, than upon the non-compound engine. commences the moment the expansion line crosses a line parall atmospheric line, and at a distance above it representing the mean pressure necessary to carry the frictional load of the engine. Whe sion falls to this point the low-pressure cylinder becomes an air-pu more or less of its stroke, the power to drive which must come high-pressure cylinder alone. Under the light loads common industries the low-pressure cylinder is thus a positive resistanc greater portion of its stroke. A careful study of this problem rev functions of a fixed intermediate clearance, always in communica the high-pressure cylinder, and having a volume bearing the sam that of the high-pressure cylinder that the high-pressure cylinde the low-pressure. Diagrams were laid out on this principle an until the best theoretical results were obtained. The designs were down on these lines, and the subsequent performance of the which some 600 have been built, have fully confirmed the judgm designers.

The effect of this constant clearance is to supply sufficient ste low-pressure cylinder under light loads to hold its expansion cu atmosphere, and at the same time leave a sufficient clearance vol high-pressure cylinder to permit of governing the engine on its co under light loads.

Economy of Engines under Varying Loads. (F W. C. Unwin's lecture before the Society of Arts, London, 1892.) eral result of numerous trials with large engines was that with load an indicated horse-power should be obtained with a consu 1½ pounds of coal per indicated horse-power for a condensing e 1¾ pounds for a non-condensing engine, figures which correspon 1¾ pounds to 2½ pounds of coal per effective horse-power. It was difficult to ascertain the consumption of coal in ordinary every but such facts as were known showed it was more than on trial.

In electric-lighting stations the engines work under a very load, and the results are far more unfavorable. An excellent W condensing engine, which on full-load trials worked with unde per effective horse-power hour, in the ordinary daily working of used 7½ pounds per effective horse-power hour in 1886, which w to 4.3 pounds in 1890 and 3.8 pounds in 1891. Probably in ve were the engines at electric-light stations working under a wo 4½ pounds per effective horse-power hour. In the case of motors working with a fluctuating load, still more extravaga

ENGINES IN ELECTRIC CENTRAL STATIONS.

Year	1886.	1890.	1892.
Coal used per hour per effective H.P.	8.4	5.6	4.9
" " " " " indicated "	6.5	4.35	3.8

-lectric-lighting stations the load factor, viz., the ratio of the average to the maximum, is extremely small, and the engines worked under unfavorable conditions, which largely accounted for the excessive fuel umption at these stations.

Steam-engines the fuel consumption has generally been reckoned on indicated horse-power. At full-power trials this was satisfactory h, as the internal friction is then usually a small fraction of the total. periment has, however, shown that the internal friction is nearly con-, and hence, when the engine is lightly loaded, its mechanical efficiency atly reduced. At full load small engines have a mechanical efficiency to 0.85, and large engines might reach at least 0.9, but if the internal on remained constant this efficiency would be much reduced at low rs. Thus, if an engine working at 100 indicated horse power had an effi-y of 0.85, then when the indicated horse-power fell to 50 the effective -power would be 35 horse-power and the efficiency only 0.7. Similarly, horse-power the effective horse-power would be 10 and the efficiency

periments on a Corliss engine at Creusot gave the following results :

tive power at full load	1.0	0.75	0.50	0.25	0.125
ensing, mechanical efficiency	0.82	0.79	0.74	0.63	0.48
ondensing, " "	0.86	0.83	0.78	0.67	0.52

light loads the economy of gas and liquid fuel engines fell off even rapidly than in steam-engines. The engine friction was large and y constant, and in some cases the combustion was also less perfect at loads. At the Dresden Central Station the gas-engines were kept ing at nearly their full power by the use of storage-batteries. The lts of some experiments are given below :

e load, per nt of full Power.	Gas-engine, cu. ft. of Gas per Brake H.P. per hour.	Petroleum Eng., Lbs. of Oil per B.H.P. per hr.	Petroleum Eng., Lbs. of Oil per B.H.P. per hr.
100	22.2	0.96	0.88
75	23.8	1.11	0.99
50	28.0	1.44	1.20
20	40.8	2.38	1.82
12½	66.3	4.25	3.07

team Consumption of Engines of Various Sizes.—W. C. win (Cassier's *Magazine*, 1894) gives a table showing results of 49 tests of ines of different types. In non-condensing simple engines, the steam umption ranged from 65 lbs. per hour in a 5-horse-power engine to 22 in a 134-H.P. Harris-Corliss engine. In non-condensing compound en-s, the only type tested was the Willans, which ranged from 27 lbs. in a .P. slow-speed engine, 122 ft. per minute, with steam-pressure of 84 lbs. 9.2 lbs. in a 40-H.P. engine, 401 ft. per minute, with steam-pressure 165 A Willans triple-expansion non-condensing engine, 39 H.P., 172 lbs. sure, and 400 ft. piston speed per minute, gave a consumption of 18.5 lbs. ondensing engines, nine tests of simple engines gave results ranging only 18.4 to 22 lbs., and, leaving out a beam pumping-engine running at slow d (240 ft. per minute) and low steam-pressure (45 lbs.), the range is only 18.4 to 19.8 lbs. In compound-condensing engines over 100 H.P., in 13 the range is from 13.9 to 20 lbs. In three triple-expansion engines the es are 11.7, 12.2, and 12.45 lbs., the lowest being a Sulzer engine of 360 In marine compound engines, the Fusiyama and Colchester, tested rof. Kennedy, gave steam consumption of 21.2 and 21.7 lbs.; and the or and Tartar triple-expansion engines gave 15.0 and 19.8 lbs.

ing the most favorable results which can be regarded as not excep-l, it appears that in test trials, with constant and full load, the expen-e of steam and coal is about as follows:

Kind of Engine.	Per Indicated Horse-power Hour.		Per Effective Horse-power Hour.	
	Coal, lbs.	Steam, lbs.	Coal, lbs.	Steam, "
ndensing	1.80	16.5	2.00	
sing	1.50	13.5	1.75	

These may be regarded as minimum values, rarely surpassed by the efficient machinery, and only reached with very good machinery [illegible] favorable conditions of a test trial.

Small Engines and Engines with Fluctuating Load [illegible] usually very wasteful of fuel. The following figures, illustrating the [illegible] economy, are given by Prof. Unwin, Cassier's *Magazine*, 1894.

COAL CONSUMPTION PER INDICATED HORSE-POWER IN SMALL ENGINES

In Workshops in Birmingham, Eng.

Probable I.H.P. at full load...	12	45	60	45	75	[illegible]
Average I.H.P. during observation..	2.96	7.37	8.2	8.6	23.64	[illegible]
Coal per I.H.P. per hour during observation, lbs.........	36.0	21.25	22.61	18.13	11.68	[illegible]

It is largely to replace such engines as the above that power will [illegible] tributed from central stations.

Steam Consumption in Small Engines.

Tests at Royal Agricultural Society's show at Plymouth, Eng. *Eng[illegible] ing*, June 27, 1890.

Rated H.P.	Compound or Simple.	Diam. of Cylinders.		Stroke, ins.	Max. Steam-pressure.	Per Brake H.P. per hour.	
		h.p.	l.p.			Coal.	Water.
5	simple	7	.....	10	75	12.12	78.1 lbs.
3	compound	3	6	6	110	4.82	42.03 "
2	simple	4½	.....	7½	75	11.77	89.9 "

Steam-consumption of Engines at Various S[illegible] (Profs. Denton and Jacobus, Trans. A. S. M. E., x. 722)—17 × 30 [illegible] non-condensing, fixed cut-off, Meyer valve.

STEAM-CONSUMPTION, LBS. PER I.H.P. PER HOUR.

Figures taken from plotted diagram of results.

Revs. per min.....	8	12	16	20	24	32	40	48	56	[illegible]
⅛ cut-off, lbs.....	39	35	32	30	29.3	29	28.7	28.5	28.3	[illegible]
¼ " "	39	34	31	29.5	29	28.4	28	27.5	27.1	[illegible]
½ " "	39	36	34	33	32	30.8	29.8	29.2	28.8	[illegible]

STEAM-CONSUMPTION OF SAME ENGINE; FIXED SPEED, 60 REVS. PE[illegible]

Varying cut-off compared with throttling-engine for same ho[illegible] and boiler-pressures:

Cut-off, fraction of stroke	0.1	0.15	0.2	0.25	0.3	0.4	0.5	0.6
Boiler-pressure, 90 lbs...	29	27.5	27	27	27.2	27.8	28.5	...
" 60 lbs..	39	34.2	32.2	31.5	31.4	31.6	32.2	34.1

THROTTLING-ENGINE, ⅞ CUT-OFF, FOR CORRESPONDING HORSE-PO[illegible]

Boiler-pressure, 90 lbs...	42	37	33.8	31.5	29.8	....	....	...
" 60 lbs...	....	50.1	49	46.8	44.6	41	....	...

Some of the principal conclusions from this series of tests are as [illegible]

1. There is a distinct gain in economy of steam as the speed incr[illegible] ½, ⅛, and ¼ cut-off at 90 lbs. pressure. The loss in economy for [illegible] cut-off is at the rate of 1/12 lb. of water per H.P. for each decr[illegible] revolution per minute from 86 to 26 revolutions, and at the rate [illegible] water below 26 revolutions. Also, at all speeds the ¼ cut-off is [illegible] nomical than either the ⅛ or ½ cut-off.

2. At 90 lbs. boiler-pressure and above ⅛ cut-off, to produce a [illegible] requires about 20% less steam than to cut off at ⅞ stroke and reg[illegible] throttle.

3. For the same conditions with 60 lbs. boiler-pressure, [illegible] throttling, the same mean effective pressure at ⅞ cut-off that

f about 1/16, requires about 30% more steam than for the latter

Piston-speed in Engines. (Proc. Inst. M. E., July, 1883, p. torpedo boat is an excellent example of the advance towards ds, and shows what can be accomplished by studying lightness gth in combination. In running at 22½ knots an hour, an engine ders of 16 in. stroke will make 480 revolutions per minute, which ft. per minute for piston-speed; and it is remarked that engines that high rate work much more smoothly than at lower speeds, he difficulty of lubrication diminishes as the speed increases.

h-speed Corliss Engine.—A Corliss engine, 20 × 42 in., has ing a wire-rod mill at the Trenton Iron Co.'s works since 1877, at itions or 1120 ft. piston-speed per minute (Trans. A. S. M. E., ii. ston-speed of 1200 ft. per min. has been realized in locomotive

imitation of Engine-speed. (Chas. T. Porter, in a paper imitation of Engine-speed, Trans. A. S. M. E., xiv. 806.)—The imitation to high rotative speed in stationary reciprocating steam-not found in the danger of heating or of excessive wear, nor, as lly believed, in the centrifugal force of the fly-wheel, nor in the to knock in the centres, nor in vibration. He gives two objections gh speeds: First, that "engines ought not to be run as fast as e;" second, the large amount of waste room in the port, which d for proper steam distribution. In the important respect of of steam, the high-speed engine has thus far proved a failure. n was looked for from high speed, because the loss by condensa-given surface would be divided into a greater weight of steam, but tation has not been realized. For this unsatisfactory result we y the blame chiefly on the excessive amount of waste room. The method of expressing the amount of waste room in the percentage it to the total piston displacement, is a misleading one. It should sed as the percentage which it adds to the length of steam admis-example, if the steam is cut off at 1/5 of the stroke, 8% added by room to the total piston displacement means 40% added to the steam admitted. Engines of four, five and six feet stroke may be run at from 700 to 800 ft. of piston travel per minute, but for izes, says Mr. Porter, 600 ft. per minute should be the limit.

nce of the Steam-jacket.—Tests of numerous engines with ut steam-jackets show an exceeding diversity of results, ranging y from 30% saving down to zero, or even in some cases showing an s. The opinions of engineers at this date (1894) is also as diverse as t, but there is a tendency towards a general belief that the jacket is able an appendage to an engine as was formerly supposed. An ex-*umé* of facts and opinions on the steam-jacket is given by Prof. in Trans. A. S. M. E., xiv. 462. See also Trans. A. S. M. E., xiv. 40; xiii. 176; xii. 426 and 1340; and Jour. F. I., April, 1891, p. 276. ing are a few statements selected from these papers.

lts of tests reported by the research committee on steam-jackets by the British Institution of Mechanical Engineers in 1886, indi-creased efficiency due to the use of the steam-jacket of from 1% to according to varying circumstances.

asserts that "it has been abundantly proved that steam-e not only advisable but absolutely necessary, in order that high xpansion may be efficiently carried out and the greatest possible of heat attained."

od finds the gain by its use, under the conditions of ordinary s a general average, to be about 20% on small and 8% or 9% on nes, varying through intermediate values with intermediate sizes, nderstood that the jacket has an effective circulation, and that s and sides are jacketed.

r Unwin considers that "in all cases and on all cylinders the useful; provided, of course, ordinary, not superheated, steam is the advantages may diminish to an amount not worth the interest ost."

r Cotterill says: Experience shows that a steam-jacket is advan-*t the amount to be gained* will vary according to circumstances. *ses it may be that* the advantage is small. Great cau *drawing conclusions* from any special set of experi *of jacketing.*

Mr. E. D. Leavitt has expressed the opinion that, in his practic jackets produce an increase of efficiency of from 15% to 20%.

In the Pawtucket pumping engine, 15 and 30⅛ × 30 in., 50 revs. steam-pressure 125 lbs. gauge, cut-off ¼ in h.p. and ⅛ in l.p. cy barrels only jacketed, the saving by the jackets was from 1% to 4%.

The superintendent of the Holly Mfg. Co. (compound pumping says: "In regard to the benefits derived from steam-jackets on cylinders, I am somewhat of a skeptic. From data taken on our gines and tests made I am yet to be convinced that there is any value in the steam-jacket." . . . "You might practically say is no difference."

Professor Schröter from his work on the triple-expansion engine burg, and from the results of his tests of the jacket efficiency on engine of the Sulzer type in his own laboratory, concludes: (1) of the jacket may vary within very wide limits, or even becom tive. (2) The shorter the cut-off the greater the gain by the jacket. (3) The use of higher pressure in the jacket than in the produces an advantage. The greater this difference the better. high-pressure cylinder may be left unjacketed without great los others should always be jacketed.

The test of the Laketon triple-expansion pumping-engine show of 8.3% by the use of the jackets, but Prof. Denton points out (Tr M. E., xiv. 1412) that all but 1.9% of the gain was ascribable to th range of expansion used with the jackets.

Test of a Compound Condensing Engine with an out Jackets at different Loads. (R. C. Carpenter, Tra M. E., xiv. 428.)—Cylinders 9 and 16 in. × 14 in. stroke; 112 lbs. boiler rated capacity 100 H.P.; 265 revs. per min. Vacuum, 23 in. From of several tests curves are plotted, from which the following princi are taken.

Indicated H.P.	30	40	50	60	70	80	90	100	110
Steam per I.H.P. per hour:									
With jackets, lbs.	22.6	21.4	20.3	19.6	19	18.7	18.6	18.9	19.[illegible]
Without jackets, lbs.	....	....	....	22.	20.5	19.6	19.2	19.1	19.[illegible]
Saving by jacket, p. c.	....	....	....	10.9	7.3	4.6	3.1	1.0	–1.[illegible]

This table gives a clue to the great variation in the apparent sa the steam-jacket as reported by different experimenters. Wit ticular engine it appears that when running at its most economi 100 H.P., without jackets, very little saving is made by use of t When running light the jacket makes a considerable saving, but loaded it is a detriment.

At the load which corresponds to the most economical rate, wit in jackets, or 100 H.P., the use of the jacket makes a saving of at a load of 60 H.P. the saving by use of the jacket is about 1 shape of the curve indicates that the relative advantage of the ja be still greater at lighter loads than 60 H.P.

Counterbalancing Engines.—Prof. Unwin gives the f counterbalancing vertical engines:

$$W_1 = W_2 \frac{r}{p}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad .$$

in which W_1 denotes the weight of the balance weight and p th its centre of gravity, W_2 the weight of the crank-pin and half th the connecting-rod, and r the length of the crank. For horizont

$$W_1 = \tfrac{2}{3}(W_2 + W_3)\frac{r}{p} \quad \text{to} \quad \tfrac{3}{4}(W_2 + W_3)\frac{r}{p}. \quad . \quad .$$

in which W_3 denotes the weight of the piston, piston-rod, cros the other half of the weight of the connecting-rod.

The American Machinist, commenting on these formulæ, say ontal engines formula (2) is often used; formula (1) will gi alance too light for vertical engines. We should use f mputing the counterbalance for both horizontal and v cepti[illegible] [illegible]otives, in which the counterbalance should

eventing Vibrations of Engines.—Many suggestions have made for remedying the vibration and noise attendant on the working big engines which are employed to run dynamos. A plan which has great satisfaction is to build hair-felt into the foundations of the e. An electric company has had a 90-horse-power engine removed its foundations, which were then taken up to the depth of 4 feet. A of felt 5 inches thick was then placed on the foundations and run up 2 feet sides, and on the top of this the brickwork was built up.—*Safety Valve.*

am-engine Foundations Embedded in Air.—In the sugar-ery of Claus Spreckels, at Philadelphia, Pa., the engines are distributed ically all over the buildings, a large proportion of them being on upper s. Some are bolted to iron beams or girders, and are consequently ent of all foundation. Some of these engines ran noiselessly and satis-rily, while others produced more or less vibration and rattle. To cor-he latter the engineers suspended foundations from the bottoms of the es, so that, in looking at them from the lower floors, they were literally ing in the air.—*Iron Age*, Mar. 13, 1890.

st of Coal for Steam-power.—The following table shows the unt and the cost of coal per day and per year for various horse-powers, 1 to 1000, based on the assumption of 4 lbs. of coal being used per hour orse-power. It is useful, among other things, in estimating the saving may be made in fuel by substituting more economical boilers and nes for those already in use. Thus with coal at $3.00 per ton, a saving 9000 per year in fuel may be made by replacing a steam plant of 1000 ., requiring 4 lbs. of coal per hour per horse-power, with one requiring 2 lbs.

Coal Consumption, at 4 lbs. per H.P. per hour; 10 hours a day; 300 days in a Year.					$1.50.		$2.00.		$3.00.		$4.00.	
Lbs.	Long Tons.		Short Tons.		Per Short Ton.		Per Short Ton.		Per Short Ton.		Per Short Ton.	
					Cost in Dollars.		Cost in Dollars.		Cost in Dollars.		Cost in Dollars.	
Per Day.	Per Day.	Per Year.	Per Day.	Per Year	Per Day.	Per Year	Per Day.	Per Year.	Per Day.	Per Year.	Per Day.	Per Year
40	.0179	5.357	.02	6	.03	9	.04	12	.06	18	.08	24
400	.1786	53.57	.20	60	.30	90	.40	120	.60	180	.80	240
1,000	.4464	133.92	.50	150	.75	225	1.00	300	1.50	450	2.00	600
2,000	.8928	267.85	1.00	300	1.50	450	2.00	600	3.00	900	4.00	1,200
3,000	1.3393	401.78	1.50	450	2.25	675	3.00	900	4.50	1,350	6.00	1,800
4,000	1.7857	535.71	2.00	600	3.00	900	4.00	1,200	6.00	1,800	8.00	2,400
6,000	2.6785	80[illegible].56	3.00	900	4.50	1,350	6.00	1,800	9.00	2,700	12.00	3,600
8,000	3.5714	1,071.42	4.00	1,200	6.00	1,800	8.00	2,400	12.00	3,600	16.00	4,800
10,000	4.4642	1,339.27	5.00	1,500	7.50	2,250	10.00	3,000	15.00	4,500	20.00	6,000
12,000	5.3571	1,607.13	6.00	1,800	9.00	2,700	12.00	3,600	18.00	5,400	24.00	7,200
14,000	6.2500	1,874.98	7.00	2,100	10.50	3,150	14.00	4,200	21.00	6.200	28.00	8,400
16,000	7.1428	2,142.84	8.00	2,400	12.00	3,600	16.00	4,800	24.00	7,200	32.00	9,600
18,000	8.0356	2,410.69	9.00	2,700	13.50	4,050	18.00	5,400	27.00	8,100	36.00	10,800
20,000	8.9285	2,678.55	10.00	3,000	15.00	4,500	20.00	6,000	30.00	9,000	40.00	12,000
24,000	10.7142	3,214.26	12.00	3,600	18.00	5,400	24.00	7,200	36.00	10,800	48.00	14,400
28,000	12.4999	3,749.97	14.00	4,200	21.00	6,300	28.00	8,400	42.00	11,600	56.00	16,800
32,000	14.2856	4,285.68	16.00	4,800	24.00	7,200	32.00	9,600	48.00	12,400	64.00	19,200
36,000	16.0713	4,821.39	18.00	5,400	27.00	8,100	36.00	10,800	54.00	14,200	72.00	21,600
40,000	17.8570	5,357.10	20.00	6,000	30.00	9,000	40.00	12,000	60.00	18,000	80.00	24,000

oring Steam Heat.—There is no satisfactory method for equalizing load on the engines and boilers in electric-light stations. Storage-batteries been used, but they are expensive in first cost, repairs, and attention. Halpin, of London, proposes to store heat during the day in specially *tructed reservoirs. As the water in the boilers is raised to 250 lbs. pres-it is conducted to cylindrical reservoirs resembling English horizonta s, and stored there for use when wanted.* In this way a comparative *boiler-plant can be used* for heating the water to 250 lbs. pr *h the twenty-four hours* of the day, and the stored wa *on at any time,* according to the magnitude of the d steam-engines are to be worked by the steam generated by the [illegible] pressure from this water, and the valves are to be arranged in [illegible] that the steam shall work at 130 lbs. pressure. A reservoir 8 ft. in [illegible] and 30 ft. long, containing 84,000 lbs. of heated water at 250 lbs. [illegible] would supply 5250 lbs. of steam at 130 lbs. pressure. As the steam [illegible] tion of a condensing electric-light engine is about 18 lbs. per horse-[illegible] hour, such a reservoir would supply 286 effective horse-power [illegible] 1878, in France, this method of storing steam was used on a [illegible] M. Francq, the engineer, designed a smokeless locomotive to work by [illegible] power supplied by a reservoir containing 400 gallons of water at [illegible] pressure. The reservoir was charged with steam from a stationary [illegible] at one end of the tramway.

Cost of Steam-power. (Chas. T. Main, A. S. M. E., x. 48.)—[illegible] costs in New England in 1888, per horse-power, based on engines of [illegible]

	Compound Engine.	Condensing Engine.
1. Cost engine and piping, complete...........	$25.00	$20.00
2. Engine-house...............................	8.00	7.50
3. Engine foundations	7.00	5.50
4. Total engine plant.................	40.00	33.00
5. Depreciation, 4% on total cost...	1.60	1.32
6. Repairs, 2% " " "	0.80	0.66
7. Interest, 5% " " "	2.00	1.65
8. Taxation, 1.5% on ¾ cost........	0.45	0.371
9. Insurance on engine and house...........	0.165	0.138
10. Total of lines 5, 6, 7, 8, 9...........	5.015	4.139
11. Cost boilers, feed-pumps, etc..	9.33	18.33
12. Boiler-house.........	2.92	4.17
13. Chimney and flues...	6.11	7.30
14. Total boiler-plant.................	18.36	24.80
15. Depreciation, 5% on total cost.............	0.918	1.240
16. Repairs, 2% " " "	.367	.496
17. Interest, 5% " " "	.918	1.240
18. Taxation, 1.5% on ¾ cost..................	.207	.279
19. Insurance, 0.5% on total cost...............	.092	.124
20. Total of lines 15 to 19	2.502	3.379
21. Coal used per I.H.P. per hour, lbs.........	1.75	2.50
22. Cost of coal per I.H.P. per day of 10¼ hours at $5.00 per ton of 2240 lbs.........	cts. 4.00	cts. 5.72
23. Attendance of engine per day..............	0.60	0.40
24. " " boilers " "	0.53	0.75
25. Oil, waste, and supplies, per day..........	0.25	0.22
26. Total daily expense.	5.38	7.09
27. Yearly running expense, 308 days, per I.H.P..................................	$16.570	$21.837
28. Total yearly expense, lines 10, 20, and 27..	24.087	29.355
29. Total yearly expense per I.H.P. for power if 50% of exhaust-steam is used for heating	12.597	14.387
30. Total if all ex.-steam is used for heating...	8.624	7.324

When exhaust-steam or a part of the receiver-steam is used for [illegible] part of the steam in a condensing engine is diverted from [illegible] and used for other purposes than power, the value of such [illegible] from the cost of the total amount of steam generated, in order he cost properly chargeable to power. The figures in lines 29 ased on an assumption made by Mr. Main of losses of heat 25% between the boiler and the exhaust-pipe, an allowance ably too large.

ROTARY STEAM-ENGINES.

urbines.—The steam turbine is a small turbine wheel which am as the ordinary turbine does with water. (For description s and the Dow steam turbines see Modern Mechanism, p. 298, rsons turbine is a series of parallel-flow turbines mounted side shaft; the Dow turbine is a series of radial outward-flow tur- like a series of concentric rings in a single plane, a stationary ing between each pair of movable rings. The speeds of the es enormously exceed those of any form of engine with recip- on, or even of the so-called rotary engines. The three- and four- nes of the Brotherhood type, in which the several cylinders rouped radially about a common crank and shaft, often exceed ns per minute, and have been driven, experimentally, above steam turbine of Parsons makes 10,000 and even 20,000 revolu- e Dow turbine is reputed to have attained 25,000. (See Trans. ol. x. p. 680, and xii. p. 888; Trans. Assoc. of Eng'g Societies, 3; *Eng'g*, Jan. 13, 1888, and Jan. 8, 1892; *Eng'g News*, Feb. 27. turbine, exhibited in 1889, weighed 68 lbs., and developed 10 consumption of 47 lbs. of steam per H.P. per hour, the steam g 70 lbs. The Dow turbine is used to spin the fly-wheel of the do. The dimensions of the wheel are 13.8 in. diam., 6.5 in. of gyration 5.57 in. The energy stored in it at 10,000 revs. 0,000 ft.-lbs.

aval Steam Turbine, shown at the Chicago exhibition, tion wheel somewhat similar to the Pelton water-wheel. The irected by a nozzle against the plane of the turbine at quite a and tangentially against the circumference of the medium the blades. The angle of the blades is the same at the side of d discharge. The width of the blade is constant along the ess of the turbine.

is expanded to the pressure of the surroundings before arriv- des. This expansion takes place in the nozzle, and is caused king its sides diverging. As the steam passes through this ecific volume is increased in a greater proportion than the of the channel, and for this reason its velocity is increased. omentum, till the end of the expansion at the last sectional zzle. The greater the expansion in the nozzle the greater its is point. A pressure of 75 lbs. and expansion to an absolute ne atmosphere give a final velocity of about 2625 ft. per second. is carried further in this steam turbine than in ordinary steam- s is on account of the steam expanding completely during its ressure of the surroundings.

ng the greatest possible effect the admission to the blades must blows and the velocity of discharge as low as possible. These ould require in the steam turbine an enormous velocity of high as 1300 to 1650 ft. per second. The centrifugal force, puts a limit to the use of very high velocities. In the 5 horse- e the velocity of periphery is 574 ft. per second, and the num- tions 30,000 per minute.

arefully the turbine may be manufactured it is impossible, on evenness of the material, to get its centre of gravity to corre- to its geometrical axle of revolution; and however small this y be, it becomes very noticeable at such high velocities. De ceeded in solving the problem by providing the turbine with a . This yielding shaft allows the turbine at the high rate of ust itself and revolve around its true centre of gravity, the *the shaft meanwhile* describing a surface of revolution.

g-box the speed is reduced from 30,000 revolutions to 3000 *driver on the turbine* shafts, which sets in motion a cog- *es its own diameter*. These gearings are provided with spiral *an angle of* about 45°. The shaft of the larger co- *ed of 3000 revolutions*, is provided at its outer er *ther transmission of* the power.

Rotary Steam-engines, other than steam turbines, have invented by the thousands, but not one has attained a commercial s The possible advantages, such as saving of space, to be gained by a engine are overbalanced by its waste of steam.

The Tower Spherical Engine, one of the most recent for rotary-engine, is described in Proc. Inst. M. E., 1885, also in Mechanism, p. 296.

DIMENSIONS OF PARTS OF ENGINES.

The treatment of this subject by the leading authorities on the stea gine is very unsatisfactory, being a confused mass of rules and fo based partly upon theory and partly upon practice. The practice of b shows an exceeding diversity of opinion as to correct dimensions treatment given below is chiefly the result of a study of the works of Ra Seaton, Unwin, Thurston, Marks, and Whitham, and is largely a con tion of a series of articles by the author published in the *America chinist*, in 1894, with many alterations and much additional matter. der to make a comparison of many of the formulæ they have been a to the assumed cases of six engines of different sizes, and in some this comparison has led to the construction of new formulæ.

Cylinder. (Whitham.)—Length of bore = stroke + breadth of ring − 1/8 to 1/2 in; length between heads = stroke + thickness of p sum of clearances at both ends; thickness of piston = breadth of thickness of flange on one side to carry the ring + thickness of fo plate.

Thickness of flange or follower....	5/8 to 1/2 in.	3/4 in.	1
For cylinder of diameter............	8 to 10 in.	36 in.	60 to

Clearance of Piston. (Seaton.)—The clearance allowed vari the size of the engine from 1/8 to 3/8 in. for roughness of castings and 1/8 in. for each working joint. Naval and other very fast-running have a larger allowance. In a vertical direct-acting engine the part wear so as to bring the piston nearer the bottom are three, viz., th journals, the crank-pin brasses, and piston-rod gudgeon-brasses.

Thickness of Cylinder. (Thurston.)—For engines of th types and under moderate steam-pressures, some builders have fo years restricted the stress to about 2550 lbs. per sq. in.

$$t = ap_1D + b. \quad . \quad . \quad . \quad . \quad . \quad .$$

is a common proportion; t, D, and b being thickness, diam., and a added quantity varying from 0 to 1/2 in., all in inches; p_1 is the initia anced steam-pressure per sq. in. In this expression b is made la horizontal than for vertical cylinders, as, for example, in large eng in the one case and 0.2 in the other, the one requiring re-boring m the other. The constant a is from 0.0004 to 0.0005; the first value f cal cylinders, or short strokes; the second for horizontal engine long strokes.

Thickness of Cylinder and its Connections for M Engines. (Seaton.)—D = the diam. of the cylinder in inches; p = the safety-valves in lbs. per sq. in.; f, a constant multiplier = thic barrel + .25 in.

Thickness of metal of cylinder barrel or liner, not to be less than 3000 when of cast iron.* . . . .

$$\text{Thickness of cylinder-barrel} = \frac{p \times D}{5000} + 0.6 \text{ in.} \quad . \quad . \quad . \quad . \quad .$$

" " liner = $1.1 \times f$. . . . .

Thickness of liner when of steel $p \times D \div 6000 + 0.5$

" metal of steam-ports $= 0.6 \times f$.

" " valve-box sides $= 0.65 \times f$.

* When made of exceedingly good material, at least, ... thickness ... 0.8 of that given by the above rules.

ss of metal of valve-box covers $= 0.7 \times f$.

" cylinder bottom $= 1.1 \times f$, if single thickness.

" " " $= 0.65 \times f$, if double "

" " covers $= 1.0 \times f$, if single "

" " " $= 0.6 \times f$, if double "

cylinder flange $= 1.4 \times f$.

" cover-flange $= 1.3 \times f$.

" valve-box " $= 1.0 \times f$.

" door-flange $= 0.9 \times f$.

" face over ports $= 1.2 \times f$.

" " " $= 1.0 \times f$, when there is a false-face.

" false-face $= 0.8 \times f$, when cast iron.

" " $= 0.6 \times f$, when steel or bronze.

am gives the following from different authorities:

$$\text{Van Buren: } \begin{cases} t = 0.0001Dp + 0.15\sqrt{D}; & (5) \\ t = 0.03\sqrt{Dp}. & (6) \end{cases}$$

$$\text{Tredgold: } \quad t = \frac{(D + 2.5)p}{1900}. \qquad (7)$$

$$\text{Weisbach: } \quad t = 0.8 + 0.00033pD. \qquad (8)$$

$$\text{Seaton: } \quad t = 0.5 + 0.0004pD. \qquad (9)$$

$$\text{Haswell: } \begin{cases} t = 0.0004pD + \tfrac{1}{16} \text{ (vertical)}; & (10) \\ t = 0.0005pD + \tfrac{1}{8} \text{ (horizontal)}. & (11) \end{cases}$$

am recommends (6) where provision is made for the reboring, and mple strength and rigidity are secured, for horizontal or vertical s of large or small diameter; (9) for large cylinders using steam 0 lbs. gauge-pressure, and

$$t = 0.003D\sqrt{p} \text{ for small cylinders.} \qquad (12)$$

Marks gives $t = 0.00028pD$. (13)

a smaller value than is given by the other formulæ quoted; but ays that it is not advisable to make a steam-cylinder less than 0.75 under any circumstances.

llowing table gives the calculated thickness of cylinders of engines and 50 in. diam., assuming p the maximum unbalanced pressure on n = 100 lbs. per sq. in. As the same engines will be used for calcu- f other dimensions, other particulars concerning them are here r reference.

DIMENSIONS, ETC., OF ENGINES.

No.	1 and 2.	3 and 4.	5 and 6.
d horse-power .... I.H.P.	50	450	1250
cyl., in .... D	10	30	50
eet L	1 2	2½ 5	4 8
r min. .... r	250 125	130 65	90 45
eed, ft. per min. .... S	500	650	700
piston, sq. in .... a	78.54	706.86	1963.5
ective pressure ... M.E.P.	42	32.3	30
al unbalanced press. ... P	7854	70,686	196,350
al per sq. in. .... p	100	100	100

THICKNESS OF CYLINDER BY FORMULA.	1 and 2.	3 and 4.	5 and 6.
(1) $.0004pD + 0.5$, short stroke...	.90	1.70	2.50
(1) $.0005pD + 0.5$, long stroke	1.00	2.00	3.00
(2) $.00033pD$	.33	.99	1 67
(3) $.0002pD + 0.6$	.80	1.40	1.60
(5) $.0001pD + .15\sqrt{D}$	.57	1.12	1.56
(6) $.03\sqrt{Dp}$	.95	1.64	2.12
(7) $\frac{(D + 2.5)}{1900}p$	.66	1.71	2.76
(8) $.00033pD + 0.8$	1.13	1.79	2.45
(9) $.0004pD + 0.5$	.90	1.70	2.50
(10) $.0004pD + \frac{1}{8}$ (vertical)	.53	1.33	2.13
(11) $.0005pD + \frac{1}{8}$ (horizontal)	.63	1.63	2.63
(12) $.003D\sqrt{p}$ (small engines)......	.30(?)	....	....
(13) $.00028pD$	.28(?)	.84(?)	1.40(?)
Average of first eleven	.76	1.48	2.26

The average corresponds nearly to the formula $t = .00037Dp + 0.4$ in. A convenient approximation is $t = .0004Dp + 0.3$ in., which gives for

Diameters	10	20	30	40	50	60 in.
Thicknesses...........	.70	1.10	1.50	1.90	2.30	2.70 in.

The last formula corresponds to a tensile strength of cast iron of 12,000 lbs., with a factor of safety of 10 and an allowance of 0.3 in. for reboring.

Cylinder-heads.—Thurston says: Cylinder-heads may be given a thickness, at the edges and in the flanges, exceeding somewhat that of the cylinder. An excess of not less than 25% is usual. It may be thinner in the middle. Where made, as is usual in large engines, of two disks with intermediate radiating, connecting ribs or webs, that section which is safe against shearing is probably ample. An examination of the designs of experienced builders, by Professor Thurston, gave

$$t = \frac{Dp}{3000} + \tfrac{1}{4} \text{ inch}, \qquad (1)$$

D being the diameter of that circle in which the thickness is taken.

Thurston also gives $\quad t = .005D\sqrt{p} + 0.25. \qquad (2)$

Marks gives $\quad t = 0.003D\sqrt{p}. \qquad (3)$

He also says a good practical rule for pressures under 100 lbs. per sq. in. is to make the thickness of the cylinder-heads $1\frac{1}{4}$ times that of the walls; and applying this factor to his formula for thickness of walls, or $.00028pD$, we have

$$t = .00035pD. \qquad (4)$$

Whitham quotes from Seaton,

$$t = \frac{pD + 500}{2000}, \text{ which is equal to } .0005pD + .25 \text{ inch}. \qquad (5)$$

Seaton's formula for cylinder bottoms, quoted above, is

$$t = 1.1f, \text{ in which } f = .0002pD + .85 \text{ inch, or } t = .00022pD + .93. \qquad (6)$$

Applying the above formulæ to the engines of 10, 30, and 50 inches diameter, with maximum unbalanced steam-pressure of 100 lbs. per sq. in., we have

Cylinder diameter, inches =	10	30	50
(1) $t = .00033Dp + .25$ =	.58	1.25	1.82
(2) $t = .005D\sqrt{p} + .25$ =	.75	1.75	2.75
(3) $t = .003D\sqrt{p}$ =	.30	.90	1.50
(4) $t = .00035Dp$ =	.35	1.05	1.75
(5) $t = .0005Dp + .25$ =	.75	1.75	2.75
(6) $t = .00022Dp + .93$ =	1.15	1.59	2.03
Average of 6	.65	1.38	2.10

verage is expressed by the formula $t = .00036Dp + .31$ inch.
r's "Modern Locomotive Construction," p. 24, gives for locomotive r-heads for pressures up to 120 lbs.:

meters, in.........	19 to 22	16 to 18	14 to 15	11 to 13	9 to 10
ess, in...	1¼	1	1	⅞	¾

g the pressure at 120 lbs. per sq. in., the thicknesses 1¼ in. and ¾ in. inders 22 and 10 in. diam., respectively, correspond to the formula $.035Dp + .33$ inch.

-stiffened Cylinder-covers.—Seaton objects to webs for ng cast-iron cylinder-covers as a source of danger. The strain on is one of tension, and if there should be a nick or defect in the dge of the web the sudden application of strain is apt to start a He recommends that high-pressure cylinders over 24 in. and low-e cylinders over 40 in. diam. should have their covers cast hollow, o thicknesses of metal. The depth of the cover at the middle should it ¼ the diam. of the piston for pressures of 80 lbs. and upwards, t of the low-pressure cylinder-cover of a compound engine equal to the high-pressure cylinder. Another rule is to make the depth at idle not less than 1.3 times the diameter of the piston-rod. In the Navy the cylinder-covers are made of steel castings, ¾ to 1¼ in. enerally cast without webs, stiffness being obtained by their form, s often a series of corrugations.

nder-head Bolts.—Diameter of bolt-circle for cylinder-head = er of cylinder + 2 × thickness of cylinder + 2 × diameter of bolts. lts should not be more than 6 inches apart (Whitham).

s gives for number of bolts $b = \frac{.7854D^2p}{5000c} = .0001571\frac{D^2p}{c}$, in which $c =$ a single bolt, p = boiler-pressure in lbs. per sq. in.; 5000 lbs. is taken afe strain per sq. in. on the nominal area of the bolt.

n says: Cylinder-cover studs and bolts, when made of steel, should uch a size that the strain in them does not exceed 5000 lbs. per sq. in. f less than ⅞ inch diameter it should not exceed 4500 lbs. per sq. in. f iron the strain should be 20% less.

ston says: Cylinder flanges are made a little thicker than the cylin-d usually of equal thickness with the flanges of the heads. Cylinder-hould be so closely spaced as not to allow springing of the flanges kage, say, 4 to 5 times the thickness of the flanges. Their diameter be proportioned for a maximum stress of not over 4000 to 5000 lbs. uare inch.

= diameter of cylinder, p = maximum steam-pressure, b = number s, s = size or diameter of each bolt, and 5000 lbs. be allowed per sq. ominal area of the bolt, $.7854D^2p = 3927bs^2$; whence $bs^2 = .0002D^2p$; $02\frac{D^2p}{s^2}$; $s = .01414\sqrt{\frac{p}{b}}$. For the three engines we have:

Diameter of cylinder, inches........	10	30	50
Diameter of bolt-circle, approx	13	35	57.5
Circumference of circle, approx....	40.8	110	180
Minimum No. of bolts, circ. ÷ 6.....	7	18	30
Diam. of bolts, $s = .01414D\sqrt{\frac{p}{b}}$......	¾ in.	1.00	1.29

iameter of bolt for the 10-inch cylinder is 0.54 in. by the formula, nch is as small as should be taken, on account of possible overstrain wrench in screwing up the nut.

Piston. Details of Construction of Ordinary Pis- (Seaton.)—*Let D be the diameter of the piston in inches, p the effec-ssure per square inch on it, x a constant multiplier, found as follows:*

$$x = \frac{D}{60} \times \sqrt{p} + 1.$$

The thickness of front of piston near the boss $= 0.2 \times x$
" " " " rim $= 0.17 \times x$.
" back " $= 0.18 \times x$.
" boss around the rod $= 0.3 \times x$.
" flange inside packing-ring $= 0.23 \times x$.
" " at edge $= 0.25 \times x$.
" packing-ring $= 0.15 \times x$.
" junk-ring at edge $= 0.23 \times x$.
" " inside packing-ring $= 0.21 \times x$.
" " at bolt-holes $= 0.35 \times x$.
" metal around piston edge $= 0.25 \times x$.
The breadth of packing-ring $= 0.63 \times x$.
" depth of piston at centre $= 1.4 \times x$.
" lap of junk-ring on the piston $= 0.45 \times x$.
" space between piston body and packing-ring $= 0.3 \times x$.
" diameter of junk-ring bolts $= 0.1 \times x + 0.25$
" pitch " " " $=$ 10 diameters.
" number of webs in the piston $= (D + 20) \div 12$
" thickness " " " $= 0.18 \times x$.

Marks gives the approximate rule: Thickness of piston-head $= \sqrt[4]{}$
which $l =$ length of stroke, and $d =$ diameter of cylinder in inches.
iam says in a horizontal engine the rings support the piston, or at
part of it, under ordinary conditions. The pressure due to the weight
the piston upon an area equal to 0.7 the diameter of the cyl
breadth of ring-face should never exceed 200 lbs. per sq. in. He also
a formula much used in this country: Breadth of ring-face $= 0.15 \times$
eter of cylinder.

For our engines we have diameter = ...	10	30
	Thickness of piston-h	
Marks, $\sqrt[4]{lD}$; long stroke........................	3.31	[illegible]
Marks, " ; short stroke........................	3.94	[illegible]
Seaton, depth at centre $= 1.4x$...................	4.30	9.80
Seaton, breadth of ring $= .63x$....................	1.89	4.41
Whitham, breadth of ring $= .15D$.................	1.50	4.50

Diameter of Piston Packing-rings. — These are gen
turned, before they are cut, about ¼ inch diameter larger than the cyl
for cylinders up to 20 inches diameter, and then enough is cut out of the
to spring them to the diameter of the cylinder. For larger cylinder
rings are turned proportionately larger. Seaton recommends an
of 1% of the diameter of the cylinder.

Cross-section of the Rings.—The thickness is commonly
1/30th of the diam. of cyl. + ⅛ inch, and the width = thickness +
For an eccentric ring the mean thickness may be the same as for a
uniform thickness, and the minimum thickness = ⅔ the maximum.

A circular issued by J. H. Dunbar, manufacturer of packing
Youngstown, O., says: Unless otherwise ordered, the thickness of rin
be made equal to .03 × their diameter. This thickness has been
to be satisfactory in practice. It admits of the ring being made abou
to the foot larger than the cylinder, and has, when new, a tension of
two pounds per inch of circumference, which is ample to prevent le
if the surface of the ring and cylinder are smooth.

As regards the width of rings, authorities "scatter" from very na
very wide, the latter being fully ten times the former. For instance,
gives $W = d\ .014 + .08$. Whitham's formula is $W = d\ .15$. In bo
mulæ W is the width of the ring in inches, and d the diameter of the c
in inches. Unwin's formula makes the width of a 20″ ring $W =$
$+ .08 = .36''$, while Whitham's is $20 \times .15 = 3''$ for the same diam
ring. There is much less difference in the practice of engine-builders
respect, but there is still room for a standard width of ring. It is
that for cylinders over 16″ diameter ¾″ is a popular and practical
and ½″ for cylinders of that size and under.

Fit of Piston-rod into Piston. (Seaton.)—The most e
and reliable practice is to turn the piston-rod end with a
inch for small engines, and ⅛ inch for large ones, make the

...tion of the rod is three fourths of that of the body, then ...t parallel; the rod should then fit into the piston so ...ween it and the shoulder for large pistons, and 1/16 in. ...lder prevents the rod from splitting the piston, and ... turned true after long wear without encroaching on

...d to the rod by a nut, and the size of the rod should ... on the section at the bottom of the thread does not ... in. for iron, 7000 lbs. for steel. The depth of this nut ... diameter which would be found by allowing these ...d be locked to prevent its working loose.

...ston-rods.—Unwin gives

$$d'' = bD\sqrt{p}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1$$

...der diameter in inches, p is the maximum unbalanced ... in., and the constant $b = 0.0167$ for iron, and $b =$ ...on, from an examination of a considerable number

$$'' = \sqrt[4]{\frac{D^2pL^2}{a}} + \frac{D}{80}, \text{ nearly}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

...ines), in which $a = 10{,}000$ and upward in the various ...arine screw engines or ordinary fast engines on ... values, while "low-speed engines" being less ...hock give $a = 15{,}000$, often.

...ton-rod to the piston and to the crosshead should ... at least 8 or 10. Marks gives

$$\text{... for iron; for steel } d'' = 0.0105D\sqrt{p}; \quad . \quad . \quad (3)$$

$$...\overline{l^2p}, \text{ for iron; for steel } d'' = 0.03525\sqrt[4]{D^2l^2p}, \quad (4)$$

... of stroke, all dimensions in inches. Deduce the ...r (3), and if this diameter is less than $1/12l$, then use

$$\text{...eter of piston-rod} = \frac{\text{Diameter of cylinder}}{F}\sqrt{p}.$$

... values of F:

..., direct-acting	$F = 60$
...return connnecting-rod, 2 rods	$F = 80$
...inary stroke, direct-acting............	$F = 50$
... " "	$F = 48$
...y long " "	$F = 45$
...dium stroke, oscillating	$F = 45$

...ry long, as compared with the stroke usual for the ... of cylinder.

...ansive engine p, the effective pressure should be ...orking pressure, or 15 lbs. above that to which the ...ded; for a compound engine the value of p for the ...d be taken as the absolute pressure, less 15 lbs., ...safety-valve; for the medium-pressure the ... half the absolute boiler-pressure; and for ...ssure to which the escape-valve is loaded ...te pressure, which can be got in the re... ...dvantage to make all the rods of a com... ... the rule.

...the engines of 10, 30, and 50 in. diameter, ...9:

Diameter of Piston-rods.

Diameter of Cylinder, inches........ ...	10		30
Stroke, inches........................ ...	12	24	30
Unwin, iron, $.0167D\sqrt{p}$................	1.67	1.67	5.01
Unwin, steel, $.0144D\sqrt{p}$..............	1.44	1.44	4.32
Thurston $\sqrt[4]{\frac{D^2pL^2}{10,000}}+\frac{D}{80}$ (L in feet).	1.13	......	3.12
Thurston, same with $a = 15,000$......	......	1.40	......
Marks, iron, $.0179D\sqrt{p}$.........	1.79	......	5.37
Marks, iron, $.03901\sqrt[4]{D^2l^2p}$	1.35	1.91	3.70
Marks, steel, $.0105D\sqrt{p}$...............	(1.05)	......	(3.15)
Marks, steel, $.03525\sqrt[4]{D^2l^2p}$.....	1.22	1.73	3.34
Seaton, naval engines, $\frac{D}{60}\sqrt{p}$..........	1.67	......	5.01
Seaton, land engine, $\frac{D}{45}\sqrt{p}$....	......	2.22	......
Average of four for iron............	1.49	1.82	4.30

The figures in brackets opposite Marks' third formula
since they are less than ⅛ of the stroke, and the figu
fourth formula would be taken instead. The figure 1.79
formula would be rejected for the engine of 24-inch strok

An empirical formula which gives results approximati
ages is $d'' = .013\sqrt{Dlp}$.

The calculated results from this formula, for the six en
tively, 1.42, 1.88, 3.90, 5.61, 6.37, 9.01.

Piston-rod Guides.—The thrust on the guide, wh
rod is at its maximum angle with the line of the piston-
the formula: Thrust = total load on piston × tangent o
of connecting-rod = $p \tan \theta$. This angle, θ, is the angle
stroke of piston ÷ length of connecting-rod.

Ratio of length of connecting-rod to stroke.....	2
Maximum angle of connecting-rod with line of piston-rod..................................	14° 29′
Tangent of the angle.......................... .	.258
Secant of the angle	1.0327

Seaton says: The area of the guide-block or slipper sur
thrust is taken should in no case be less than will admit o
lbs. on the square inch; and for good working those surfa
thrust when going ahead should be sufficiently large to p
mum pressure exceeding 100 lbs. per sq. in. When the
well lubricated this allowance may be exceeded.

Thurston says: The rubbing surfaces of guides are so
if V be their relative velocity in feet per minute, and p t
pressure on the guide in lbs. per sq. in., $pV < 60,000$ and p

The lower is the safer limit; but for marine and stati
allowable to take $p = 60,000 \div V$. According to Rankin
$p = \frac{44800}{V+20}$, where p is the pressure in lbs. per sq. in. an
rubbing in feet per minute. This includes the sum of al
the two rubbing surfaces together.

Some British builders of portable engines restrict th
the guides and cross-heads to less than 40, sometimes 3

For a mean velocity of 600 feet per minute, Prof.
give ... $p > 66.7$; Rankine's gives $p = 72.3$ lbs

slides in square inches $= \dfrac{P}{p_a \sqrt{n^2 - 1}} = \dfrac{.7854d^2 p_1}{p_a \sqrt{n^2 - 1}}$,

l unbalanced pressure, p_1 = pressure per square inch
meter of cylinder, p_a = pressure allowable per square inch
= length of connecting-rod ÷ length of crank. This is
formula, $A = P \tan \theta \div p_a$. For $n = 5$, $p_1 = 100$ and p_a
For the three engines 10, 30 and 50 in. diam., this would
slides, A = 20, 180 and 500 sq. in., respectively. Whitham
pressure on the slide may be as high as 500 lbs. per sq. in.,
here is good lubrication and freedom from dust. Station-
engines are usually designed to carry 100 lbs. per sq. in.,
is case is reduced from 50% to 60% by grooves. In locomo-
ressure ranges from 40 to 50 lbs. per sq. in. of slide, on ac-
cessibility of the slide, dirt, cinder, etc.
agreement among the authorities as to the formula for
, $A = P \tan \theta \div p_a$; but the value given to p_a, the allow-
square inch, ranges all the way from 35 lbs. to 500 lbs.
ting-rod. *Ratio of length of connecting-rod to length*
rience has led generally to the ratio of 2 or 2½ to 1, the
ng and easy-working rod, the former a rather short, but
e one (Thurston). Whitham gives the ratio of from 2 to 4½.
2 to 4.
the Connecting-rod.—The calculation of the diameter of
on a theoretical basis, considering it as a strut subject to
e and bending stresses, and also to stress due to its inertia,
gines, is quite complicated. See Whitham, Steam-engine
urston, Manual of S. E., p. 100. Empirical formulas are as
lar rods, largest at the middle, D = diam. of cylinder, l =
ting-rod in inches, p = maximum steam-pressure per sq. in.

liam. at middle, $d'' = 0.0272 \sqrt{Dl \sqrt{p}}$.
liam. at necks, d'' = 1.0 to 1.1 × diam. of piston-rod.

am. at middle, $d'' = \dfrac{D}{55} \sqrt{p}$.

am. at necks, $d'' = \dfrac{D}{60} \sqrt{p}$.

n., $d'' = 0.0179D \sqrt{p}$. if diam. is greater than 1/24 length.

n., $d'' = 0.02758 \sqrt{Dl \sqrt{p}}$ if diam. found by (5) is less than

liam. at middle, $d'' = a \sqrt{D L \sqrt{p}} + C$, D in inches, L in
d C = ½ inch for fast engines, a = 0.08 and C = ¾ inch for

ys: The rod may be considered as a strut free at both ends,
its diameter accordingly.

The diam. at the ends may be 0.875 of the diam. at the middle. Seaton's empirical formula when translated into terms of D and
same as the second one by Marks, viz., $d'' = 0.02758 \sqrt{Dl \sqrt{p}}$. Wh
(1) is also practically the same.

(10) Taking Seaton's more complex formula, with length of con
rod $= 2.5 \times$ length of stroke, and $r = 12$ and 16, respectively, it red
Diam. at middle $= .02294 \sqrt{P}$ and $.02411 \sqrt{P}$ for short and long en
gines, respectively.

Applying the above formulas to the engines of our list, we have

Diameter of Connecting-rods.

Diameter of Cylinder, inches.	10		30		
Stroke, inches	12	24	30	60	[illegible]
Length of connecting-rod l	30	60	75	150	[illegible]
(3) $d'' = \frac{D}{55} \sqrt{p} = .0182D \sqrt{p}$	1.82	1.82	5.46	5.46	[illegible]
(5) $d'' = .0179D \sqrt{p}$	1.79	......	5.37	......	[illegible]
(6) $d'' = .02758 \sqrt{Dl \sqrt{p}}$	......	2.14	......	5.85	......
(7) $d'' = 0.15 \sqrt{DL \sqrt{p}} + \frac{1}{2}$	2.87	......	7.00	......	[illegible]
(7) $d'' = 0.08 \sqrt{DL \sqrt{p}} + \frac{3}{4}$	......	2.54	......	5.65	......
(9) $d'' = .03 \sqrt{P}$	2.67	2.67	7.97	7.97	[illegible]
(10) $d'' = .02294 \sqrt{P}$; $.02411 \sqrt{P}$	2.03	2.14	6.09	6.41	[illegible]
Average	2.24	2.26	6.38	6.27	[illegible]

Formulæ 5 and 6 (Marks), and also formula 10 (Seaton), give the
diameters for the long-stroke engine; formulæ 7 give the larger dia
for the short-stroke engines. The average figures show but little dif
in diameter between long- and short-stroke engines; this is what m
expected, for while the connecting-rod, considered simply as a c
would require an increase of diameter for an increase of length, th
remaining the same, yet in an engine generally the shorter the con
rod the greater the number of revolutions, and consequently the grea
strains due to inertia. The influences tending to increase the di
therefore tend to balance each other, and to render the diameter to
extent independent of the length. The average figures correspond
to the simple formula $d'' = .021D \sqrt{p}$. The diameters of rod for the
diameters of engine by this formula are, respectively, 2.10, 6.30, and
Since the total pressure on the piston $P = .7854D^2p$, the formula is
lent to $d' = .0237 \sqrt{P}$.

Connecting-rod Ends.—For a connecting-rod end of the
type, where the end is secured with two bolts, each bolt should be
tioned for a safe tensile strength equal to two thirds the maximum
thrust in the connecting-rod.

The cap is to be proportioned as a beam loaded with the maxim
of the connecting-rod, and supported at both ends. The calculation
be made for rigidity as well as strength, allowing a maximum defle
1/100 inch. For a strap-and-key connecting-rod end the strap is desig
tensile strength, considering that two thirds of the pull on the conn
rod may come on one arm. At the point where the metal is slotted
key and gib, the straps must be thickened to make the cross-sectio
to that of the remainder of the strap. Between the end of the strap
slot the strap is liable to fail in double shear, and sufficient metal m
provided at the end to prevent such failure.

The breadth of the key is generally one fourth of the width of t
and the length, parallel to the strap, should be such that the cr
will have a shearing strength equal to the tensile strength of th
the strap. The taper of the key is generally about ⅝ inch to th

ed Connecting-rods.—In modern high-speed engines it is y to make the connecting-rods of rectangular instead of circular he sides being parallel, and the depth increasing regularly from ead end to the crank-pin end. According to Grashof, the bending the rod due to its inertia is greatest at 6/10 the length from the end, and, according to this theory, that is the point at which the ould be greatest, although in practice the section is made greatest nk-pin end.

or Thurston furnishes the author with the following rule for tapered g-rod of rectangular section: Take the section as computed by the $'' = 0.1\sqrt{DL\sqrt{p}} + 3/4$ for a circular section, and for a rod 4/3 the gth, placing the computed section at 2/3 the length from the small carrying the taper straight through this fixed section to the large s brings the computed section at the surge point and makes it han the rod for which a tapered form is not required.

the above formula, multiplying L by 4/3, and changing it to l in becomes $d = 1/30\sqrt{Dl\sqrt{p}} + 3/4''$. Taking a rectangular section ne area as the round section whose diameter is d, and making the the section h = twice the thickness t, we have $.7854d^2 = ht = 2t^2$, $= .627d = .0209\sqrt{Dl\sqrt{p}} + .47''$, which is the formula for the thick-stance between the parallel sides of the rod. Making the depth at head end $= 1.5t$, and at 2/3 the length $= 2t$, the equivalent depth at end is $2.25t$. Applying the formula to the short-stroke engines of ples, we have

of cylinder, inches	10	30	59
ches	12	30	48
connecting-rod	30	75	120
s, $t = .0209\sqrt{Dl\sqrt{p}} + .47 =$	1.61	3.60	5.59
crosshead end, $1.5t =$	2.42	5.41	8.39
crank end, $2\frac{1}{4}t$	3.62	8.11	12.58

icknesses t, found by the formula $t = .0209\sqrt{Dl\sqrt{p}} + .47$, agree ith the more simple formula $t = .01D\sqrt{p} + .60''$, the thicknesses d by this formula being respectively 1.6, 3.6, and 5.6 inches.

rank-pin.—A crank-pin should be designed (1) to avoid heating, rength, (3) for rigidity. The heating of a crank-pin depends on the on its rubbing-surface, and on the coefficient of friction, which ries greatly according to the effectiveness of the lubrication. It also upon the facility with which the heat produced may be carried us it appears that locomotive crank-pins may be prevented to some om overheating by the cooling action of the air through which they high speed.

arks gives $l = .0000247\,fpND^2 = 1.038f\,\dfrac{\text{(I.H.P.)}}{L}$ . . . . (1)

ham gives $l = 0.9075f\,\dfrac{\text{(I.H.P.)}}{L}$, ... (2)

l = length of crank-pin journal in inches, f = coefficient of friction, ay be taken at .03 to .05 for perfect lubrication, and .08 to .10 for im- p = mean pressure in the cylinder in pounds per square inch; D ter of cylinder in inches; N = number of single strokes per minute; indicated horse-power; L = length of stroke in feet. These are independent of the diameter of the pin, and Marks states as a aw, within *reasonable limits as to* pressure and speed of rubbing, *r a bearing is made, for a given* pressure and number of revolution *r it will work; and* its diameter has no effect upon its hea *he above formulæ* are deduced empirically from dimensio *of existing marine* engines. Marks says that about one *equired for crank-pins* of propeller engines will serve for t *l engines,* and one tenth for locomotive engines, maki formula for locomotive crank-pins $l = .00000247 f p N D^2$, or if p = … = .06, and $N = 600$, $l = .013 D^2$.

Whitham recommends for pressure per square inch of projected … naval engines 500 pounds, for merchant engines 400 pounds, for paddle … engines 800 to 900 pounds.

Thurston says the pressure should, in the steam-engine, never … or 600 pounds per square inch for wrought-iron pins, or about … figure for steel. He gives the formula for length of a steel pin, …

$$l = PR \div 600{,}000, \quad . \quad . \quad . \quad . \quad .$$

in which P and R are the mean total load on the pin in pounds … number of revolutions per minute. For locomotives, the divisor … taken as 500,000. Where iron is used this figure should be reduced … and 250,000 for the two cases taken. Pins so proportioned, if well … well lubricated, may always be depended upon to run cool; if … formed, perfectly cylindrical, well finished, and kept well oiled, … can be relied upon. It is assumed above that good bronze or … bearings are used.

Thurston also says: The size of crank-pins required to prevent … the journals may be determined with a fair degree of precision by … the formulæ given below:

$$l = \frac{P(V+20)}{44{,}800d} \text{ (Rankine, 1865); } \quad . \quad . \quad . \quad . \quad .$$

$$l = \frac{PV}{60{,}000d} \text{ (Thurston, 1862); } \quad . \quad . \quad . \quad . \quad .$$

$$l = \frac{PN}{350{,}000} \text{ (Van Buren, 1866). } \quad . \quad . \quad . \quad . \quad .$$

The first two formulæ give what are considered by their authors fair … ing proportions, and the last gives minimum length for iron pins. (… velocity of rubbing-surface in feet per minute.)

Formula (1) was obtained by observing locomotive practice in which … liability exists of annoyance by dust, and great risk occurs from … bility while running, and (2) by observation of crank-pins of naval … engines. The first formula is therefore not well suited for marine …

Steel can usually be worked at nearly double the pressure admissible … iron running at similar speed.

Since the length of the crank-pin will be directly as the power … upon it and inversely as the pressure, we may take it as

$$l = a\frac{\text{I.H.P.}}{L}, \quad . \quad . \quad . \quad . \quad . \quad . \quad .$$

in which a is a constant, and L the stroke of piston, in feet. The values … the constant, as obtained by Mr. Skeel, are about as follows: $a = 0.04$ where water can be constantly used; $a = 0.045$ where water is not generally …; $a = 0.05$ where water is seldom used; $a = 0.06$ where water is never …

Unwin gives

$$l = a\frac{\text{I.H.P.}}{r}, \quad . \quad . \quad . \quad . \quad . \quad . \quad .$$

in which r = crank radius in inches, $a = 0.3$ to $a = 0.4$ for iron and for … engines, and $a = 0.066$ to $a = 0.1$ for the case of the best steel and for … motive work, where it is often necessary to shorten up outside pins as … as possible.

J. B. Stanwood (*Eng'g*, June 12, 1891), in a table of dimensions of parts … American Corliss engines from 10 to 30 inches diameter of cylinder, … sizes of crank-pins which approximate closely to the formula

$$l = .275D'' + .5 \text{ in.}; \quad d = .25D''. \quad . \quad . \quad . \quad . \quad .$$

By calculating lengths of iron crank-pins for the engines 10, 20, and 30 … *diameter*, long and short stroke, by the several formulæ above given, … *found that* there is a great difference in the results, so that one formula … *certain cases* gives a length three times as great as another. … *(6)* give lengths much greater than the others. Marks (1), … Thurston … l = .06 I.H.P. ÷ L, and Unwin (8), l = 0.4 I.H.P. ÷ r, … sults … more closely.

ated lengths of iron crank-pins for the several cases by formula and (8) are as follows:

Length of Crank-pins.

cylinder.......... D	10	10	30	30	50	50
.... L (ft.)	1	2	2½	5	4	8
per minute.......... R	250	125	130	65	90	45
r.......... I.H.P.	50	50	450	450	1,250	1,250
ressure.......... lbs.	7,854	7,854	70,686	70,686	196,350	196,350
re per cent of max.......	42	42	32.3	32.3	30	30
re.......... P.	3,299	3,299	22,832	22,832	58,905	58,905
ank-pin..........						
, $l = .9075 \times .06$ I.H.P. $\div L$	2.18	1.09	8.17	4.08	14.18	7.09
$l = 1.038 \times .05$ I.H.P. $\div L$	2.39	1.30	9.34	4.67	16.22	8.11
$l = .06$ I.H.P. $\div L$.......	3.00	1.50	10.80	5.40	18.75	9.38
$l = .4$ I.H.P. $\div r$.........	3.33	1.67	12.0	6.0	20.83	10.42
$l = .3$ I.H.P. $\div r$.........	2.50	1.25	9.0	4.5	15.62	7.81
....	2.72	1.36	9.86	4.93	17.12	8.56
st steel, $l = .1 \frac{\text{I.H.P.}}{r}$	.83	.42	3.0	1.5	5.21	2.61
steel, $l = \frac{PR}{600,000}$.......	1.37	.69	4.95	2.47	8.84	4.42

ated lengths for the long-stroke engines are too low to prevent essures. See "Pressures on the Crank-pins," below.

ength of the Crank-pin is determined substantially as is crank. In overhung cranks the load is usually assumed as s extremity, and, equating its moment with that of the resist-pin,

$$\tfrac{1}{2}Pl = 1/32 t\pi d^3, \quad \text{and} \quad d = \sqrt[3]{\frac{5.1Pl}{t}},$$

= diameter of pin in inches, P = maximum load on the piston, imum allowable stress on a square inch of the metal. For iron ken at 9000 lbs. For steel the diameters found by this formula iced 10%. (Thurston.)

es the same formula in another form, viz.:

$$d = \sqrt[3]{\frac{5.1}{t}}\sqrt[3]{Pl} = \sqrt{\frac{5.1}{t}}\sqrt{P\frac{l}{d}}.$$

n to be used when the ratio of length to diameter is assumed. ght iron, t = 6000 to 9000 lbs. per sq. in.,

$$\sqrt[3]{\frac{5.1}{t}} = .0947 \text{ to } .0827; \quad \sqrt{\frac{5.1}{t}} = .0291 \text{ to } .0238.$$

, t = 9000 to 13,000 lbs. per sq. in.,

$$\sqrt[3]{\frac{5.1}{t}} = .0827 \text{ to } .0723; \quad \sqrt{\frac{5.1}{t}} = .0238 \text{ to } .0194.$$

gives $d = 0.0827\sqrt[3]{Pl} = 2.1058\sqrt[3]{\frac{l \times \text{I.H.P.}}{LR}}$ for strength, and

$\overline{Pl^3}$ *for rigidity, and recommends* that the diameter be calcu*mulæ, and the largest* result taken. The first is the sa*ula, with t taken* at 9000 lbs. per sq. in. The second is *eous assumption.*

Marks, calculating the diameter for rigidity, gives

$$d = 0.066\sqrt[4]{pl^3D^2} = 0.945\sqrt[4]{\frac{(\mathrm{H.P.})l^3}{LN}};$$

p = maximum steam-pressure in pounds per square inch, D = diame cylinder in inches, L = length of stroke in feet, N = number of single per minute. He says there is no need of an investigation of the stre a crank-pin, as the condition of rigidity gives a great excess of str

Marks's formula is based upon the assumption that the whole load concentrated at the outer end, and cause a deflection of .01 inch at point.

It is serviceable, he says, for steel and for wrought iron alike.

Using the average lengths of the crank-pins already found, we hav following for our six engines :

Diameter of Crank-pins.

Diameter of cylinder	10	10	30	30	30
Stroke, ft.	1	2	2½	5	4
Length of crank-pin,	2.72	1.36	9.86	4.93	17.[illegible]
Unwin, $d = \sqrt[3]{\frac{5.1Pl}{t}}$	2.29	1.82	7.34	5.82	12.[illegible]
Marks, $d = .066\sqrt[4]{pl^3D^2}$	1.39	.85	6.44	3.78	12.[illegible]

Pressures on the Crank-pins.—If we take the mean pressure the crank-pin = mean pressure on piston, neglecting the effect of the ing angle of the connecting-rod, we have the following, using the lengths already found, and the diameters according to Unwin and M

Engine No.	1	2	3	4	5
Diameter of cylinder, inches	10	10	30	30	30
Stroke, feet	1	2	2½	5	4
Mean pressure on pin, pounds	3,299	3,299	22,832	22,832	[illegible]
Projected area of pin, Unwin	6.23	236	72.4	28.7	[illegible]
" " " " Marks	3.78	1.16	63.5	18.6	[illegible]
Pressure per square inch, Unwin	530	1,398	315	796	[illegible]
" " " " Marks	873	2,845	360	1,228	[illegible]

The results show that the application of the formulæ for length a eter of crank-pins give quite low pressures per square inch of p area for the short-stroke high-speed engines of the larger sizes, but pressures for all the other engines. It is therefore evident that aft lating the dimensions of a crank-pin according to the formulæ given results should be modified, if necessary, to bring the pressure pe inch down to a reasonable figure.

In order to bring the pressures down to 500 pounds per square divide the mean pressures by 500 to obtain the projected area, or of length by diameter. Making $l = 1.5d$ for engines Nos. 1, 2, 4 a revised table for the six engines is as follows :

Engine, No.	1	2	3	4	5
Length of crank-pin, inches	3.15	3.15	9.86	8.37	17.12
Diameter of crank-pin	2.10	2.10	7.34	5.58	12.40

Crosshead-pin or Wrist-pin.—Whitham says the bearing *for the wrist-pin is found by the formula for crank-pin design.* Se *the diameter at the middle must, of course, be sufficient to withs bending action, and generally from this cause ample surface is giv good working; but in any case the area, calculated by multiplying* *eter of the journal by its length, should be such that the pressu exceed 1200 lbs. per sq. in., taking the maximum load on the total* [illegible] *on it.*

[illegible] *ines with the gudgeon shrunk into the jaws* rod, and working in brasses fitted into a recess in the piston-rod end and red by a wrought-iron cap and two bolts, Seaton gives:

ameter of gudgeon = 1.25 × diam. of piston-rod.
ngth of gudgeon = 1.4 × diam. of piston-rod.

the pressure on the section, as calculated by multiplying length by neter, exceeds 1200 lbs. per sq. in., this length should be increased.

B. Stanwood, in his "Ready Reference" book, gives for length of shead-pin 0.25 to 0.3 diam. of piston, and diam. = 0.18 to 0.2 diam. of on. Since he gives for diam. of piston-rod 0.14 to 0.17 diam. of piston, dimensions for diameter and length of crosshead-pin are about 1.25 and iam. of piston-rod respectively. Taking the maximum allowable press-at 1200 lbs. per sq. in. and making the length of the crosshead-pin = of its diameter, we have $d = \sqrt{P} \div 40$, $l = \sqrt{P} \div 30$, in which P = max-m total load on piston in lbs., d = diam. and l = length of pin in inches. the engines of our example we have:

neter of piston, inches	10	30	50
imum load on piston, lbs	7854	70,686	196,350
neter of crosshead-pin, inches	2.22	6.65	11.08
gth of crosshead-pin, inches	2.96	8.86	14.77
wood's rule gives diameter, inches	1.8 to 2	5.4 to 6	9.0 to 10
wood's rule gives length, inches	2.5 to 3	7.5 to 9	12.5 to 15
wood's largest dimensions give pressure per sq. in., lbs	1309	1329	1309

ch pressures are greater than the maximum allowed by Seaton.

he Crank-arm.—The crank-arm is to be treated as a lever, so that is the thickness in direction parallel to the shaft-axis and b its breadth section x inches from the crank-pin centre, then, bending moment M hat section = Px, P being the thrust of the connecting-rod, and f the strain per square inch,

$$Px = \frac{fab^2}{6} \text{ and } \frac{a \times b^2}{6} = \frac{T}{f}, \text{ or } a = \frac{6T}{b^2 \times f}; \; b = \sqrt{\frac{6T}{fa}}.$$

a crank-arm were constructed so that b varied as $\sqrt{x}$ (as given by the ve rule) it would be of such a curved form as to be inconvenient to manu-cture, and consequently it is customary in practice to find the maxi-m value of b and draw tangent lines to the curve at the points; these s are generally, for the same reason, tangential to the boss of the crank-at the shaft.

he shearing strain is the same throughout the crank-arm; and, conse-ntly, is large compared with the bending strain close to the crank-pin; so it is not sufficient to provide there *only* for bending strains. The ion at this point should be such that, in addition to what is given by the ulation from the bending moment, there is an extra square inch for ry 8000 lbs. of thrust on the connecting-rod (Seaton).

he length of the boss h into which the shaft is fitted is from 0.75 to 1.0 he diameter of the shaft D, and its thickness e must be calculated from twisting strain PL. (L = length of crank.)

or different values of length of boss h, the following values of thickness oss e are given by Seaton:

hen $h = D$, then $e = 0.35\ D$; if steel, 0.3.
$h = 0.9\ D$, then $e = 0.38\ D$, if steel, 0.32.
$h = 0.8\ D$, then $e = 0.40\ D$, if steel, 0.33.
$h = 0.7\ D$, then $e = 0.41\ D$, if steel, 0.34.

he crank-eye or boss into which the pin is fitted should bear the same tion to the pin that the boss does to the shaft.

he diameter of the shaft-end onto which the crank is fitted should be × diameter of shaft.

hurston says: *The empirical proportions* adopted by builders will com-*ly be found to fall well within* the calculated safe margin. These pro-*ons are, from the practice of* successful designers, about as follows:

the wrought-iron crank. the hub is 1.75 to 1.8 times the least diamet *t part of the shaft carrying* full load; the eye is 2.0 to 2.25 the *the inserted portion of the* pin, and their depths are, for the *he diameter of shaft,* and for the eye, 1.25 to 1.5 the diamete

The web is made 0.7 to 0.75 the width of adjacent hub or eye, and is g depth of 0.5 to 0.6 that of adjacent hub or eye.

For the cast-iron crank the hub and eye are a little larger, rang diameter respectively from 1.8 to 2 and from 2 to 2.2 times the diame shaft and pin. The flanges are made at either end of nearly the ful of hub or eye. Cast-iron has, however, fallen very generally into dis

The crank-shaft is usually enlarged at the seat of the crank to ab its diameter at the journal. The size should be nicely adjusted to al the shrinkage or forcing on of the crank. A difference of diameter fifth of 1%, will usually suffice: and a common rule of practice gi allowance of but one half of this, or .001.

The formulæ given by different writers for crank-arms practically since they all consider the crank as a beam loaded at one end and f the other. The relation of breadth to thickness may vary according taste of the designer. Calculated dimensions for our six engines are lows:

Dimensions of Crank-arms.

Diam. of cylinder, ins...	10	10	30	30	50
Stroke S, ins..............	12	24	30	60	48
Max. pressure on pin P, (approx.) lbs	7854	7854	70,686	70,686	196,350
Diam. crank-pin d.......	2.10	2.10	7.34	5.58	12.40
Diam. shaft, $a\sqrt[3]{\frac{\text{I.H.P.}}{R}}D$ ($a = 4.69$, 5.09 and 5.22)..	2.74	3.46	7.70	9.70	12.55
Length of boss, $.8D$.....	2.19	2.77	6.16	7.76	10.04
Thickness of boss, $.4D$..	1.10	1.39	3.08	3.88	5.02
Diam. of boss, $1.8D$......	4.93	6.23	13.86	17.46	22.59
Length crank-pin eye, $.8d$	1.76	1.76	5.87	4.46	9.92
Thickness of crank-pin eye, $.4d$...............	.88	.88	2.94	2.23	4.46
Max. mom. T at distance $\frac{1}{2}S - \frac{1}{2}D$ from centre of pin, inch-lbs.......	37,149	80,661	788,149	1,848,439	3,479,322
Thickness of crank-arm $a = .75D$	2.05	2.60	5.78	7.28	9.41
Greatest breadth, $b = \sqrt{\frac{6T}{9000a}}$	3.48	4.55	9.54	13.0	15.7
Min. mom. T_0 at distance d from centre of pin $= Pd$	16,493	16,493	528,835	394,428	2,434,740
Least breadth, $b_1 = \sqrt{\frac{6T_0}{9000a}}$	2.32	2.06	7.81	6.01	13.13

The Shaft.—Twisting Resistance.—From the general for torsion, we have: $T = \frac{\pi}{16} d^3S = .19635d^3S$, whence $d = \sqrt[3]{\frac{5.1T}{S}}$,

T = torsional moment in inch-pounds, d = diameter in inches, and shearing resistance of the material in pounds per square inch.

If a constant force P were applied to the crank-pin tangentially to the work done per minute would be

$$P \times L \times \frac{2\pi}{12} \times R = 33{,}000 \times \text{I.H.P.},$$

in which L = length of crank in inches, and R = revs. per min., mean twisting moment $T = \frac{\text{I.H.P.}}{R} \times 63{,}025$. Therefore

$$d = \sqrt[3]{\frac{5.1T}{S}} = \sqrt[3]{\frac{321{,}427\,\text{I.H.P.}}{RS}},$$

ay take the form

$$d = \sqrt[3]{\frac{\text{I.H.P.}}{R} \times F}, \text{ or } d = a\sqrt[3]{\frac{\text{I.H.P.}}{R}},$$

F and a are factors that depend on the strength of the material he factor of safety. Taking S at 45,000 pounds per square inch for iron, and at 60,000 for steel, we have, for simple twisting by a uni- gential force,

f safety	= 5	6	8	10	5	6	8	10
n......	F = 35.7	42.8	57.1	71.4	a = 3.3	3.5	3.85	4.15
el.....	F = 26.8	32.1	42.8	53.5	a = 3.0	3.18	3.5	3.77

, taking for safe working strength of wrought iron 9000 lbs., steel ., and cast iron 4500 lbs., gives $a = 3.294$ for wrought iron, 2.877 for d 4.15 for cast iron. Thurston, for crank-axles of wrought iron, = 4.15 or more.

says: For wrought iron, f, the safe strain per square inch, should ed 9000 lbs., and when the shafts are more than 10 inches diameter, Steel, when made from the ingot and of good materials, will ad- stress of 12,000 lbs. for small shafts, and 10,000 lbs. for those above diameter.

fference in the allowance between large and small shafts is to com- for the defective material observable in the heart of large shafting, the hammering failing to affect it.

rmula $d = a\sqrt[3]{\frac{\text{I.H.P.}}{R}}$ assumes the tangential force to be uniform

it is the only acting force. For engines, in which the tangential ries with the angle between the crank and the connecting-rod, and variation in steam-pressure in the cylinder, and also is influenced ertia of the reciprocating parts, and in which also the shaft may be d to bending as well as torsion, the factor a must be increased, to for the maximum tangential force and for bending.

gives the following table showing the relation between the maxi- d mean twisting moments of engines working under various condi- e momentum of the moving parts being neglected, which is allow-

Description of Engine.	Steam Cut-off at	Max. Twist Divided by Mean Twist. Mome't	Cube Root of the Ratio.
rank expansive...	0.2	2.625	1.38
"	0.4	2.125	1.29
"	0.6	1.835	1.22
"	0.8	1.698	1.20
nder expansive, cranks at 90°....	0.2	1.616	1.17
" "	0.3	1.415	1.12
" "	0.4	1.298	1.09
" "	0.5	1.256	1.08
" "	0.6	1.270	1.08
" "	0.7	1.329	1.10
" "	0.8	1.357	1.11
linder compound, cranks 120°....	h.p. 0.5, l.p. 0.66	1.40	1.12
" l. p. cranks e one a other, and h.p. midway	" "	1.26	1.08

also gives the following rules for ordinary practice for ordinary der marine engines:

eter of the tunnel-shafts $= \sqrt[3]{\frac{\text{I.H.P}}{R} \times F}$, or $a\sqrt[3]{\frac{\text{I.H.P.}}{\text{R}}}$.

Compound engines, cranks at right angles:

Boiler pressure 70 lbs., rate of expansion 6 to 7, $F = 70$, $a = 4.12$.
Boiler pressure 80 lbs., rate of expansion 7 to 8, $F = 72$, $a = 4.16$.
Boiler pressure 90 lbs., rate of expansion 8 to 9, $F = 75$, $a = 4.22$.

Triple compound, three cranks at 120 degrees:

Boiler pressure 150 lbs., rate of expansion 10 to 12, $F = 62$, $a = 3.9$
Boiler pressure 160 lbs., rate of expansion 11 to 13, $F = 64$, $a = 4.$
Boiler pressure 170 lbs., rate of expansion 12 to 15, $F = 67$, $a = 4.0$

Expansive engines, cranks at right angles, and the rate of expans boiler-pressure 60 lbs., $F = 90$, $a = 4.48$.

Single-crank compound engines, pressure 80 lbs., $F = 96$, $a = 4.58$.

For the engines we are considering it will be a very liberal allowanc ratio of maximum to mean twisting moment if we take it as equal t ratio of the maximum to the mean pressure on the piston. The fact then, in the formula for diameter of the shaft will be multiplied by the root of this ratio, or $\sqrt[3]{\frac{100}{42}} = 1.34$, $\sqrt[3]{\frac{100}{32.3}} = 1.45$, and $\sqrt[3]{\frac{100}{30}} = 1.49$ fo 10, 30, and 50-in. engines, respectively. Taking $a = 3.5$, which corres to a shearing strength of 60,000 and a factor of safety of 8 for steel, 45,000 and a factor of 6 for iron, we have for the new coefficient a_1 formula $d_1 = a_1\sqrt[3]{\frac{\text{I.H.P.}}{R}}$, the values 4.69, 5.08, and 5.22, from whic obtain the diameters of shafts of the six engines as follows:

Engine No.	1	2	3	4	5
Diam. of cyl.	10	10	30	30	50
Horse-power, I.H.P.	50	50	450	450	1250
Revs. per min., R	250	125	130	65	90
Diam. of shaft $d = a_1\sqrt[3]{\frac{\text{I.H.P.}}{R}}$	2.74	3.46	7.67	9.70	12.55

These diameters are calculated for twisting only. When the shaft is subjected to bending strain the calculation must be modified as below

Resistance to Bending.—The strength of a circular section to resist bending is one half of that to resist twisting. If B is the ben moment in inch-lbs., and d the diameter of the shaft in inches,

$$B = \frac{\pi d^3}{32} \times f; \text{ and } d = \sqrt[3]{\frac{B}{f} \times 10.2};$$

f is the safe strain per square inch of the material of which the sh composed, and its value may be taken as given above for twisting (Se

Equivalent Twisting Moment.—When a shaft is subje both twisting and bending simultaneously, the combined strain on an tion of it may be measured by calculating what is called the equiv *twisting moment;* that is, the two strains are so combined as to be tr as a twisting strain only of the same magnitude and the size of shaf culated accordingly. Rankine gave the following solution of the com action of the two strains.

If T = the twisting moment, and B = the bending moment on a secti a shaft, then the equivalent twisting moment $T_1 = B + \sqrt{B^2 + T^2}$.

Seaton says: Crank-shafts are subject always to twisting, bending shearing strains; the latter are so small compared with the former they are usually neglected directly, but allowed for indirectly by mean the factor f.

The two principal strains vary throughout the revolution, and the m mum equivalent twisting moment can only be obtained accurately eries of calculations of bending and twisting moments taken at fixed als, and from them constructing a curve of strains.

Considering the engines of our examples to have overhung crank aximum bending moment resulting from the thrust of the connecti crank-pin will take place when the engine is passing the the effect of the inertia of the reciprocating parts, uct of the total pressure on the piston by the distance parallel lines passing through the centres of the crank-pin and of the ft bearing, at right angles to their axes; which distance is equal to ength of crank-pin bearing + length of hub + ½ length of shaft-bearing + y clearance that may be allowed between the crank and the two bearings. r our six engines we may take this distance as equal to ½ length of ank-pin + thickness of crank-arm + 1.5 × the diameter of the shaft as eady found by the calculation for twisting. The calculation of diameter then as below:

Engine No.	1	2	3	4	5	6
iam. of cyl., in. .	10	10	30	30	50	50
orse-power........	50	50	450	450	1250	1250
vs. per min.. ..	250	125	130	65	90	45
ax. press. on pis, P	7,854	7,854	70,686	70,686	196,350	196,350
verage,* L in	6.32	7.94	22.20	26.00	36.80	42.25
mo. $PL=B$ in.-lb	49,637	62,361	1,569,222	1,837,836	7,225,680	8,295,788
vist. mom. T......	47,124	94,248	1,060,290	2,120,580	4,712,400	9,424,800
uiv. Twist. mom. $T_1=B+\sqrt{B^2+T^2}$ (approx.).........	118,000	175,000	3,463,000	4,647,000	15,840,000	20,850,000

* Leverage = distance between centres of crank-pin and shaft bearing = + 2.25d.

Having already found the diameters, on the assumption that the shafts re subjected to a twisting moment T only, we may find the diameter for sisting combined bending and twisting by multiplying the diameters eady found by the cube roots of the ratio $T_1 \div T$, or

	1.40	1.27	1.46	1.34	1.64	1.36
ving corrected diameters $d_1 =$...	3.84	4.39	11.35	12.99	20.58	21.52

By plotting these results, using the diameters of the cylinders for abscissas d diameters of the shafts for ordinates, we find that for the long-stroke gines the results lie almost in a straight line expressed by the formula, ameter of shaft = .43 × diameter of cylinder; for the short-stroke engines line is slightly curved, but does not diverge far from a straight line ose equation is, diameter of shaft = .4 diameter of cylinder. Using these o formulas, the diameters of the shafts will be 4.0, 4.3, 12.0, 12.9, 20.0, 21.5. . B. Stanwood, in *Engineering*, June 12, 1891, gives dimensions of shafts Corliss engines in American practice for cylinders 10 to 30 in. diameter. e diameters range from 4 15/16 to 14 15/16, following precisely the equation, ameter of shaft = ½ diameter of cylinder − 1/16 inch.

Fly-wheel Shafts.—Thus far we have considered the shaft as resist- the force of torsion and the bending moment produced by the pressure the crank-pin. In the case of fly-wheel engines the shaft on the opposite e of the bearing from the crank-pin has to be designed with reference to bending moment caused by the weight of the fly-wheel, the weight of shaft itself, and the strain of the belt. For engines in which there is an board bearing, the weight of fly-wheel and shaft being supported by bearings, the point of the shaft at which the bending moment is a ximum may be taken as the point midway between the two bearings or the middle of the fly-wheel hub, and the amount of the moment is the duct of the weight supported by one of the bearings into the distance m the centre of that bearing to the middle point of the shaft. The shaft hus to be treated as a beam supported at the ends and loaded in the ddle. In the case of an overhung fly-wheel, the shaft having only one ring, the point of maximum moment should be taken as the middle of bearing, and its amount is very nearly the product of half the weight the fly-wheel and the shaft into the distance from the middle of its hub m the middle of the bearing. The bending moment should be calculated l combined with the *twisting moment* as above shown, to obtain the *ivalent twisting moment, and* the diameter necessary at the point of *ximum moment calculated therefrom.*

the case of our six engines we assume that the weights of the *ls, together with the shaft,* are double the weight of fly-whee *ed from the formula;* $W = 785,400 \dfrac{d^2 s}{R^2 D^2}$ (given under Fly-wh that the shaft is supported by an outboard bearing, the distance betw[illegible] the two bearings being 2½, 5, and 10 feet for the 10-in., 30-in., and [illegible] engines, respectively. The diameters of the fly-wheels are taken [illegible] that their rim velocity will be a little less than 6000 feet per minute.

Engine No.	1	2	3	4	5	[illegible]
Diam. of cyl., inches	10	10	30	30	50	[illegible]
Diam. of fly-wheel, ft.	7.5	15	14.5	29	21	[illegible]
Revs. per min	250	125	130	65	90	[illegible]
Half wt. fly-wh'l and shaft, lb.	268	536	5,968	11,936	26,384	[illegible]
Lever arm for max. mom., in.	15	15	30	30	60	[illegible]
Max. bending moment, in.-lb.	4020	8040	179,040	358,080	1,583,070	[illegible]

As these are very much less than the bending moments calculated f[illegible] the pressures on the crank-pin, the diameters already found are suffi[illegible] for the diameter of the shaft at the fly-wheel hub.

In the case of engines with heavy band fly-wheels and with long fly-w[illegible] shafts it is of the utmost importance to calculate the diameter of the [illegible] with reference to the bending moment due to the weight of the fly-w[illegible] and the shaft.

B. H. Coffey (*Power*, October, 1892) gives the formula for combined [illegible]ing and twisting resistance, $T_1 = .196d^3S$, in which $T_1 = B + \sqrt{B^2 +}$[illegible] being the maximum, not the mean twisting moment; and finds [illegible] working values for $.196S$ as below. He says: Four points should be c[illegible]ered in determining this value: First, the nature of the material; s[illegible] the manner of applying the loads, with shock or otherwise; third, the [illegible] of the bending moment to the torsional moment—the bending mome[illegible] revolving shaft produces reversed strains in the material, which tend t[illegible]ture it; fourth, the size of the section. Inch for inch, large sectio[illegible] weaker than small ones. He puts the dividing line between large an[illegible] sections at 10 in. diameter, and gives the following safe values of $S \times$ [illegible] steel, wrought iron, and cast iron, for these conditions.

VALUE OF $S \times .196$.

Ratio.	Heavy Shafts with Shock.			Light shafts with Shock. Heavy Shafts No Shock.			Light S[illegible] No Sh[illegible]	
B to T.	Steel.	Wro't Iron.	Cast Iron.	Steel.	Wro't Iron.	Cast Iron.	Steel.	Wr[illegible] Iro[illegible]
3 to 10 or less	1045	880	440	1566	1320	660	2090	[illegible]
3 to 5 or less	941	785	393	1410	1179	589	1882	[illegible]
1 to 1 or less	855	715	358	1281	1074	537	1710	[illegible]
B greater than T	784	655	328	1176	984	492	1568	[illegible]

Mr. Coffey gives as an example of improper dimensions the [illegible] shaft of a 1500 H.P. engine at Willimantic, Conn., which broke whil[illegible] gine was running at 425 H.P. The shaft was 17 ft. 5 in. long betwee[illegible] of bearings, 18 in. diam. for 8 ft. in the middle, and 15 in. diam. for [illegible] mainder, including the bearings. It broke at the base of the fillet [illegible]ing the two large diameters, or 56½ in. from the centre of the beari[illegible] calculates the mean torsional moment to be 446,654 inch-pounds, [illegible] maximum at twice the mean; and the total weight on one bearing [illegible] lbs., which, multiplied by 56½ in., gives 4,945,445 in.-lbs. bending mo[illegible] the fillet. Applying the formula $T_1 = B + \sqrt{B^2 + T^2}$, gives for eq[illegible] twisting moment 9,971,045 in.-lbs. Substituting this value in the [illegible] $T_1 = .196, Sd^3$ gives for S the shearing strain 15,070 lbs. per sq. in., [illegible] metal had a shearing strength of 45,000 lbs., a factor of safety of [illegible] Mr. Coffey considers that 6000 lbs. is all that should be allowed for [illegible] these circumstances. This would give $d = 20.35$ in. If we take fr[illegible] Coffey's table a value of $.196S = 1100$, we obtain $d^3 = 9000$ nearly, or [illegible] in., instead [illegible] the actual diameter.

Lengt[illegible] [illegible]aft-bearings.—There is as great a differe[illegible] opinion [illegible], and as great a variation in practice concernin[illegible] journ[illegible] there is concerning crank-pins. The [illegible]

rnal being determined from considerations of its heating, the
ns concerning heating of crank-pins apply also to shaft-bearings,
rmulæ for length of crank-pins to avoid heating may also be use
the total load upon the bearing the resultant of all the pressures
on it, by the pressure on the crank, by the weight of the fly-wheel
pull of the belt. After determining this pressure, however,
sort to empirical values for the so-called constants of the formula
riables, which depend on the power of the bearing to carry aw
d upon the quantity of heat generated, which latter depends on t
re, on the number of square feet of rubbing surface passed o
inute, and upon the coefficient of friction. This coefficient is an
gly variable quantity, ranging from .01 or less with perfectly
urnals, having end-play, and lubricated by a pad or oil-bath, to .10
th ordinary oil-cup lubrication.

For shafts resisting torsion only, Marks gives for length of bea
$.00247fpND^2$, in which f is the coefficient of friction, p the mean
pounds per square inch on the piston, N the number of single str
inute, and D the diameter of the piston. For shafts under the c
ress due to pressure on the crank-pin, weight of fly-wheel, etc.,
e following: Let Q = reaction at bearing due to weight, S = st
eam pressure on piston, and R_1 = the resultant force; for horizontal
$= \sqrt{Q^2 + S^2}$, for vertical engines $R_1 = Q + S$, when the pressur
rank is in the same direction as the pressure of the shaft on its b
nd $R_1 = Q - S$ when the steam pressure tends to lift the shaft
earings. Using empirical values for the work of friction per squ
f projected area, taken from dimensions of crank-pins in marine
e finds the formula for length of shaft-journals $l = .0000325fR$
ecommends that to cover the defects of workmanship, neglect o
nd the introduction of dust, f be taken at .16 or even greater.
hat 500 lbs. per sq. in. of projected area may be allowed for steel or w
ron shafts in brass bearings with good results if a less pressure is no
ble without inconvenience. Marks says that the use of empirical ru
do not take account of the number of turns per minute has resulted
ings much too long for slow-speed engines and too short for hig
engines.

Whitham gives the same formula, with the coefficient .00002575.

Thurston says that the maximum allowable mean intensity of
may be, for all cases, computed by his formula for journals, $l = \frac{\ldots}{60,\ldots}$
by Rankine's, $l = \frac{P(V+20)}{44,800d}$, in which P is the mean total pressure in
V the velocity of rubbing surface in feet per minute, and d the dia
the shaft in inches. It must be borne in mind, he says, that the fricti
on the main bearing next the crank is the sum of that due the actio
piston on the pin, and that due that portion of the weight of wh
shaft and of pull of the belt which is carried there. The outboard
carries practically only the latter two parts of the total. The cra
journals will be made longer on one side, and perhaps shorter on th
than that of the crank-pin, in proportion to the work falling upon ea
to their respective products of mean total pressure, speed of rubbi
faces, and coefficients of friction.

Unwin says: Journals running at 150 revolutions per minute a
only one diameter long. Fan shafts running 150 revolutions per minu
journals six or eight diameters long. The ordinary empirical mode
portioning the length of journals is to make the length proportiona
diameter, and to make the ratio of length to diameter increase w
speed. For wrought-iron journals:

Revs. per min. =	50	100	150	200	250	500	1000
Length ÷ diam. =	1.2	1.4	1.6	1.8	2.0	3.0	5.0.

$\frac{l}{d} = .00\ldots$

Cast-iron journals may have $l \div d = 9/10$, and steel journals $l \div$
of the above values.

Unwin gives the following, calculated from the formula l -
which r is the crank radius in inches, and H.P. the horse-pow
to the crank-pin.

THEORETICAL JOURNAL LENGTH IN INCHES.

Load on Journal in pounds.	Revolutions of Journal per minute.					
	50	100	200	300	500	1000
1,000	.2	.4	.8	1.2	2.	4.
2,000	.4	.8	1.6	2.4	4.	8.
4,000	.8	1.6	3.2	4.8	8.	16.
5,000	1.0	2.	4.	6.	10.	20.
10,000	2.	4.	8.	12.	20.	40.
15,000	3.	6.	12.	18.	30.	....
20,000	4.	8.	16.	24.	40.	....
30,000	6.	12.	24.	36.	....	....
40,000	8.	16.	32.	....	....	....
50,000	10.	20.	40.	....	....	....

Applying these different formulæ to our six engines, we have:

Engine No.	1	2	3	4	5	6
Diam. cyl.	10	10	30	30	50	50
Horse-power	50	50	450	450	1,250	[illegible]
Revs. per min.	250	125	130	65	90	[illegible]
Mean pressure on crank-pin $= S$	3,299	3,299	23,185	23,185	58,905	[illegible]
Half wt. of fly-wheel and shaft $= Q$	268	536	5,968	11,936	26,470	[illegible]
Resultant press. on bearing $\sqrt{Q^2+S^2}=R_1$	3,310	3,335	23,924	26,194	64,580	[illegible]
Diam. of shaft journal	3.84	4.39	11.35	14.99	20.58	[illegible]
Length of shaft journal:						
Marks, $l = .0000325fR_1N(f=.10)$	5.38	2.71	20.87	11.07	37.78	[illegible]
Whitham, $l = .0000515fR_1R(f=10)$.	4.27	2.15	16.53	8.77	29.95	[illegible]
Thurston, $l = \dfrac{PV}{60{,}000d}$	3.61	1.82	14.00	7.43	25.36	[illegible]
Rankine, $l = \dfrac{P(V+20)}{44{,}800d}$	5.22	2.78	21.70	10.85	35.16	[illegible]
Unwin, $l = (.004R + 1)d$	7.68	6.59	17.25	16.36	27.99	[illegible]
Unwin, $l = \dfrac{0.4\ \text{H.P.}}{r}$	3.33	1.60	12.00	6.00	20.83	[illegible]
Average	4.92	2.99	17.05	10.00	29.54	[illegible]

If we divide the mean resultant pressure on the bearing by the projected area, that is, by the product of the diameter and length of the journal, using the greatest and smallest length out of the seven lengths for each journal given above, we obtain the pressure per square inch upon the bearing, as follows:

Engine No	1	2	3	4	5	6
Pressure per sq. in., shortest journal.	259	455	176	336	151	[illegible]
Longest journal	112	115	97	123	83	[illegible]
Average journal	175	254	124	202	106	[illegible]
Journal of length = diam	......	173	......	155	......	[illegible]

Many of the formulæ give for the long-stroke engines a length of journal less than the diameter, but such short journals are rarely used in practice. The last line in the above table has been calculated on the supposition that als of the long-stroke engines are made of a length equal to the

limensions of Corliss engines given by J. B. Stanwood (*Eng.*, June the lengths of the journals for engines of diam. of cyl. 10 to 20 in. ame as the diam. of the cylinder, and a little more than twice the the journal. For engines above 20 in. diam. of cyl. the ratio of diam. is decreased so that an engine of 30 in. diam. has a journal ;, its diameter being $14\frac{1}{16}$ in. These lengths of journal are greater e given by any of the formulæ above quoted.
hus appears to be a hopeless confusion in the various formulæ for shaft journals, but this is no more than is to be expected from the in the coefficient of friction, and in the heat-conducting power of n actual use, the coefficient varying from .10 (or even .16 as given) down to .01, according to the condition of the bearing surfaces fficiency of lubrication. Thurston's formula, $l = \frac{PV}{60,000d}$, reduces to $l = .000004363PR$, in which P = mean total load on journal, and lutions per minute. This is of the same form as Marks' and 's formulæ, in which, if f the coefficient of friction be taken at .10, cients of PR are, respectively, .0000065 and .00000515. Taking the these three formulæ, we have $l = .0000053PR$, if $f = .10$ or $l =$ R for any other value of f. The author believes this to be as safe as any for length of journals, with the limitation that if it brings f length of journal less than the diameter, then the length should equal to the diameter. Whenever with $f = .10$ it gives a length inconvenient or impossible of construction on account of limited n provision should be made to reduce the value of the coefficient i below .10 by means of forced lubrication, end play, etc., and to ay the heat, as by water-cooled journal-boxes. The value of P taken as the resultant of the mean pressure on the crank, and the ght on the bearing by the weight of the shaft, fly-wheel, etc., as l by the formula already given, viz., $R_1 = \sqrt{Q^2 + S^2}$ for horizontal nd $R_1 = Q + S$ for vertical engines.
six engines the formula $l = .0000053PR$ gives, with the limitation ng-stroke engines that the length shall not be less than the diam- ollowing:

o...........................	1	2	3	4	5	6
journal........	4.39	4.39	16.48	12.99	30.80	21.52
per square inch on journal..	196	173	128	155	102	171

-shafts with Centre-crank and Double-crank
In centre-crank engines, one of the crank-arms, and its adjoining alled the after journal, usually transmit the power of the engine rk to be done, and the journal resists both twisting and bending while the other journal is subjected to bending moment only. fter crank-journal the diameter should be calculated the same as erhung crank, using the formula for combined bending and twist- ent, $T_1 = B + \sqrt{B^2 + T^2}$, in which T_1 is the equivalent twisting B the bending moment, and T the twisting moment. This value be used in the formula diameter $= \sqrt[3]{\frac{5.1T}{S}}$. The bending mo- aken as the maximum load on piston multiplied by one fourth of h of the crank-shaft between middle points of the two journal if the centre crank is midway between the bearings, or by one listance measured parallel to the shaft from the middle of the to the middle of the after bearing. This supposes the crank- be a beam loaded at its middle and supported at the ends, but would make the bending moment only one half of this, consider- aft to be a beam secured or fixed at the ends, with a point of con- one fourth of the length from the end. The first supposition is but *since the bending moment will in any case be much less than ng moment, the resulting diameter will be but little greater than n's supposition is used.* For the forward journal, which is sub- *nding moment only,* **diameter of shaft** $= \sqrt[3]{\frac{10.2B}{S}}$.

is the maximum bending moment and S the safe shearing str metal per square inch.

For our six engines, assuming them to be centre-crank engi sidering the crank-shaft to be a beam supported at the ends a the middle, and assuming lengths between centres of shaft given below, we have:

Engine No.	1	2	3	4	5
Length of shaft, assumed, inches, L	20	24	48	60	76
Max. press. on crank-pin, P	7,854	7,854	70,686	70,686	196,[illegible]
Max. bending moment, $B = \frac{1}{4}PL$, inch-lbs.	39,270	49,637	848,232	1,060,290	3,729,[illegible]
Twisting moment, T	47,124	94,248	1,060,290	2,120,580	4,712,[illegible]
Equiv. twisting moment, $B + \sqrt{B^2 + T^2}$	101,000	156,000	2,208,000	3,430,000	9,740,[illegible]
Diameter of after journal, $d = \sqrt[3]{\frac{5.1T_1}{8000}}$	3.98	4.60	11.15	13.00	18.[illegible]
Diam. of forward journal, $d_1 = \sqrt[3]{\frac{10.2B}{8000}}$	3.68	3.99	10.28	11.16	16.[illegible]

The lengths of the journals would be calculated in the same the case of overhung cranks, by the formula $l = .000053fPR$, the resultant of the mean pressure due to pressure of steam o and the load of the fly-wheel, shaft, etc., on each of the t Unless the pressures are equally divided between the two calculated lengths of the two will be different; but it is usual to make them both of the same length, and in no case to mak less than the diameter. The diameters also are usually made two journals, using the largest diameter found by calculation.

The crank-pin for a centre crank should be of the same len overhung crank, since the length is determined from cons heating, and not of strength. The diameter also will usually since it is made great enough to make the pressure per square projected area (product of length by diameter) small enough free lubrication, and the diameter so calculated will be great quired for strength.

Crank-shaft with Two Cranks coupled at 9 whole power of the engine is transmitted through the after jo after crank-shaft, the greatest twisting moment is equal to 1. maximum twisting moment due to the pressure on one of the If T = the maximum twisting moment produced by the steam one of the pistons, then T_1 the maximum twisting moment on t of the crank-shaft, and on the line-shaft, produced when each an angle of 45° with the centre line of the engine, is $1.414T$. this value in the formula for diameter to resist simple tors $\sqrt[3]{\frac{5.1T}{S}}$, we have $d = \sqrt[3]{\frac{5.1 \times 1.414T}{S}}$, or $d = 1.932\sqrt[3]{\frac{T}{S}}$, i the maximum twisting moment produced by one of the piston eter in inches, and S = safe working shearing strength of t For the forward journal of the after crank, and the after jo forward crank, the torsional moment is that due to the pressu on the forward piston only, and for the forward journal of crank, if none of the power of the engine is transmitted t [illegible]onal moment is zero, and its diameter is to be calculate [illegible] only.

Combin[illegible]orsion and Flexure.—Let B, [illegible] the forward crank due to maxim either [illegible] ard piston, B_2 = bending moment on either journal of the after crank to maximum pressure on after piston, T_1 = maximum twisting moment fter journal of forward crank, and T_2 = maximum twisting moment on journal of after crank due to pressure on the after piston.

en equivalent twisting moment on after journal of forward crank = B_1 $B_1^2 + T_1^2$.

forward journal of after crank = $B_2 + \sqrt{B_2^2 + T_1^2}$.

after journal of after crank = $B_2 + \sqrt{B_2^2 + (T_1 + T_2)^2}$.

ese values of equivalent twisting moment are to be used in the formula diameter of journals $d = \sqrt[3]{\frac{5.1T}{S}}$. For the forward journal of the ard crank-shaft $d = \sqrt[3]{\frac{10.2B_1}{S}}$.

is customary to make the two journals of the forward crank of one eter, viz., that calculated for the after journal.

or a Three-cylinder Engine with cranks at 120°, the greatest ting moment on the after part of the shaft, if the maximum pressures he three pistons are equal, is equal to twice the maximum pressure on one piston, and it takes place when two of the cranks make angles of with the centre line, the third crank being at right angles to it. (For de-stration, see Whitham's "Steam-engine Design," p. 252.) For combined ion and flexure the same method as above given for two crank engines dopted for the first two cranks; and for the third, or after crank, if all power of the three cylinders is transmitted through it, we have the ivalent twisting moment on the forward journal = $B_2 + \sqrt{B_2^2+(T_1+T_2)^2}$, on the after journal = $B_3 + \sqrt{B_3^2 + (T_1 + T_2 + T_3)^2}$, B_3 and T_3 being pectively the bending and twisting moments due to the pressure on the d piston.

rank-shafts for Triple-expansion Marine Engines, ording to an article in *The Engineer*, April 25, 1890, should be made er than the formulæ would call for, in order to provide for the stresses to the racing of the propeller in a sea-way, which can scarcely be cal-ted. A kind of unwritten law has sprung up for fixing the size of a k-shaft, according to which the diameter of the shaft is made about D, where D is the diameter of the high-pressure cylinder. This is for shafts. When the speeds are high, as in war-ships, and the stroke t, the formula becomes $0.4D$, even for hollow shafts.

he Valve-stem or Valve-rod.—The valve-rod should be designed ove the valve under the most unfavorable conditions, which are when stem acts by thrusting, as a long column, when the valve is unbalanced lanced valve may become unbalanced by the joint leaking) and when it perfectly lubricated. The load on the valve is the product of the area the greatest unbalanced pressure upon it per square inch, and the co-ient of friction may be as high as 20%. The product of this coefficient the load is the force necessary to move the valve, which equals the imum thrust on the valve-rod. From this force the diameter of the e-rod may be calculated by Hodgkinson's formula for columns. An irical formula given by Seaton is: Diam. of rod = $d = \sqrt{\frac{lbp}{F}}$, in which length and b = breadth of valve, in inches; p = maximum absolute sure on the valve in lbs. per sq. in., and F a coefficient whose values are, ron: long rod 10,000, short 12,000; for steel: long rod 12,000, short 14,500. hitham gives the short empirical rule: Diam. of valve-rod = 1/30 diam. l. = 1/8 diam of piston-rod.

ze of Slot-link. (Seaton.)—Let D be the diam. of the valve rod

$$D = \sqrt{\frac{lbp}{12,000}};$$

Diameter of block-pin when overhung	= D.
" " " secured at both ends	= 0.75 × D.
" *eccentric-rod pins*	= 0.7 ×
" *suspension-rod pins*	= 0.55 ×
" " " pin when overhung	= 0.75 ×

Breadth of link	$= 0.8$ to $0.9 \times D$.
Length of block	$= 1.8$ to $1.6 \times D$.
Thickness of bars of link at middle	$= 0.7 \times D$.

If a single suspension rod of round section, its diameter = 0.7 ×
If two suspension rods of round section, their diameter = 0.55 ×

Size of Double-bar Links.—When the distance between eccentric pins = 6 to 8 times throw of eccentrics (throw = eccen half-travel of valve at full gear) D as before:

Depth of bars	$= 1.25 \times D + \frac{3}{4}$ in.
Thickness of bars	$= 0.5 \times D + \frac{1}{4}$ in.
Length of sliding-block	$= 2.5$ to $3 \times D$.
Diameter of eccentric-rod pins	$= 0.8 \times D + \frac{1}{4}$ in.
" centre of sliding-block	$= 1.3 \times D$.

When the distance between eccentric-rod pins = 5 to 5½ times eccentrics:

Depth of bars	$= 1.25 \times D + \frac{1}{2}$ in.
Thickness of bars	$= 0.5 \times D + \frac{1}{4}$ in.
Length of sliding-block	$= 2.5$ to $3 \times D$.
Diameter of eccentric-rod pins	$= 0.75 \times D$.

Diameter of eccentric bolts (top end) at bottom of thread = 0.42 of iron, and $0.38 \times D$ when of steel.

The Eccentric.—Diam. of eccentric-sheave = $2.4 \times$ throw of $+ 1.2 \times$ diam. of shaft. D as before

Breadth of the sheave at the shaft	$= 1.15 \times D +$
Breadth of the sheave at the strap	$= D + 0.6$ in
Thickness of metal around the shaft	$= 0.7 \times D +$
Thickness of metal at circumference	$= 0.6 \times D +$
Breadth of key	$= 0.7 \times D +$
Thickness of key	$= 0.25 \times D +$
Diameter of bolts connecting parts of strap	$= 0.6 \times D +$

THICKNESS OF ECCENTRIC-STRAP.

When of bronze or malleable cast iron:

Thickness of eccentric-strap at the middle	$= 0.4 \times D +$
" " " " " " sides	$= 0.3 \times D +$

When of wrought iron or cast steel:

Thickness of eccentric-strap at the middle	$= 0.4 \times D$
" " " " " " sides	$= 0.27 \times D$

The Eccentric-rod.—The diameter of the eccentric-rod and at the eccentric end may be calculated in the same way as connecting-rod, the length being taken from centre of strap t pin. Diameter at the link end = $0.8D + 0.2$ inch.

This is for wrought-iron; no reduction in size should be made f Eccentric-rods are often made of rectangular section.

Reversing-gear should be so designed as to have more tha strength to withstand the strain of both the valves and their same time under the most unfavorable circumstances; it will the stiffness requisite for good working.

Assuming the work done in reversing the link-motion, W, to be due to overcoming the friction of the valves themselves through travel, then, if T be the travel of valves in inches; for a compoun

$$W = \frac{T}{12}\left(\frac{l \times b \times p}{5}\right) + \frac{T}{12}\left(\frac{l^1 \times b^1 \times p^1}{5}\right);$$

l^1, b^1 and p^1 being length, breadth and maximum steam-pressur of the second cylinder; and for an expansive engine

$$W = 2 \times \frac{T}{12}\left(\frac{l \times b \times p}{5}\right), \text{ or } \frac{T}{30}(l \times b \times p).$$

To provide for the friction of link-motion, eccentrics and of for abnormal ... of the same, take the work at one ... above ...

strain at any part of the gear having motion when reversing. rk so found by the space moved through by that part in feet; is the strain in pounds; and the size may be found from the s of construction for any of the parts of the gear. (Seaton.)

rames or Bed-plates.—No definite rules for the design mes have been given by authors of works on the steam-engine. ons are left to the designer who uses "rule of thumb," or xisting engines. F. A. Halsey (*Am. Mach.*, Feb. 14, 1895) has parison of proportions of the frames of horizontal Corliss veral builders. The method of comparison is to compute from nents the number of square inches in the smallest cross-sec- frame, that is, immediately behind the pillow-block, also to total maximum pressure upon the piston, and to divide the ty by the former. The result gives the number of pounds n the piston allowed for each square inch of metal in the finds that the number of pounds per square inch of smallest ame ranges from 217 for a 10×30-in. engine up to 575 for a A 30×60-inch engine shows 350 lbs., and a 32-inch engine een running for many years shows 667 lbs. Generally the se with the size of the engine, and more cross-section of metal th relatively long strokes than with short ones.

bove Mr. Halsey formulates the general rule that in engines speed, and having strokes up to one and one-half times the the cylinder, the load per square inch of smallest section a 10-inch engine 300 pounds, which figure should be increased res up to 500 pounds for a 30-inch cylinder of same relative high speeds or for longer strokes the load per square inch uced.

FLY-WHEELS.

n of a fly-wheel is to store up and to restore the periodical fluc- ergy given to or taken from an engine or machine, and thus ximately constant the velocity of rotation. Rankine calls the

the coefficient of fluctuation of speed or of unsteadiness, in

e mean actual energy, and ΔE the excess of energy received or rmed, above the mean, during a given interval. The ratio of excess or deficiency of energy ΔE to the whole energy exerted or revolution General Morin found to be from 1/6 to ¼ for r engines using expansion; the shorter the cut-off the higher or a pair of engines with cranks coupled at 90° the value of the ¼, and for three engines with cranks at 120°, 1/12 of its value inder engines. For tools working at intervals, such as punch- and plate-cutting machines, coining-presses, etc., ΔE is nearly hole work performed at each operation.

reduces the coefficient $\frac{\Delta E}{2E_0}$ to a certain fixed amount, being r ordinary machinery, and 1/50 or 1/60 for machinery for fine

reciprocal of the intended value of the coefficient of fluctua- ΔE the fluctuation or energy, I the moment of inertia of the e, and a_0 its mean angular velocity, $I = \frac{mg\Delta E.}{a_0}$. As the rim of usually heavy in comparison with the arms, I may be taken in which W = weight of rim in pounds, and r the radius of the $V = \frac{mg\Delta E}{a_0^2 r^2} = \frac{mg\Delta E}{v^2}$, if v be the velocity of the rim in feet per usual mean radius of the fly-wheel in steam-engines is from imes the length of the crank. The ordinary values of the prod- nit of time being the second, lie between 1000 and 2000 feet. m Rankine, *S. E.*, p. 62.)

ives for engines with automatic valve-gear W = 250,000

A = area of piston in square inches, S = stroke in '

ssure in lbs. per sq. in., R = revolutions per min *wheel in feet.* Thurston also gives for ordi

an ordinary load, and 4 per cent when heavily loaded.
ordinary purposes, $V = 2\frac{1}{2}$ to 3 per cent. For good
purposes, such as cotton-spinning, the variation shoul
per cent.

F. M. Rites (Trans. A. S. M. E., xiv. 100) develops a ne
of rim, viz., $W = \frac{C \times \text{I.H.P.}}{R^3 D^2}$, and weight of rim per hor
which C varies from 10,000,000,000 to 20,000,000,000; also
of C, he obtains for the energy of the fly-wheel $\frac{Mv}{2}$
$\frac{C \times \text{H.P.} (3.14)^2 D^2 R^2}{R^3 D^2 \times 64.4 \times 3600} = \frac{850,000 \text{ H.P.}}{R}$. Fly-wheel energy

The limit of variation of speed with such a weight of
power per fraction of revolution is less than .0023.

The value of the constant C given by Mr. Rites was de
of the Westinghouse single-acting engines used for el
double-acting engines in ordinary service a value of $C =$
probably be ample.

From these formulæ it appears that the weight of the
horse-power should vary inversely with the cube of the
square of the diameter.

J. B. Stanwood (*Eng'g*, June 12, 1891) says: Whene
lowest piston-speed probable for an engine of a certai
weight for that speed approximates closely to the formu

$$W = 700,000 \frac{d^2 s}{D^2 R^2}.$$

W = weight in pounds, d = diameter of cylinder in i
inches, D = diameter of wheel in feet, R = revolutions
sponding to 480 feet piston-speed.

In a Ready Reference Book published by Mr. Stanwo
he gives the same formula, with coefficients as follows:
gines, ordinary duty, 350,000; same, electric-lighting, 70
high-speed engines, 1,000,000; for Corliss engines, ord
electric-lighting 1,000,000.

Thurston's formula above given, $W = \frac{aAS}{R^2 D^2}$, with $a =$

engines operating—
Hammering and crushing machinery.............. $d = 5$
Pumping and shearing machinery.................. $d = 20$ to 30
Weaving and paper-making machinery............ $d = 40$
Milling machinery.. $d = 50$
Spinning machinery... $d = 50$ to 100
Ordinary driving-engines (mounted on bed-plate),
belt transmission... $d = 35$
Gear-wheel transmission.................................. $d = 50$

Theiss's formula for weight of fly-wheel in pounds is $W = i \times \frac{d \times \text{I.H.P.}}{V^2 \times n}$,

re d is the coefficient of steadiness, V the mean velocity of the fly-el rim in feet per second, n the number of revolutions per minute, i = efficient obtained by graphical solution, the values of which for dif-nt conditions are given in the following table. In the lines under "cut-p means "compression to initial pressure," and O "no compression":

VALUES OF i. SINGLE-CYLINDER NON-CONDENSING ENGINES.

speed, ft. per min.	Cut-off, 1/6.		Cut-off, ¼.		Cut-off, ⅓.		Cut-off, ½.	
	Comp. p	O	Comp. p	O	Comp. p	O	Comp. p	O
200	272,690	218,580	242,010	209,170	220,760	201,920	193,340	182,840
400	240,810	187,430	208,200	179,460	188,510	170,040	174,630	167,860
600	194,670	145,400	168,590	136,460	165,210	146,610	....	....
800	158,200	108,690	162,070	135,260	....	....	....	....

SINGLE-CYLINDER CONDENSING ENGINES.

per min.	Cut-off, ⅛.		Cut-off, 1/6.		Cut-off, ¼.		Cut-off, ⅓.		Cut-off, ½.	
	Comp. p	O	Comp. p	O	Comp. p	O	Comp. p	O	Comp. p	O
0	265,560	176,560	234,160	173,660	204,210	167,140	189,600	161,830	172,690	156,990
0	194,550	117,870	174,380	118,350	164,720	133,080	174,630	151,680	....	....
0	148,780	140,090	....	....	....	....	....	....	....	....

TWO-CYLINDER ENGINES, CRANKS AT 90°.

speed, ft. per min.	Cut-off, 1/6.		Cut-off, ¼.		Cut-off, ⅓.		Cut-off, ½.	
	Comp. p	O	Comp. p	O	Comp. p	O	Comp. p	O
00	71,980	Mean 60,140	59,420	Mean 54,340	49,272	Mean 50,000	37,920	Mean 36,950
00	70,160		57,000		49,150		35,500	
00	70,040		57,480		49,220			
00	70,040		60,140					

THREE-CYLINDER ENGINES, CRANKS AT 120°.

per Min.	Cut-off, 1/6.		Cut-off, ¼.		Cut-off, ⅓.		Cut-off, ½.	
	Comp. p	O	Comp. p	O	Comp. p	O	Comp. p	O
7	33,810	32,240	33,810	35,500	34,540	33,450	35,2[illegible]	[illegible]
	30,190	31,570	35,140	33,810	36,470	32,850	33.[illegible]	[illegible]

As a mean value of i for these engines we may use 33

Centrifugal Force in Fly-wheels.—Let W = w pounds; R = mean radius of rim in feet; r = revolutions 32.16; v = velocity of rim in feet per second = $2\pi Rr \div 60$.

Centrifugal force of whole rim $= F = \frac{Wv^2}{gR} = \frac{4W\pi^2Rr^2}{3600g} =$

The resultant, acting at right angles to a diameter of ha tends to disrupt one half of the wheel from the other half, a the section of the rim at each end of the diameter. The resu

radial forces taken at right angles to the diameter is $1 \div \frac{1}{2}\pi$

of these forces; hence the total force F is to be divided = 6.2832 to obtain the tensile strain on the cross-section of strain on the cross-section $= S = .00005427 WRr^2$. The rim of cast iron 1 inch square in section is $2\pi R \times 3.125 =$ whence strain per square inch of sectional area of rim $= S$ $= .0002664D^2r^2 = .0000270V^2$, in which D = diameter of whe is velocity of rim in feet per minute. $S_1 = .0972v^2$, if v is ta second.

For wrought iron.......... $S_1 = .0011366R^2r^2 = .0002842D^2r$
For steel......... .... $S_1 = .0011593R^2r^2 = .0002901D^2r$
For wood................. $S_1 = .0000888R^2r^2 = .0000222D^2r$

The specific gravity of the wood being taken at 0.6 = 37. or 1/12 the weight of cast iron.

Example.—Required the strain per square inch in the ri wheel 30 ft. diameter, 60 revolutions per minute.

Answer. $15^2 \times 60^2 \times .0010$

Required the strain per square inch in a cast-iron whe mile a minute. *Answer.* $.000027 \times 5280^2 = 752.7$ lbs.

In cast-iron fly-wheel rims, on account of their thickness, in securing soundness, and a tensile strength of 10,000 lbs much as can be assumed with safety. Using a factor of saf maximum allowable strain in the rim of 1000 lbs. per sq. sponds to a rim velocity of 6085 ft. per minute.

For any given material, as cast iron, the strength to resist depends only on the velocity of the rim, and not upon its b

Chas. E. Emery (*Cass. Mag.*, 1892) says: By calculation of the arms is available to strengthen the rim, or a trifl wheel centres are relatively large. The arms, however, are verse strains, from belts and from changes of speed, and th no certainty that the arms and rim will be adjusted so together in resisting disruption, so the plan of considering and making it strong enough to resist disruption by centrif safe limits, as is assumed in the calculations above, is the s

It does not appear that fly-wheels of customary constr unsafe at the comparatively low speeds now in comm materials are used in construction. The cause of rupture have failed is usually either the "running away" of the en be caused by the breaking or slackness of a governor-b design or defective materials of the fly-wheel.

Chas. T. Porter (Trans. A. S. M. E., xiv, 808) states th bursting of a fly-wheel with a solid rim in a high-speed eng attributes the bursting of wheels built in segments to ins of the flanges and bolts by which the segments are held to Thurston, "Manual of the Steam-engine," Part II, page 41

Arms of Fly-wheels and Pulleys. — Profes *Mach.*, July 30, 1891) gives the following formula for arms section of cast-iron wheels:

W = load in pounds acting on one arm; S = strain on inch of width, taken at 56 for single and 112 for double be belt in inches; n = number of arms; L = length of arm in

of arm at hub; d = depth of arm at hub, both in

$h = \frac{WL}{...d^2}$. The breadth of the arm is its least dimensi ellipse, and the depth the major axis. This formul ...fety of 10...

he formula, first assume some depth for the arm, and calculate d breadth to go with it. If it gives too round an arm, assume a little greater, and repeat the calculation. A second trial will ays give a good section.

of the arms at the hub having been calculated, they may be educed at the rim end. The actual amount cannot be calculated, re too many unknown quantities. However, the depth and be reduced about one third at the rim without danger, and this well-shaped arm.

re often cast in halves, and bolted together. When this is done t care should be taken to provide sufficient metal in the bolts. to be the very weakest point in such pulleys. The combined area at each joint should be about 28/100 the cross-section of the pul point. (Torrey.)

ves $d = 0.6337\sqrt[3]{\frac{BD}{n}}$ for single belts;

$$d = 0.798\sqrt[3]{\frac{BD}{n}} \text{ for double belts;}$$

e diameter of the pulley, and B the breadth of the rim, both in ese formulæ are based on an elliptical section of arm in which $d = 2.5b$ on a width of belt $= 4/5$ the width of the pulley rim, a driving force transmitted by the belt of 56 lbs. per inch of width belt and 112 lbs. for a double belt, and a safe working stress of 2250 lbs. per square inch.

ey's formula we make $b = 0.4d$, it reduces to

$$b = \sqrt[3]{\frac{WL}{187.5}};\quad d = \sqrt[3]{\frac{WL}{12}}.$$

—Given a pulley 10 feet diameter; 8 arms, each 4 feet long; face, ide; belt, 30 inches: required the breath and depth of the arm at ccording to Unwin,

$$6337\sqrt[3]{\frac{BD}{n}} = 0.633\sqrt[3]{\frac{36 \times 120}{8}} = 5.16 \text{ for single belt, } b = 2.06;$$

$$98\sqrt[3]{\frac{BD}{n}} = 0.798\sqrt[3]{\frac{36 \times 120}{8}} = 6.50 \text{ for double belt, } b = 2.60.$$

g to Torrey, if we take the formula $b = \frac{WL}{30d^2}$ and assume $d = 5$ es, respectively, for single and double belts, we obtain $b = 1.08$ spectively, or practically only one half of the breadth according nd, since transverse strength is proportional to breadth, an arm lf as strong.

formula is said to be based on a factor of safety of 10, but this be only apparent and not real, since the assumption that the ch arm is equal to the strain on the belt divided by the number to say the least, inaccurate. It would be more nearly correct to e strain of the belt is divided among half the number of arms. es the same assumption in developing his formula, but says it is gh sense true, and that a large factor of safety must be allowed. re takes the low figure of 2250 lbs. per square inch for the safe rength of cast iron. Unwin says that his equations agree well ce.

ers of Fly-wheels for Various Speeds.—If 6000 feet *be the maximum velocity of rim allowable, then* 6000 ÷ *evolutions per minute*, and D = diameter of whe

$$\frac{6000}{\pi R} = \frac{1910}{R}.$$

MAXIMUM DIAMETER OF FLY-WHEEL ALLOWABLE FOR DIFFERENT N
OF REVOLUTIONS.

Revolutions per minute.	Assuming Maximum Speed of 5000 feet per minute.		Assuming Maximum of 6000 feet per m	
	Circum. ft.	Diam. ft.	Circum. ft.	Dia
40	125	39.8	150.	
50	100	31.8	120.	
60	83.3	26.5	100.	
70	71.4	22.7	85.72	
80	62.5	19.9	75.00	
90	55.5	17.7	66.66	
100	50.	15.9	60.00	
120	41.67	13.3	50.00	
140	35.71	11.4	42.86	
160	31.25	9.9	37.5	
180	27.77	8.8	33.33	
200	25.00	8.0	30.00	
220	22.73	7.2	27.27	
240	20.83	6.6	25.00	
260	19.23	6.1	23.08	
280	17.86	5.7	21.43	
300	16.66	5.3	20.00	
350	14.29	4.5	17.14	
400	12.5	4.0	15.00	
450	11.11	3.5	13.33	
500	10.00	3.2	12.00	

Strains in the Rims of Fly-band Wheels Produ Centrifugal Force. (James B. Stanwood, Trans. A. S. M. E., —Mr. Stanwood mentions one case of a fly-band wheel where the velocity on a 17′ 9″ wheel is over 7500 ft. per minute.

In band saw-mills the blade of the saw is operated successf wheels 8 and 9 ft. in diameter, at a periphery velocity of 9000 to 10, minute. These wheels are of cast iron throughout, of heavy thick a large number of arms.

In shingle-machines and chipping-machines where cast-iron disks 5 ft. in diameter are employed, with knives inserted radially, th frequently 10,000 to 11,000 ft. per minute at the periphery.

If the rim of a fly-wheel alone be considered, the tensile strain per square inch of the rim section is $T = \frac{V^2}{10}$ nearly, in which V in feet per second; but this strain is modified by the resistance of which prevent the uniform circumferential expansion of the rim, a bending as well as a tensile strain. Mr. Stanwood discusses the band-wheels due to transverse bending of a section of the rim pair of arms.

When the arms are few in number, and of large cross-section will be strained transversely to a greater degree than with a great of lighter arms. To illustrate the necessary rim thicknesses f rim velocities, pulley diameters, number of arms, etc., the follo is given, based upon the formula

$$t = \frac{.475d}{N^2\left(\frac{F}{V^2} - \frac{1}{10}\right)},$$

in which t = thickness of rim in inches, d = diameter of pulle N = *number of arms*, V = velocity of rim in feet per second, *reatest st* ounds per square inch to which any fibre *value* at 6000 lbs. per sq. in.

Thickness of Rims in Solid Wheels.

r of in s.	Velocity of Rim in feet per second.	Velocity of Rim in feet per minute.	No. of Arms.	Thickness in inches.
	50	3,000	6	2/10
	88	5,280	6	15/32
	88	5,280	6	15/16
	184	11,040	16	2⅛
	184	11,040	36	⅞

nit of rim velocity for all wheels be assumed to be 88 ft. per sec- l to 1 mile per minute, $F = 6000$ lbs., the formula becomes

$$t = \frac{.475d}{.67N^2} = 0.7\frac{d}{N^2}.$$

heels are made in halves or in sections, the bending strain may to make t greater than that given above. Thus, when the joint f way between the arms, the bending action is similar to a beam simply at the ends, uniformly loaded, and t is 50% greater. Then la becomes

$$t = \frac{.712d}{N^2\left(\frac{F}{V^2} - \frac{1}{10}\right)}.$$

xed maximum rim velocity of 88 ft. per second and $F = 6000$ lbs.,

In segmental wheels it is preferable to have the joints opposite

Wheels in halves, if very thin rims are to be employed, should le arms along the line of separation.

n should be given to the proportions of large receiving and tight- eys. The thickness of rim for a 48-in. wheel (shown in table) with ocity of 88 ft. per second, is 15/16 in. Many wrecks have been the failure of receiving or tightening pulleys whose rims have beer Fly-wheels calculated for a given coefficient of steadiness are fre- ghter than the minimum safe weight. This is true especially of els. A rough guide to the minimum weight of wheels can be de- n our formulæ. The arms, hub, lugs, etc., usually form from one one third the entire weight of the wheel. If b represents the face in inches, the weight of the rim (considered as a simple annular be $w = .82dtb$ lbs. If the limit of speed is 88 ft. per second, then heels $t = 0.7d \div N^2$. For sectional wheels (joint between arms) $\div N^2$. Weight of rim for solid wheels, $w = .57d^2b \div N^2$ in pounds. rim in sectional wheels with joints between arms, $w = .86d^2b \div$ nds. Total weight of wheel: for solid wheel, $W = .76d^2b \div N^2$ to N^2, in pounds. For segmental wheels with joint between arms, $^2b \div N^2$ to $1.3d^2b \div N^2$, in pounds.

bject is further discussed by Mr. Stanwood, in vol. xv., and by ano Lanza, in vol. xvi., Trans. A. S. M. E.)

oden Rim Fly-wheel, built in 1891 for a pair of Corliss en- ne Amoskeag Mfg. Co.'s mill, Manchester, N. H., is described by ing in Trans. A. S. M. E., xiii. 618. It is 30 ft. diam. and 108 in. face. s 12 inches thick, and is built up of 44 courses of ash plank, 2, 3, es thick, reduced about ⅛ inch in dressing, set edgewise, so as to ts, and glued and bolted together. There are two hubs and two ns, 12 in each, all of cast iron. The weights are as follows:

ght (calculated) of ash rim........................	31,855 lbs.
of 24 arms (foundry 45,020)..................	40,349 "
" 2 hubs (" 35,030)	31,394± "
ter-weights in 6 arms	664 "
l, excluding bolts and screws..................	104,262± "

was tested at 76 revs. per min., being a surface speed minute.

Mr. Manning discusses the relative safety of cast iron and of wheels as follows: As for safety, the speeds being the same cases, the hoop tension in the rim per unit of cross-section would be as the weight per cubic unit; and its capacity to stand the strain di the tensile strength per square unit; therefore the tensile strength by the weights will give relative values of different materials. C weighing 450 lbs. per cubic foot and with a tensile strength of 1,44 per square foot would give a value of 1,440,000 ÷ 450 = 3200, whilst which the rim was made, weghing 34 lbs. per cubic foot, and with lbs. tensile strength per square foot, gives a result 1,152,000 ÷ 34 and 33,882 ÷ 3200 = 10.58, or the wood-rimmed pulley is ten tim than the cast-iron when the castings are good. This would allow th rimmed pulley to increase its speed to $\sqrt{10.58} = 3.25$ times that of cast-iron one with equal safety.

Wooden Fly-wheel of the Willimantic Linen Co. trated in *Power*, March, 1893.)—Rim 28 ft. diam., 110 in. face. Th carried upon three sets of arms, one under the centre of each belt, arms in each set.

The material of the rim is ordinary whitewood, ⅞ in. in thickness, segments not exceeding 4 feet in length, and either 5 or 8 inches i These were assembled by building a complete circle 13 inches in wid with the 8 inch inside and the 5-inch outside, and then beside it ano cle with the widths reversed, so as to break joints. Each piece a added was brushed over with glue and nailed with three-inch wire the pieces already in position. The nails pass through three and fourth thickness. At the end of each arm four 14-inch bolts sec rim, the ends being covered by wooden plugs glued and driven into of the wheel.

Wire-wound Fly-wheels for Extreme Speeds. (*Eng* August 2, 1890.)—The power required to produce the Mannesmann very large, varying from 2000 to 10,000 H.P., according to the dimen the tube. Since this power is only needed for a short time (it take to 45 seconds to convert a bar 10 to 12 ft. long and 4 in. in diamete tube), and then some time elapses before the next bar is ready, an e 1200 H.P. provided with a large fly-wheel for storing the energy wi power enough for one set of rolls. These fly-wheels are so large a such great speeds that the ordinary method of constructing them c followed. A wheel at the Mannesmann Works, made in Komotau, l in the usual manner, broke at a tangential velocity of 125 ft. pe The fly-wheels designed to hold at more than double this speed co cast-iron hub to which two steel disks, 20 ft. in diameter, are bolted the circumference of the wheel thus formed 70 tons of No. 5 wire a under a tension of 50 lbs. In the Mannesmann Works at Landor such a wheel makes 240 revolutions a minute, corresponding to a t velocity of 15,080 ft. or 2.85 miles per minute.

THE SLIDE-VALVE.

Definitions.—*Travel* = total distance moved by the valve.

Throw of the Eccentric = eccentricity of the eccentric = distance centre of the shaft to the centre of the eccentric disk = ½ the tra valve. (Some writers use the term "throw" to mean the whole the valve.)

Lap of the valve, also called outside lap or steam-lap = distance or steam edge of the valve extends beyond or laps over the stea the port when the valve is in its central position.

Inside lap, or *exhaust-lap* = distance the inner or exhaust ed valve extends beyond or laps over the exhaust edge of the port valve is in its central position. The inside lap is sometimes mad even negative, in which latter case the distance between the ed valve and the edge of the port is sometimes called exhaust clea *inside clearance*.

Lead of the valve = the distance the steam-port is opened when t is on its centre and the piston is at the beginning of the stroke.

Lead-angle = the angle between the position of the crank wh begins to be opened and its position when the piston is at the stroke.

valve [illegible] ve lead when the steam-port opens be

stroke. If the piston begins its stroke before the admission of
ns the valve is said to have negative lead, and its amount is the
edge of the valve over the edge of the port at the instant when
stroke begins.
le = the angle through which the eccentric must be rotated to
steam edge to travel from its central position the distance of the

advance of the eccentric = lap-angle + lead angle.
dvance = lap + lead.

of Lap, Lead, etc., upon the Steam Distribution.—
e-travel 2¾ in., lap ¾ in., lead 1/16 in., exhaust-lap ⅛ in., re-
nk position for admission, cut-off, release and compression, and
ort-opening. (Halsey on Slide-valve Gears.) Draw a circle of
fh = travel of valve. From O the centre set off Oa = lap and ab
ect perpendiculars Oe, ac, bd; then ec is the lap-angle and cd the
, measured as arcs. Set off $fg = cd$, the lead-angle, then Og is
n of the crank for steam admission. Set off $2ec + cd$ from h to i;
the crank-angle for cut-off, and $fk \div fh$ is the fraction of stroke
at cut-off. Set off Ol = exhaust-lap and draw lm; em is the
p angle. Set off $hn = ec + cd - em$, and On is the position of
elease. Set off $fp = ec + cd + em$, and Op is the position of crank
ession, $fo \div fh$ is the fraction of stroke completed at release, and
the fraction of the return stroke completed when compression
h, the throw of the eccentric, minus Oa the lap, equals ah the
port-opening.
e has neither lap nor lead, the line joining the centre of the eccen-

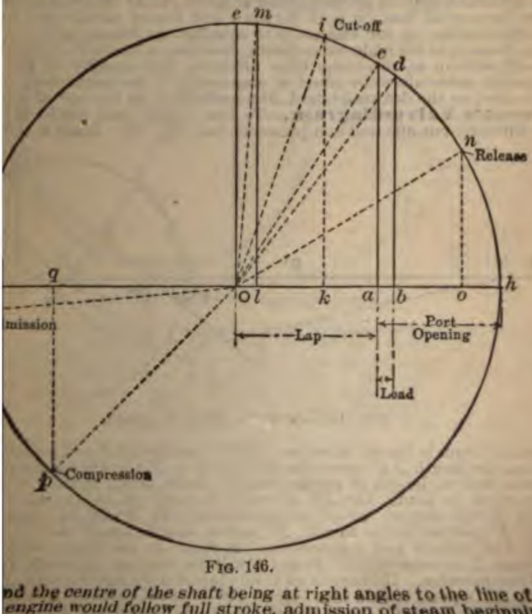


FIG. 146.

nd the centre of the shaft being at right angles to the line of the
engine would follow full stroke, admission of steam beginning at
ng of the stroke and ending at the end of the stroke.
to the valve enables us to cut off steam before the
ccentric being advanced on the shaft an amount
bles steam to be admitted at the beginning of the

before lap was added, and advancing it a further amount equal to the l[illegible] angle causes steam to be admitted before the beginning of the stroke.

Having given lap to the valve, and having advanced the eccentric on shaft from its central position at right angles to the crank, through angular advance = lap-angle and lead-angle, the four events, admi[illegible] cut-off, release or exhaust-opening, and compression or exhaust-clo[illegible] take place as follows: Admission, when the crank lacks the lead-angle having reached the centre; cut-off, when the crank lacks two lap-angles one lead-angle of having reached the centre. During the admission steam the crank turns through a semicircle less twice the lap-angle. greatest port-opening is equal to half the travel of the valve less the Therefore for a given port-opening the travel of the valve must be [illegible]creased if the lap is increased. When exhaust-lap is added to the valve delays the opening of the exhaust and hastens its closing by an angle rotation equal to the exhaust-lap angle, which is the angle through w[illegible] the eccentric rotates from its middle position while the exhaust edge of valve uncovers its lap. Release then takes place when the crank lacks lap-angle and one lead-angle minus one exhaust-lap angle of having rea[illegible] the centre, and compression when the crank lacks lap-angle + lead-ang[illegible] exhaust-lap angle of having reached the centre.

The above discussion of the relative position of the crank, piston, valve for the different points of the stroke is accurate only with a conn[illegible]ing-rod of infinite length.

For actual connecting-rods the angular position of the rod cause distortion of the position of the valve, causing the events to take place late in the forward stroke and too early in the return. The correction this distortion may be accomplished to some extent by setting the valve as to give equal lead on both forward and return stroke, and by alte[illegible] the exhaust-lap on one end so as to equalize the release and compress[illegible] F. A. Halsey, in his Slide-valve Gears, describes a method of equalizing cut-off without at the same time affecting the equality of the lead. designing slide-valves the effect of angularity of the connecting-rod sh[illegible] be studied on the drawing-board, and preferably by the use of a model.

Sweet's Valve-diagram.—To find outside and inside lap of v[illegible] for different cut-offs and compressions (see Fig. 147): Draw a circle w[illegible]

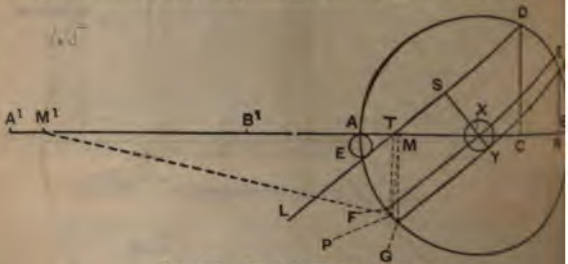


FIG. 147.—Sweet's Valve-diagram.

diameter equals travel of valve. Draw diameter *BA* and continue [illegible] so that the length AA^1 bears the same ratio to *XA* as the length of conn[illegible]ing-rod does to length of engine-crank. Draw small circle *E* with a di[illegible]eter equal to lead. Lay off *AC* so that ratio of *AC* to *AB* = c[illegible] parts of the stroke. Erect perpendicular *CD*. Draw *DL* tangent t[illegible] draw *XS* perpendicular to *DL*; *XS* is then outside lap of valve.

To find release and compression: If there is no inside lap, draw through *X* parallel to *DL*. *F* and *E* will be position of crank for re[illegible] and compression. If there is an inside lap, draw a circle about *X*, [illegible] radius *XY* equals inside lap. Draw HG tangent to this circle and pa[illegible] to *DL*; then *H* and *G* are crank position for release and compre[illegible] Draw *HN* and *MG*, then *AN* is piston position at release and *AM* [illegible]sition at compression, *AB* being considered stroke of engine.

[illegible] make c[illegible] alike on each stroke it is necessary to [illegible] [illegible] lap [illegible] of valve, and to decrease by the same [illegible]

p on back end of valve. To determine this amount, through M with $MM^1 = AA^1$, draw arc $M\,P$, from P draw PT perpendicular to AB, f is the amount to be added to inside lap on crank end, and to be d from inside lap on back end of valve, inside lap being XY.
e *Bilgram Valve Diagram*, see Halsey on Slide-valve Gears.
Zeuner Valve-diagram is given in most of the works on the ngine, and in treatises on valve-gears, as Zeuner's, Peabody's, and

Fig. 148.—Zeuner's Valve-diagram.

er's. The following is condensed from Holmes on the Steam-engine: e a circle, with radius OA equal to the half travel of the valve. measure off OB equal to the outside lap, and BC equal to the lead. he crank-pin occupies the dead centre A, the valve has already to the right of its central position by the space $OB + BC$. From C e perpendicular CE and join OE. Then will OE be the position d by the line joining the centre of the eccentric with the centre of nk-shaft at the commencement of the stroke. On the line OE as r describe the circle OCE; then any chords, as Oe, OE, Oe', will nt the spaces travelled by the valve from its central position when nk-pin occupies respectively the positions opposite to D, E, and F. the port is opened at all the valve must have moved from its central by an amount equal to the lap OB. Hence, to obtain the space by he port is opened, subtract from each of the arcs Oe, OE, etc., a equal to OB. This is represented graphically by describing from O a circle with radius equal to the lap OB; then the spaces fe, gE, ercepted between the circumferences of the lap-circle Bfe' and the rcle OCE, will give the extent to which the steam-port is opened. point k, at which the chord Ok is common to both valve and lap is evident that the valve has moved to the right by the a d is consequently just on the point of opening the s steam is admitted before the commencement of the s ccupies the position OH, and while the portion HA

lution still remains to be accomplished. When the crank-pin reaches position A, that is to say, at the commencement of the stroke, the p[illegible] already opened by the space $OC - OB = BC$, called the lead. From [illegible] point forward till the crank occupies the position OE the port continues open, but when the crank is at OE the valve has reached the farthest [illegible] of its travel to the right, and then commences to return, till when [illegible] position OF the edge of the valve just covers the steam-port, as is s[illegible] by the chord Oe', being again common to both lap and valve circles. [illegible] when the crank occupies the position OF the cut-off takes place and steam commences to expand, and continues to do so till the exhaust [illegible] For the return stroke the steam-port opens again at H' and closes at F'.

There remains the exhaust to be considered. When the line joining centres of the eccentric and crank-shaft occupies the position opposite OG at right angles to the line of dead centres, the crank is in the line [illegible] right angles to OE; and as OP does not intersect either valve-circle valve occupies its central position, and consequently closes the port [illegible] amount of the inside lap. The crank must therefore move through [illegible] an angular distance that its line of direction OQ must intercept a chord [illegible] the valve-circle OK equal in length to the inside lap before the port can [illegible] opened to the exhaust. This point is ascertained precisely in the [illegible] manner as for the outside lap, namely, by drawing a circle from centre [illegible] with a radius equal to the inside lap; this is the small inner circle [illegible] figure. Where this circle intersects the two valve-circles we get four p[illegible] which show the positions of the crank when the exhaust opens and c[illegible] during each revolution. Thus at Q the valve opens the exhaust on the [illegible] of the piston which we have been considering, while at R the exhaust c[illegible] and compression commences and continues till the fresh steam is [illegible] mitted at H.

Thus the diagram enables us to ascertain the exact position of the c[illegible] when each critical operation of the valve takes place. Making a r[illegible] these operations of one side of the piston, we have: Steam admitted be[illegible] the commencement of the stroke at H. At the dead centre A the val[illegible] already opened by the amount BC. At E the port is fully opened, valve has reached one end of its travel. At F steam is cut off, consequ[illegible] admission lasted from H to F. At P valve occupies central position, ports are closed both to steam and exhaust. At Q exhaust opened, [illegible] quently expansion lasted from F to Q. At K exhaust opened to maxi[illegible] extent, and valve reached the end of its travel to the left. At R exh[illegible] closed, and compression begins and continues till the fresh steam is adm[illegible] at H.

PROBLEM.—The simplest problem which occurs is the following: [illegible] the length of throw, the angle of advance of the eccentric, and the la[illegible] the valve, find the angles of the crank at which the steam is admitted cut off and the exhaust opened and closed. Draw the line OE, represe[illegible] the half-travel of the valve or the throw of the eccentric at the given [illegible] of advance with the perpendicular OG. Produce OE to K. On OE an[illegible] as diameters describe the two valve-circles. With centre and radii eq[illegible] the given laps describe the outside and inside lap-circles. Then the [illegible] section of these circles with the two valve-circles give points through [illegible] the lines OH, OF, OQ, and OR can be drawn. These lines give the re[illegible] positions of the crank.

Numerous other problems will be found in Holmes on the Steam-e[illegible] including problems in valve-setting and the application of the Zeune[illegible] gram to link motion and to the Meyer valve-gear.

Port Opening.—The area of port opening should be such that t[illegible] locity of the steam in passing through it should not exceed 6000 ft. p[illegible] The ratio of port area to piston area will then vary with the piston-sp[illegible] follows:

For speed of piston, ft. per min.	100	200	300	400	500	600	700	800	900	10[illegible]
Port area = piston area ×	.017	.033	.05	.067	.083	.1	.107	.133	.15	.16[illegible]

For a velocity of 6000 ft. per min.,

$$\text{Port area} = \frac{\text{sq. of diam. of cyl.} \times \text{piston speed}}{7639}.$$

The length of the port opening may be equal to or somewhat less [illegible] [illegible] of the cylinder, and the width = area of port opening ÷ [illegible] [illegible] exhaust ports should be wide[illegible] [illegible] bridge between [illegible] exhaust due to overtravel of v[illegible] [illegible] leak of ste[illegible] gives: Width of exhaust port = width of steam port + lve − width of bridge.

om Peabody's Valve-gears.)—The lead, or the amount that en when the engine is on a dead point, varies, with the type engine, from a very small amount, or even nothing, up to ⅜ nore. Stationary-engines running at slow speed may have 16 inch lead. The effect of compression is to fill the waste nd of the cylinder with steam; consequently, engines having ssion need less lead. Locomotive-engines having the valves the ordinary form of Stephenson link-motion may have hen running slowly and with a long cut-off, but when at speed ut-off the lead is at least ¼ inch; and locomotives that have ich gives constant lead commonly have ¼ inch lead. The he angle the crank makes with the line of dead points at may vary from 0° to 8°.

ad.—Weisbach (vol. ii. p. 296) says: Experiment shows that ning of the exhaust ports is especially of advantage, and in es the lead of the valve upon the side of the exhaust, or the 1/25 to 1/15; i.e., the slide-valve at the lowest or highest posi-ton has made an opening whose height is 1/25 to 1/15 of the f the slide valve. The outside lead of the slide-valve or the team side, on the other hand, is much smaller, and is often he whole throw of the valve.

f Changing Outside Lap, Inside Lap, Travel and Angular Advance. (Thurston.)

ission	Expansion	Exhaust	Compression
ter, sooner	occurs earlier, continues longer	is unchanged	begins at same point
anged	begins as before, continues longer	occurs later, ceases earlier	begins sooner, continues longer
sooner, es longer	begins later, ceases sooner	begins later, ceases later	begins later, ends sooner
earlier, naltered	begins sooner, per. the same	begins earlier, per. unchanged	begins earlier, p.r. the same

the following relations (Weisbach-Dubois, vol. ii. p. 307):
l of valve, p = maximum port opening;
n-lap, l = exhaust-lap;

of steam-lap to half travel $= \frac{L}{.5S}$, $L = \frac{R}{2} \times S$;

of exhaust lap to half travel $= \frac{l}{.5S}$, $l = \frac{r}{2} \times S$;

$2L = 2p + 2R + S$; $S = \frac{2p}{1 - R}$.

HOF between positions of crank at admission and at cut-off, l β = angle *QOR* between positions of crank at release and at npression, then $R = \frac{1}{2}\frac{\sin(180^\circ - a)}{\sin \frac{1}{2}a}$; $r = \frac{1}{2}\frac{\sin(180^\circ - \beta}{\sin \frac{1}{2}\beta}$.

Lap and of Port-opening to Valve-travel.—The 831, giving the ratio of lap to travel of valve and ratio of travel g, is abridged from one given by Buel in Weisbach-Dubois, alculated from the above formulæ. Intermediate values may ie formulæ, or with sufficient accuracy by interpolation from the table. By the table on page 830 the crank-angle may be , the angle between its position when the engine is on the *position at cut-off*, release, or compression, when these are *tions of the stroke.* To illustrate the use of the tables the *ple is given by Buel:* width of port = 2.2 in.; w *of port + 0.3 in.;* over overtravel = 2.5 = 2½ *times stroke;* cut-off, .75 of stroke; le, 10°. From the first table we find cran add lead-angle, making 124.6.° From the second table, for angle betw admission and cut-off, 125°, we have ratio of travel to port-opening = or for 124.6° = 3.74, which, multiplied by port-opening 2.5, gives 9.4 travel. The ratio of lap to travel, by the table, is .2324, or 9.45 × .2324 = in. lap. For exhaust-lap we have, for release at .95, crank-angle = 1 add lead-angle 10° = 161.3°. From the second table, by interpolation, r of lap to travel = .0811, and .0811 × 9.45 = 0.77 in., the exhaust-lap.

Lap-angle = ½ (180° − lead-angle − crank-angle at cut-off);
= ½ (180° − 10 − 114.6) = 27.7°.
Angular advance = lap-angle × lead-angle = 27.7 + 10 = 37.7°.
Exhaust lap-angle = crank-angle at release + lap-angle + lead-angle −
= 151.3 + 27.7 + 10 − 180° = 9°.

Crank-angle at compression measured on return stroke } = 180° − lap-angle − lead-angle − exhaust lap-an
= 180 − 27.7 − 10 − 9 = 133.3° ; corresponding
table, to a piston position of .81 of the return stroke; or
Crank-angle at compression = 180° − (angle at release − angle at cut
+ lead-angle;
= 180 − (151.3 − 114.6) + 10 = 133.3°.

The positions determined above for cut-off and release are for the forw stroke of the piston. On the return stroke the cut-off will take plac the same angle, 114.6°, corresponding by table to 66.6% of the re stroke, instead of 75%. By a slight adjustment of the angular adv and the length of the eccentric rod the cut-off can be equalized. width of the bridge should be at least 2.5 + 0.25 − 2.2 = 0.55 in.

Crank Angles for Connecting-rods of Different Lengt

FORWARD AND RETURN STROKES.

Fraction of Stroke from Commencement.	Ratio of Length of Connecting-rod to Length of Stroke.											
	2		2½		3		3½		4		5	
	For.	Ret.	For.	Ret.	For.	Ret.	For.	Ret.	For.	Ret.	For.	Ret.
.01	10.3	13.2	10.5	12.8	10.6	12.6	10.7	12.4	10.8	12.3	10.9	12.1
.02	14.6	18.7	14.9	18.1	15.1	17.8	15.2	17.5	15.3	17.4	15.5	17.1
.03	17.9	22.9	18.2	22.2	18.5	21.8	18.7	21.5	18.8	21.3	19.0	[illegible]
.04	20.7	26.5	21.1	25.7	21.4	25.2	21.6	24.9	21.8	24.6	22.0	[illegible]
.05	23.2	29.6	23.6	28.7	24.0	28.2	24.2	27.8	24.4	27.5	24.7	[illegible]
.10	33.1	41.9	33.8	40.8	34.3	40.1	34.6	39.6	34.9	39.2	35.2	[illegible]
.15	41	51.5	41.9	50.2	42.4	49.3	42.9	48.7	43.2	48.3	43.6	47.7
.20	48	59.6	48.9	58.2	49.6	57.3	50.1	56.6	50.4	56.2	50.9	[illegible]
.25	54.3	66.9	55.4	65.4	56.1	64.4	56.6	63.7	57.0	63.3	57.6	[illegible]
.30	60.3	73.5	61.5	72.0	62.2	71.0	62.8	70.3	63.3	69.8	63.9	[illegible]
.35	66.1	79 8	67.3	78.3	68.1	77.3	68.8	76.6	69 2	76.1	69.9	[illegible]
.40	71.7	85.8	73.0	84.3	73.9	83.3	74.5	82.6	75.0	82.0	75.7	[illegible]
.45	77.2	91.5	78.6	90.1	79.6	89.1	80.2	88.4	80.7	87.9	81.4	[illegible]
.50	82.8	97.2	84.3	95.7	85.2	94.8	85.9	94.1	86.4	93.6	87.1	[illegible]
.55	88.5	102.8	89.9	101.4	90.9	100.4	91.6	99.8	92.1	99.3	[illegible]	[illegible]
.60	94.2	108.3	95.7	107.0	96.7	106.1	97.4	105.5	98.0	105.0	[illegible]	[illegible]
.65	100.2	113.9	101.7	112.7	102.7	111.9	103.4	111.2	103.9	110.8	104.7	[illegible]
.70	106.5	119.7	108.0	118.5	109.0	117.8	109.7	117.2	110.2	116.7	110.9	[illegible]
.75	113.1	125.7	114.6	124.6	115.6	123.9	116.3	123.4	116.7	123.0	117.4	[illegible]
.80	120.4	132	121.8	131.1	122.7	130.4	123.4	129.9	123.8	129.6	[illegible]	[illegible]
.85	128.5	139	129.8	138.1	130.7	137.6	131.3	137.1	131.7	136.8	[illegible]	[illegible]
.90	138.1	146.9	139.2	146.2	139.9	145.7	140.4	145.4	140.8	145.1	141.3	[illegible]
.95	150.4	156.8	[illegible]	[illegible]	151.8	156.0	152.2	155.8	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	153.5	[illegible]	[illegible]	[illegible]	154.8	158.6	155.1	158.4	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	157.1	[illegible]	[illegible]	[illegible]	158.2	161.5	158.5	161.8	158.7	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	162.2	164.9	162.5	164.8	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	169.4	167.6	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	180	180	180	180	180	180	180

tive Motions of Cross-head and Crank.—If L = length ecting-rod, R = length of crank, θ = angle of crank with centre line ie, D = displacement of cross-head from the beginning of its stroke,

$$D = R(1 - \cos \theta) + L - \sqrt{L^2 - R^2 \sin^2 \theta}.$$

Lap and Travel of Valve.

pression.	Ratio of Lap to Travel of Valve.	Ratio of Travel of Valve to Width of Port-opening.	Angle between Positions of Crank at Points of Admission and Cut-off, or Release and Compression.	Ratio of Lap to Travel of Valve.	Ratio of Travel of Valve to Width of Port-opening.	Angle between Positions of Crank at Points of Admission and Cut-off, or Release and Compression.	Ratio of Lap to Travel of Valve.	Ratio of Travel of Valve to Width of Port-opening.
	.4830	58.70	85°	.3686	7.61	135°	.1913	3.24
	.4769	43.22	90	.3536	6.83	140	.1710	3.04
	.4699	33.17	95	.3378	6.17	145	.1504	2.86
	.4619	26.27	100	.3214	5.60	150	.1294	2.70
	.4532	21.34	105	.3044	5.11	155	.1082	2.55
	.4435	17.70	110	.2868	4.69	160	.0868	2.42
	.4330	14.93	115	.2687	4.32	165	.0653	2.30
	.4217	12.77	120	.2500	4.00	170	.0436	2.19
	.4096	11.06	125	.2309	3.72	175	.0218	2.09
	.3967	9.68	130	.2113	3.46	180	.0000	2.00
	.3830	8.55						

ODS OF ADMISSION, OR CUT-OFF, FOR VARIOUS LAPS AND TRAVELS OF SLIDE-VALVES.

wo following tables are from Clark on the Steam-engine. In the first e given the periods of admission corresponding to travels of valve 12 in. to 2 in., and laps of from 2 in. to ⅜ in., with ¼ in. and ⅛ in. of With greater leads than those tabulated, the steam would be cut off than as shown in the table.

nfluence of a lead of 5/16 in. for travels of from 1⅝ in. to 6 in., and from ⅛ in. to 1⅛ in., as calculated for in the second table, is exhibited parison of the periods of admission in the table, for the same lap and The greater lead shortens the period of admission, and increases the or expansive working.

ds of Admission, or Points of Cut-off, for Given Travels and Laps of Slide-valves.

Lead.	Periods of Admission, or Points of Cut-off, for the following Laps of Valves in inches.									
	2	1¾	1½	1¼	1	⅞	¾	⅝	½	⅜
in.	%	%	%	%	%	%	%	%	%	%
¼	88	90	93	95	96	97	98	98	99	99
¼	82	87	89	92	95	96	97	98	98	99
¼	72	78	84	88	92	94	95	96	98	98
¼	50	62	71	79	86	89	91	94	96	97
[illegible]	43	56	68	77	85	88	91	94	96	97
[illegible]	32	47	61	72	82	86	89	92	95	97
[illegible]	14	35	51	66	78	83	87	90	94	[illegible]
[illegible]	......	17	39	57	72	78	83	88	[illegible]	
[illegible]	......	......	20	44	63	71	79	84		
[illegible]	......	......	......	23	50	61	71	79		
[illegible]	......	......	......	......	27	43	57	70		
	......	......	......	......	......	......	33	52		

Periods of Admission, or Points of Cut-off, for Travels and Laps of Slide-valves.

Constant lead, 3/16.

Travel.	Lap.							
Inches.	½	⅝	¾	⅞	1	1⅛	1¼	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

Diagram for Port-opening, Cut-off, and Lap.—The on the opposite page was published in *Power*, Aug., 1893. It sh glance the relations existing between the outside lap, steam port and cut-off in slide-valve engines.

In order to use the diagram to find the lap, having given the m maximum port-opening, follow the ordinate representing the lat on the horizontal scale, until it meets the oblique line representing cut-off. Then read off this height on the vertical lap scale. Th port-opening of 1¼ inch and a cut-off of .50, the intersection of the occurs on the horizontal 3. The required lap is therefore 3 in.

If the cut-off and lap are given, follow the horizontal represe latter until it meets the oblique line representing the cut-off. Then below this read the corresponding port-opening on the horizontal

If the lap and port-opening are given, the resulting cut-off may tained by finding the point of intersection of the ordinate represe port-opening with the horizontal representing the lap. The o passing through the point of intersection will give the cut-off.

If it is desired to take lead into account, multiply the lead in inc numbers in the following table corresponding to the cut-off, and result from the lap as obtained from the diagram:

Cut-off.	Multiplier.	Cut-off.	Mul[illegible]
.20	4.717	.60	[illegible]
.25	3.731	.625	[illegible]
.30	3.048	.65	[illegible]
.33	2.717	.70	[illegible]
.375	2.381	.75	[illegible]
.40	2.171	.80	[illegible]
.45	[illegible]	.85	[illegible]
		.875	[illegible]
		.90	[illegible]

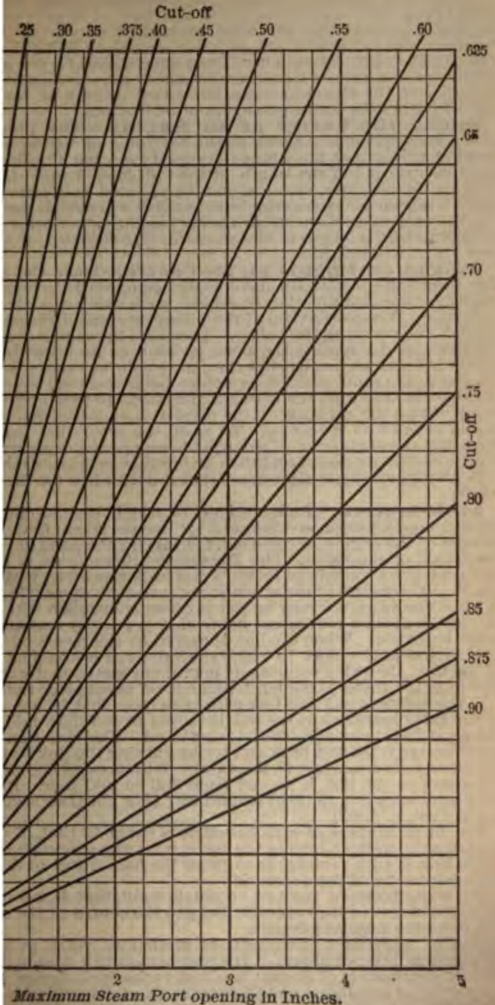


DIAGRAM FOR SLIDE VALVES.

Fig. 149.

motion, temperature, etc., and can be effected only af
valve has been set as accurately as possible when cold, t
the forward and return strokes being equalized, indicat
be taken and the length of the eccentric-rod adjusted, i
rect slight irregularities.

To Put an Engine on its Centre.—Place th
tion where the piston will have nearly completed its
opposite some point on the cross-head, such as a corner
the guide. Against the rim of the pulley or crank-disk
mark a line with it on the pulley. Then turn the engine
the cross-head is again in the same position on its inwa
bring the crank as much below the centre as it was above
pointer in the same position as before make a second m
rim. Divide the distance between the marks in two a
point. Turn the engine until the pointer is opposite th
it will then be on its centre. To avoid the error that
looseness of crank-pin and wrist-pin bearings, the engi
a little above the centre and then be brought up to it, s
will press against the same brass that it does when the
made.

Link-motion.—Link-motions, of which the Ste
most commonly used, are designed for two purposes: fi
motion of the engine, and second, for varying the point
the travel of the valve. The Stephenson link-motion
two eccentrics, called the forward and back eccentric,
ing the extremeties of the eccentic-rods; so that by var
the link the valve-rod may be put in direct connection w
or may be given a movement controlled in part by one
other eccentric. When the link is moved by the reversi
tion such that the block to which the valve-rod is attac
of the link, the valve receives its maximum travel, and
mid-gear the travel is the least and cut-off takes place

In the ordinary shifting-link with open rods, that is, n
of the valve increases as the link is moved from full to
the period of steam admission is shortened. The variat
lized for the front and back strokes by curving the link
eccentric-rods concavely to the axles. With crossed ec
decreases as the link is moved from full to mid-gear.
with stationary link the lead is constant. (For illustratio
engine, vol. ii. p. 22.)

The linear advance of each eccentric is equal to that rods ought to be long; the longer they are in proportion to the y the more symmetrical will the travel of the valve be on both e centre of motion. 3. The link ought to be short. Each of its cribes a curve in a vertical plane, whose ordinates grow larger the considered point is from the centre of the link; and as the horiion only is transmitted to the valve, vertical oscillation will cause ies. 4. The link-hanger ought to be long. The longer it is the be the arc in which the link swings to a straight line, and thus s vertical oscillation. If the link is suspended in its centre, the t are described by points equidistant on both sides from the centre ke, and hence results the variation between the forward and back-. If the link is suspended at its lower end, its lower half will have al oscillation and the upper half more. 5. The centre from which anger swings changes its position as the link is lowered or raised, auses irregularities. To reduce them to the smallest amount the lifting-shaft should be made as long as the eccentric-rod, and the the lifting-shaft should be placed at the height corresponding to l position of the centre on which the link-hanger swings."

conditions can never be fulfilled in practice, and the variations d and the period of admission can be somewhat regulated in an vay, but for one gear only. This is accomplished by giving differthe two eccentrics, which difference will be smaller the longer the rods are and the shorter the link, and by suspending the link not its centre line but at a certain distance from it, giving what is e offset."

lication of the Zeuner diagram to link-motion, see Holmes on the gine, p. 290. See also Clark's Railway Machinery (1855), Clark's gine and Zeuner's and Auchincloss's Treatises on Slide-valve

owing rules are given by the *American Machinist* for laying out a n upright slide-valve engine. By the term radius of link is meant of the link-arc *ab*, Fig. 150, drawn through the centre of the slot;

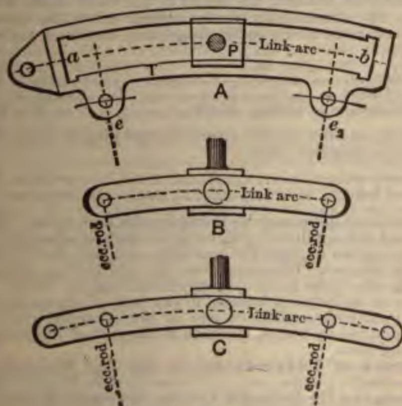


FIG. 150.

s is generally made equal to the distance from the centre of shaft of the link-block pin *P* when the latter stands midway of its travel. nce between the centres *of the* eccentric-rod pins e_1 e_2 should not an 2½ *times, and, when space will permit,* three times the thr... *ric. By the throw we mean twice* the eccentricity of the e... *k is generally suspended from* the end next to the forwa... *int in the link-arc prolonged.* This will give compa... *t of slip to the link-block* when the link is in forward *be increased when the link* is in backward gear. Th...

of slip is, however, considered of little importance, because marine en[gines], as a rule, work but very little in the backward gear. When it is nece[ssary] that the motion shall be as efficient in backward gear as in forward g[ear], then the link should be suspended from a point midway between the [two] eccentric-rod pins; in marine engine practice this point is generally loc[ated] on the link-arc; for equal cut-offs it is better to move the point of susp[en]sion a small amount towards the eccentrics.

For obtaining the dimensions of the link in inches: Let L denote [the] length of the valve, B the breadth, p the absolute steam-pressure per [sq.] in., and R a factor of computation used as below; then $R = .01\sqrt{L \times B \ldots}$

Breadth of the link	$= R \times 1.6$
Thickness T of the bar	$= R \times .8$
Length of sliding-block	$= R \times 2.5$
Diameter of eccentric-rod pins	$= (R \times .7) + \frac{1}{4}$
Diameter of suspension-rod pin	$= (R \times .6) + \frac{1}{4}$
Diameter of suspension-rod pin when overhung	$= (R \times .8) + \frac{1}{4}$
Diameter of block-pin when overhung	$= R + \frac{1}{4}$
Diameter of block-pin when secured at both ends	$= (R \times .8) + \frac{1}{4}$

The length of the link, that is, the distance from a to b, measured [on a] straight line joining the ends of the link-arc in the slot, should be such [as to] allow the centre of the link-block pin P to be placed in a line with the [eccen]tric-rod pins, leaving sufficient room for the slip of the block. Another [type] of link frequently used in marine engines is the double-bar link, and [this] type is again divided into two classes: one class embraces those links wh[ich] have the eccentric-rod ends as well as the valve-spindle end between [the] bars, as shown at B (with these links the travel of the valve is less t[han] the throw of the eccentric); the other class embraces those links, show[n at] C, for which the eccentric-rods are made with fork-ends, so as to connect [to] studs on the outside of the bars, allowing the block to slide to the end of [the] link, so that the centres of the eccentric-rod ends and the block-pin are [in] line when in full gear, making the travel of the valve equal to the throw [of] the eccentric. The dimensions of these links when the distance betw[een] the eccentric-rod pins is 2½ to 2¾ times the throw of eccentrics can [be] found as follows:

Depth of bars	$= (R \times 1.25) + \frac{1}{2}''$
Thickness of bars	$= (R \times .5) + \frac{1}{4}''$
Diameter of centre of sliding-block	$= R \times 1.3$

When the distance between the eccentric-rod pins is equal to 3 or 4 [times] the throw of the eccentrics, then

Depth of bars	$= (R \times 1.25) + \frac{5}{8}''$
Thickness of bars	$= (R \times .5) + \frac{5}{16}''$

All the other dimensions may be found by the first table. These are em[pirical] rules, and the results may have to be slightly changed to suit g[iven] conditions. In marine engines the eccentric-rod ends for all classes of l[inks] have adjustable brasses. In locomotives the slot-link is usually em[ployed], and in these the pin-holes have case-hardened bushes driven into the [pin-]holes, and have no adjustable brasses in the ends of the eccentric-rods. T[he] link in B is generally suspended by one of the eccentric-rod pins; and [the] link in C is suspended by one of the pins in the end of the link, or by o[ne of] the eccentric-rod pins.

Other Forms of Valve-Gear, as the Joy, Marshall, Hack[worth], Brown, Walschaert, Corliss, etc., are described in Clark's Steam-en[gine], vol. ii. The design of the Reynolds-Corliss valve-gear is discussed by A. [H.] Eldridge in *Power*, Sep. 1893. See also Henthorn on the Corliss [engine]. Rules for laying down the centre lines of the Joy valve-gear are giv[en in] *American Machinist*, Nov. 13, 1890. For Joy's "Fluid-pressure Rever[sing] valve," see *Eng'g*, May 25, 1894.

GOVERNORS.

Pendulum or [Ba]ll Governor.—The inclination of the [arms to the] vertical axis is such that the height [of the] revolving [...] the horizontal plane in which the [centres] of [...] assuming the weight of the rods is [...] y of the [...]

with the weight of the balls) bears to the radius r of the circle by the centres of the balls the ratio

$$\frac{h}{r} = \frac{\text{weight}}{\text{centrifugal force}} = \frac{w}{\frac{wv^2}{gr}} = \frac{gr}{v^2},$$

o is independent of the weight of the balls, v being the velocity tres of the balls in feet per second.
umber of revolutions of the balls in 1 second, $v = 2\pi rT = ar$, in the angular velocity, or $2\pi T$, and

$$h = \frac{gr^2}{v^2} = \frac{g}{4\pi^2T^2}, \text{ or } h = \frac{0.8146}{T^2} \text{ feet} = \frac{9.775}{T^2} \text{ inches},$$

aken at 32.16. If N = number of revs. per minute, $h = \frac{35190}{N^2}$

volutions per minute...... ...	40	45	50	60	75
ight in inches will be...... ...	21.99	17.38	14.08	9.775	6.256

of turns per minute required to cause the arms to take a given the vertical axis: Let l = length of the arm in inches from the suspension to the centre of gyration, and a the required angle;

$$N = \sqrt{\frac{35190}{l \cos a}} = 187.6\sqrt{\frac{1}{l \cos a}} = 187.6\sqrt{\frac{1}{h}}.$$

ple governor is not isochronous; that is, it does not revolve at a peed in all positions, the speed changing as the angle of the arms To remedy this defect loaded governors, such as Porter's, are om the balls of a common governor whose collective weight is A e hung by a pair of links of lengths equal to the pendulum arms apable of sliding on the spindle, having its centre of gravity in f rotation. Then the centrifugal force is that due to A alone, and of gravity is that due to $A + 2B$; consequently the altitude for a d is increased in the ratio $(A + 2B) : A$, as compared with that of evolving pendulum, and a given absolute variation in altitude pro-aller proportionate variation in speed than in the common gover-kine. S. E., p. 551.)
weighted governor let l = the length of the arm from the point of to the centre of gravity of the ball, and let the length of the sus-nk, l_1 = the length of the portion of the arm from the point of of the arm to the point of attachment of the link; G = the weight l, Q = half the weight of the sliding weight, h = the height of the from the point of suspension to the plane of revolution of the the angular velocity $= 2\pi T$, T being the number of revolutions per

hen $a = \sqrt{\frac{32.16}{h}\left(1+\frac{2l_1}{l}\frac{Q}{G}\right)}$; $h = \frac{32.16}{a^2}\left(1+\frac{2l_1}{l}\frac{Q}{G}\right)$ in feet, or

$\left(1+\frac{2l_1}{l}\frac{Q}{G}\right)$ in inches, N being the number of revolutions per

ous forms of governor see App. Cycl. Mech., vol. ii. 61, and Clark's ine, vol. ii. p. 65.

ange the Speed of an Engine Having a Fly-ball or.—A slight difference in the speed of a governor changes the f its weights from that required for full load to that required for It is evident therefore that, whatever the speed of the engine, the eed of the governor must be that for which the governor was de-., the speed of the governor must be kept the same. To change the he engine the *problem is to so adjust the pulleys which drive the that the engine at its new speed shall drive it just as fast s original speed. In order to increase the engine-spe e pulley upon the shaft of the engine, i.e., the driver overnor, i.e., the driven, in the proportion that the e increased.*

Fly-wheel or Shaft Governors.—At the Centennial Exh in 1876 there were shown a few steam-engines in which the governor contained in the fly-wheel or band-wheel, the fly-balls or weights re around the shaft in a vertical plane with the wheel and shifting the e tric so as automatically to vary the travel of the valve and the point off. This form of governor has since come into extensive use, especial high-speed engines. In its usual form two weights are carried on arm ends of which are pivoted to two points on the pulley near its ci ference, 180° apart. Links connect these arms to the eccentric eccentric is not rigidly keyed to the shaft but is free to move versely across it for a certain distance, having an oblong hole which of this movement. Centrifugal force causes the weights to fly toward circumference of the wheel and to pull the eccentric into a position of imum eccentricity. This force is resisted by a spring attached to each which tends to pull the weights towards the shaft and shift the eccen the position of maximum eccentricity. The travel of the valve is varied, so that it tends to cut off earlier in the stroke as the engine incr its speed. Many modifications of this general form are in use. For d sions of this form of governor see Hartnell, Proc. Inst. M. E., 1882, p Trans. A. S. M. E., ix. 300; xi. 1081; xiv. 92; xv. 929; Modern Mecha p. 399; Whitham's Constructive Steam Engineering; J. Begtrup, Am. M Oct. 19 and Dec. 14, 1893, Jan. 18 and March 1, 1894.

Calculation of Springs for Shaft-governors. (Wilson H nell, Proc. Inst. M. E., Aug. 1882.)—The springs for shaft-governors m conveniently calculated as follows, dimensions being in inches:

Let W = weight of the balls or weights, in pounds;

r_1 and r_2 = the maximum and minimum radial distances of the c of the balls or of the centre of gravity of the weights;

l_1 and l_2 = the leverages, i.e., the perpendicular distances from the tre of the weight-pin to a line in the direction of the centrifugal drawn through the centre of gravity of the weights or balls at r_1 and r_2;

m_1 and m_2 = the corresponding leverages of the springs;

C_1 and C_2 = the centrifugal forces, for 100 revolutions per minu radii r_1 and r_2;

P_1 and P_2 = the corresponding pressures on the spring;

(It is convenient to calculate these and note them down for refer

C_3 and C_4 = maximum and minimum centrifugal forces;

S = mean speed (revolutions per minute);

S_1 and S_2 = the maximum and minimum number of revolution minute;

P_3 and P_4 = the pressures on the spring at the limiting number of lutions (S_1 and S_2);

$P_4 - P_3 = D$ = the difference of the maximum and minimum pre on the springs;

V = the percentage of variation from the mean speed, or the sen ness;

t = the travel of the spring;

u = the initial pressure on the spring;

v = the stiffness in pounds per inch;

w = the maximum pressure = $u + t$.

The mean speed and sensitiveness desired are supposed to be given.

$$S_1 = S - \frac{SV}{100}; \qquad S_2 = S + \frac{SV}{100};$$

$$C_1 = 0.28 \times r_1 \times W; \qquad C_2 = 0.28 \times r_2 \times W;$$

$$P_1 = C_1 \times \frac{l_1}{m_1}; \qquad P_2 = C_2 \times \frac{l_2}{m_2};$$

$$P_3 = P_1 \times \left(\frac{S_1}{100}\right)^2; \qquad P_4 = P_2 \times \left(\frac{S_2}{100}\right)^2;$$

$$v = \frac{D}{t}, \quad u = \frac{P_3}{v}, \quad w = \frac{P_4}{v}.$$

... is usual to gi ... ing-maker the values of P_4 and of ... e proper sp ... vided, the dimensions of the spr

the formulæ for strength and extension of springs, and the
f the spring as compressed be determined.

$$\text{The governor-power} = \frac{P_3 + P_4}{2} \times \frac{t}{12}.$$

ight centripetal line, the governor-power

$$= \frac{C_3 + C_4}{2} \times \left(\frac{r_2 - r_1}{12}\right).$$

ninary determination of the governor-power it may be taken
is in all cases, although it is evident that with a curved cen-
t will be slightly less. The difference D must be constant for
ing, however great or little its initial compression. Let the
ewed up until its minimum pressure is P_5. Then to find the
$_5 + D$,

$$S_5 = 100\sqrt{\frac{P_5}{P_1}}; \qquad S_6 = 100\sqrt{\frac{P_6}{P_2}}.$$

t which the governor would be isochronous would be

$$100\sqrt{\frac{D}{P_2 - P_1}}.$$

pressure on the spring with a speed of 100 revolutions, at the
d minimum radii, was 200 lbs. and 100 lbs., respectively, then
of the spring to suit a variation from 95 to 105 revolutions will
$)^2 = 90.2$ and $200 \times \left(\frac{105}{100}\right)^2 = 220.5$. That is, the increase
rom the minimum to the maximum radius must be $220 - 90 =$

e speeds due to such a spring, screwed up to different press-
n in the following table:

er minute, balls shut...............	80	90	95	100	110	120
prings, balls shut....................	64	81	90	100	121	144
essure when balls open fully.......	130	130	130	130	130	130
prings, balls open fully..............	194	211	220	230	251	274
er minute, balls open fully.	98	102	105	107	112	117
cent of mean speed	10	6	5	3	1	−1

t which the governor would become isochronous is 114.
will give the right variation at some speed; hence in experi-
a governor the correct spring may be found from any wrong
simple calculation. Thus, if a governor with a spring whose
lbs. per inch acts best when the engine runs at 95, 90 being its
then $50 \times \left(\frac{90}{95}\right)^2 = 45$ lbs. is the stiffness of spring required.

e the speed at which the governor acts best, the springs may
until it begins to "hunt" and then slackened until the gov-
sitive as is compatible with steadiness.

DENSERS, AIR-PUMPS CIRCULATING-PUMPS, ETC.

Condenser. (Chiefly abridged from Seaton's Marine Engi-
e jet condenser is now uncommon, being generally supplanted
e condenser. With *the jet condenser a vacuum of 24 in. was
rly good, and 25 in. as much as was possible with most conden-
erature corresponding to 24 in. vacuum, or 3 lbs. press*
In practice the temperature in the hot-well varies f
sionally as much as 130° is maintained. To find the
r per pound of steam to be condensed: Let T_1 =
the exhaust pressure; T_0 = temperature of th

what was usual to allow for general traders.

Area of injection orifice = weight of injection-water

to 780.

A rough rule sometimes used is: Allow one fifteenth every cubic foot of water condensed per hour.

Another rule: Area of injection orifice = area of pist

The volume of the jet condenser is from one fourth t the cylinder. It need not be more than one third, e running engines.

Ejector Condensers.—For ejector or injector c Schutte's, etc.) the calculations for quantity of condens as for jet condensers.

The Surface Condenser—Cooling Surfac with cooling water of an initial temperature of 68° to 77 plate condensed 21.5 lbs. of steam per hour, while Jo per hour can be condensed. In practice, with the co condenser-tubes, 18 B.W.G thick, 13 lbs. of steam per the cooling-water at an initial temperature of 60°, i work when the temperature of the feed-water is to b It has been found that the surface in the condenser m surface of the boiler, and under some circumstances this. In general practice the following holds good whe sea-water is about 60°:

Terminal pres., lbs., abs....	30	20	15	12½
Sq. ft. per I.H.P............	3	2.50	2.25	2.00

For ships whose station is in the tropics the allowanc by 20%, and for ships which occasionally visit the trop give satisfactory results. If a ship is constantly emplo 10% less suffices.

Whitham (Steam-engine Design, p. 283, also Trans gives the following: $S = \frac{WL}{ck(T_1 - t)}$, in which S = cond ft.; T_1 = temperature Fahr. of steam of the press vacuum-gauge; t = mean temperature of the circu arithmetical mean of the initial and final temperature saturated steam at temperature T_1; k = perfect condu the metal used for the condensing-surface for a range per hour for brass, according to Isherwood's experime noting the efficiency of the condensing-surface; W = p niralty, however, always specify the tubes to be made of 70% of best copper and to have 1% of tin in the composition, and test the tubes ssure of 300 lbs. per sq. in. (Seaton.)

ameter of the condenser tubes varies from ½ inch in small conden-en they are very short, to 1 inch in very large condensers and long In the mercantile marine the tubes are, as a rule, ¾ inch diameter ly, and 18 B.W.G. thick (0.049 inch); and 16 B.W.G. (0.065), under ceptional circumstances. In the British Navy the tubes are also, e, ¾ inch diameter, and 18 to 19 B.W.G. thick, tinned on both sides; e condenser is made of brass the Admiralty do not require the tubes ned. Some of the smaller engines have tubes ⅝ inch diameter, and G. thick. The smaller the tubes, the larger is the surface which ot in a certain space.

merchant service the almost universal practice is to circulate the rough the tubes.

am says the velocity of flow through the tubes should not be less nor more than 700 ft. per min.

-plates are usually made of brass. Rolled-brass tube-plates e from 1.1 to 1.5 times the diameter of tubes in thickness, depending ethod of packing. When the packings go completely through the he latter, but when only partly through the former, is sufficient. or ¾-inch tubes the plates are usually ⅞ to 1 inch thick with glands -packings, and 1 to 1¼ inch thick with wooden ferrules.

ibe-plates should be secured to their seatings by brass studs and brass screw-bolts; in fact there must be no wrought iron of any de a condenser. When the tube-plates are of large area it is advis-tay them by brass-rods, to prevent them from collapsing.

ing of Tubes, etc.—The holes for ferrules, glands, or india-re usually ¼ inch larger in diameter than the tubes; but when ab-necessary the wood ferrules may be only 3/32 inch thick.

itch of tubes when packed with wood ferrules is usually ¼ inch an the diameter of the ferrule-hole. For example, the tubes are y arranged zigzag, and the number which may be fitted into a oot of plate is as follows:

of s.	No. in a sq. ft.	Pitch of Tubes.	No. in a sq. ft.	Pitch of Tubes.	No. in a sq. ft.
	172	1 5/32″	128	1¼″	110
6″	150	1 3/16″	121	1 9/32″	106
	137	1 7/32″	116	1 5/16″	99

ntity of Cooling Water.—The quantity depends chiefly upon l temperature, which in Atlantic practice may vary from 40° in the of temperate zone to 80° in subtropical seas. To raise the tempera-100° in the condenser will require three times as many thermal units former case as in the latter, and therefore only one third as much water will be required in the former case as in the latter.

= temperature of steam entering the condenser;
= " " circulating-water entering the condenser;
= " " " " leaving the condenser;
= " " water condensed from the steam;

$$Q = \text{quantity of circulating water in lbs.} = \frac{1114 + 0.3(T_1 - T_3)}{T_2 - T_0}.$$

usual to provide pumping power sufficient to supply 40 times the of steam for general traders, and as much as 50 times for ships sta-in subtropical seas, when the engines are compound. If the circulat-mp is double-acting, its capacity may be 1/53 in the former and 1/42 latter case of the capacity of the low-pressure cylinder.

***pump.**—The air-pump in all condensers abstracts the water co- and the air originally contained in the water when it entered In the case of jet-condensers it also pumps out the water o and the air which it contained.* The size of the pump is calc conditions, making allowance for efficiency of the pump.

Ordinary sea-water contains, mechanically mixed with it, 1/20 of [illegible] ume of air when under the atmospheric pressure. Suppose the [illegible] the condenser to be 2 lbs. and the atmospheric pressure 15 lbs., [illegible] the effect of temperature, the air on entering the condenser will be [illegible] to 15/2 times its original volume; so that a cubic foot of sea-water, [illegible] has entered the condenser, is represented by 19/20 of a cubic foot [illegible] and 15/40 of a cubic foot of air.

Let q be the volume of water condensed per minute, and Q the [illegible] sea-water required to condense it; and let T_2 be the temperature [illegible] condenser, and T_1 that of the sea-water.

Then $19/20\ (q + Q)$ will be the volume of water to be pumped fr[illegible] condenser per minute,

$$\text{and } \frac{15}{40}(q + Q) \times \frac{T_2 + 461°}{T_1 + 461°} \text{ the quantity of air.}$$

If the temperature of the condenser be taken at 120°, and that [illegible] water at 60°, the quantity of air will then be $.418(q + Q)$, so that th[illegible] volume to be abstracted will be

$$.95(q + Q) + .418(q + Q) = 1.368(q + Q).$$

If the average quantity of injection-water be taken at 26 times th[illegible] densed, $q + Q$ will equal $27q$. Therefore, volume to be pumped fr[illegible] condenser per minute = $37q$, nearly.

In surface condensation allowance must be made for the water [illegible] ally admitted to the boilers to make up for waste, and the air contai[illegible] it, also for slight leak in the joints and glands, so that the air-pump [illegible] about half as large as for jet-condensation.

The efficiency of a single-acting air-pump is generally taken at [illegible] that of a double-acting pump at 0.35. When the temperatur of the [illegible] 60°, and that of the (jet) condenser is 120°, Q being the volume of the [illegible] water and q the volume of the condensed water in cubic feet, and [illegible] number of strokes per minute,

$$\text{The volume of the single-acting pump} = 2.74\left(\frac{Q + q}{n}\right).$$

$$\text{The volume of the double-acting pump} = 4\left(\frac{Q + q}{n}\right).$$

The following table gives the ratio of capacity of cylinder or cylin[illegible] that of the air-pump; in the case of the compound engine, the low-pr[illegible] cylinder capacity only is taken.

Description of Pump.	Description of Engine.	[illegible]
Single-acting vertical......	Jet-condensing, expansion 1¼ to 2....	[illegible]
" " "	Surface " " 1½ to 2....	[illegible]
" "	Jet " " 3 to 5....	[illegible]
" "	Surface " " 3 to 5...	[illegible]
" "	Surface " compound............	[illegible]
Double-acting horizontal..	Jet " expansion 1¼ to 2....	[illegible]
" " ..	Surface " " 1½ to 2....	[illegible]
" " ..	Jet " " 3 to 5 ...	[illegible]
" " ..	Surface " " 3 to 5 ..	[illegible]
" " ..	Surface " compound	[illegible]

The Area through Valve-seats and past the valves should [illegible] less than will admit the full quantity of water for condensation at a [illegible] not exceeding 400 ft. per minute. In practice the area is gene[illegible] excess of this.

Area through foot-valves $= D^2 \times S \div 1000$ square inches.
Area through head-valves $= D^2 \times S \div 800$ square inches.

Diameter of discharge pipe $= D \times \sqrt{S} \div 35$ inches.
D = diam. [illegible] in inches, S = the speed in ft. per [illegible]

[illegible] Tribe ([illegible] Oct. 8, 1891) gives the following ed with jet-condensers: Volume of single-acting air-pump driven ngine = volume of low-pressure cylinder in cubic feet, multiplied divided by the number of cubic feet contained in one pound of team of the given density. For a double-acting air-pump the will apply, but the volume of steam for each stroke of the pump ut one half. Should the pump be driven independently of the en the relative speed must be considered. Volume of jet-con- volume of air-pump × 4. Area of injection-valve = vol. of air- ubic inches ÷ 520.

ating-pump.—Let Q be the quantity of cooling water in cubic number of strokes per minute, and S the length of stroke in feet.

Capacity of circulating-pump = $Q \div n$ cubic feet.

Diameter " " " $= 13.55\sqrt{\frac{Q}{n \times S}}$ inches.

owing table gives the ratio of capacity of steam-cylinder or cylin- at of the circulating-pump:

cription of Pump.	Description of Engine.	Ratio.
ngle-acting.	Expansive 1½ to 2 times.	13 to 16
" "	" 3 to 5 "	20 to 25
" "	Compound.	25 to 30
ouble "	Expansive 1½ to 2 times.	25 to 30
" "	" 3 to 5 "	36 to 46
" "	Compound.	46 to 56

ar area through the valve-seats and past the valves should be such mean velocity of flow does not exceed 450 feet per minute. The ugh the pipes should not exceed 500 ft. per min. in small pipes and ge pipes.

ntrifugal Circulating-pumps, the velocity of flow in the inlet and es should not exceed 400 ft. per min. The diameter of the fan-wheel ½ to 3 times the diam. of the pipe, and the speed at its periphery ft. per min. If W = quantity of water per minute, in American l = diameter of pipes in inches, R = revolutions of wheel per min., $\frac{W}{6.44}$; diam. of fan-wheel = not less than $\frac{1700}{R}$. Breadth of blade at

. Diam. of cylinder for driving the fan = about $2.8\sqrt{\text{diam. of pipe}}$, roke = 0.28 × diam. of fan.

pumps for Marine Engines.—With surface-condensing he amount of water to be fed by the pump is the amount condensed main engine plus what may be needed to supply auxiliary engines upply leakage and waste. Since an accident may happen to the ondenser, requiring the use of jet-condensation, the pumps of itted with surface-condensers must be sufficiently large to do duty h circumstances. With jet-condensers and boilers using salt water salt water in the boiler must be blown off at intervals to keep the o low that deposits of salt will not be formed. Sea-water contains 2 of its weight of solid matter in solution. The boiler of a surface- ng engine may be worked with safety when the quantity of salt is s that in sea-water. If Q = net quantity of feed-water required in ine to make up for what is used as steam, n = number of times the f the water in the boiler is to that of sea-water, then the gross feed- $\frac{n}{n-1}Q$. In order to be capable of filling the boiler rapidly each p is made of a capacity equal to twice the gross feed-water. Two ps should be supplied, so that one may be kept in reserve to be le the other is out of repair. If Q be the quantity of net feed-water eet, l the length of stroke of feed-pump in feet, and n the num- kes per minute,

eter of each feed-pump plunger in inches $= \sqrt{\frac{550 \times Q}{n \times l}}$.

If W be the net feed-water in pounds,

$$\text{Diameter of each feed-pump plunger in inches} = \sqrt{\frac{8 S \times W}{n \times l}}$$

An Evaporative Surface Condenser built at the Virgini cultural College is described by James H. Fitts (Trans. A. S. M. E., xi It consists of two rectangular end chambers connected by a series o zontal rows of tubes, each row of tubes immersed in a pan of Through the spaces between the surface of the water in each pan a bottom of the pan above air is drawn by means of an exhaust-fan. top of one of the end-chambers is an inlet for steam, and a horizont phragm about midway causes the steam to traverse the upper half tubes and back through the lower. An outlet at the bottom leads to t pump. The condenser, exclusive of connection to the exhaust-fan, oc a floor space of 5′ 4⅛″ × 1′ 9¾″, and 4′ 1½″ high. There are 27 ro tubes, 8 in some and 7 in others; 210 tubes in all. The tubes are of No. 20 B.W.G., ¾″ external diameter and 4′ 9½″ in length. The coolin face (internal) is 176.5 sq. ft. There are 27 cooling pans, each 4′ 9¼″ × 1 and 1 7/16″ deep. These pans have galvanized iron bottoms whic into horizontal grooves ⅛″ wide and ¼″ deep, planed into the tube The total evaporating surface is 234.8 sq. ft. Water is fed to every thi through small cocks, and overflow-pipes feed the rest. A wood casin nects one side with a 30″ Buffalo Forge Co.'s disk-wheel. This w belted to a 3″ × 4″ vertical engine. The air-pump is 5¾″ diameter 6″ stroke, is vertical and single-acting.

The action of this condenser is as follows: The passage of air ov water surfaces removes the vapor as it rises and thus hastens evapor The heat necessary to produce evaporation is obtained from the steam tubes, causing the steam to condense. It was designed to condense 3 steam per hour and give a vacuum of 22 in., with a terminal pressure cylinder of 20 lbs. absolute.

Results of tests show that the cooling-water required is practically eq amount to the steam used by the engine. And since consumption of is reduced by the application of a condenser, its use will actually redu total quantity of water required. From a curve showing the rate of eva tion per square foot of surface in still air, and also one showing the when a current of air of about 2300 ft. per min. velocity is passed ov surface, the following approximate figures are taken:

Temp. F.	Evaporation, lbs. per sq. ft. per hour.		Temp. F.	Evaporation, lbs. p sq. ft. per hour.	
	Still Air.	Current.		Still Air.	Curre
100°	0.2	1.1	140°	0.8	[illegible]
110	0.25	1.6	150	1.1	[illegible]
120	0.4	2.5	160	1.5	[illegible]
130	0.6	3.5	170	2.0	

The Continuous Use of Condensing-water is described series of articles in *Power*, Aug.-Dec., 1892. It finds its application in s tions where water for condensing purposes is expensive or difficult to ob

In San Francisco J. C. H. Stut cools the water after it has left the well by means of a system of pans upon the roof. These pans are sh troughs of galvanized iron arranged in tiers, on a slight incline, so that water flows back and forth for 1500 o. 2000 ft., cooling by evaporation radiation as it flows. The pans are about 5 ft. in width, and the water flows has a depth of about half an inch, the temperature being reduced f about 140° to 90°. The water from the hot-well is pumped up to the hig point of the cooling system and allowed to flow as above described, disch ing finally into the main tank or reservoir, whence it again flows to the denser as required. As the water in the reservoir lowers from evaporation auxiliary feed from the city mains to the condenser is operated, keeping the amount of water in circulation practically constant. mulation of ... the engines, with dust from the surroundin akes a ... ssary about once in six weeks or two m and ... rials, running condensing and non-cond s water is taken from the city mains when the whole apparatus when the engine is run non-condensing. 22 to 23 in. of vacuum ed. A better vacuum is obtained on a warm day with a brisk ng than on a cold day with but a slight movement of the air.

plant the water from the hot-well is sprayed from a number of nd also from a pipe extending around its border, into a large posure cooling it sufficiently for the obtaining of a good vacuum uous use.

em patented by Messrs. See, of Lille, France, the water is dis- n a pipe laid in the form of a rectangle and elevated above a h a series of special nozzles, by which it is projected into a fine coming into contact with the air in this state of extreme divi- er is cooled 40° to 50°, with a loss by evaporation of only one mass, and produces an excellent vacuum. A 3000-H.P. cooler stem has been erected at Lannoy, one of 2500 H.P. at Madrid, and H.P. at Liege, as well as others at Roubaix and Tourcoing. The d be used upon a roof if ground space were limited.

lf-cooling" system of H. R. Worthington the injection-water is a tank, and after having passed through the condenser is dis- heated condition to the top of a cooling tower, where it is scat- ans of distributing-pipes and trickles down through a cellular ade of 6-in. terra-cotta pipes, 2 ft. long, stood on end. The led by a blast of air furnished by a disk fan at the bottom of the he absorption of heat caused by a portion of the water being nd is led to the tank to be again started on its circuit. (*Eng'g* 5, 1896.)

aporative condenser of T. Ledward & Co. of Brockley, London, ickles over the pipes of the large condenser or radiator, and by carries away the heat necessary to be abstracted to condense inside. The condensing pipes are fitted with corrugations ith circular ribs, whereby the radiating or cooling surface is eased. The pipes, which are cast in sections about 76 in. long by , have a cooling surface of 26 sq. ft., which is found sufficient rable conditions to permit of the condensation of 20 to 30 lbs. r hour when producing a vacuum of 13 lbs. per sq. in. In a f this type at Rixdorf, near Berlin, a vacuum ranging from 24 mercury was constantly maintained during the hottest weather The initial temperature of the cooling-water used in the appara- otice ranged from 80° to 85° F., and the temperature in the sun, e condenser was exposed, varied each day from 100° to 115° F. experiments it was found that it was possible to run one engine of 100 horse-power and maintain the full vacuum without the cooling-water at all on the pipes, radiation afforded by the pipes ng to condense the steam for this power.

condensing water-cooler, the hot water coming from the con- rs at the top of a wooden structure about twenty feet in height, veyed into a series of parallel narrow metal tanks. The water from these tanks is spread as a thin film over a series of wooden suspended vertically about 3½ inches apart within the tower. set of partitions, corresponding to the number of metal tanks, lf-way down the tower. From there down to the well is sus- cond set of partitions placed at right angles to the first set. This e rapidity of the downflow of the water, and also thoroughly vater, thus affording a better cooling. A fan-blower at the base of rives a strong current of air with a velocity of about twenty feet against the thin film of water running down over the partitions. ted that for an effectual cooling two thousand times more air must be forced through the apparatus. With such a velocity orbs about two per cent of aqueous vapor. The action of the current is twofold: first, it absorbs heat from the hot water by warmed by radiation; and, secondly, it increases the evapora- process absorbs a great amount of heat. These two cooling different during the different seasons of the year. During the *ths the direct cooling effect of* the cold air is greater, whil- *ver the heat absorption by evaporation* is the more impor- *ng all the year round*, the effect remains very much the *on is never so great* that the deficiency of water wo *the additional amount* of water resulting from the co *very cold winter* months it may be necessary to occa *surplus water*. It was found that the vacuum obt-

this continual use of the same condensing-water varied during
between 27.5 and 28.7 inches. The great saving of space is evid
the fact that only the five-hundredth part of the floor-space is req
if cooling tanks or ponds were used. For a 100-horse-power en
floor-space required is about four square yards by a height of tw
For one horse-power 3.6 square yards cooling-surface is necessa
vertical suspension of the partitions is very essential. With a vent
inches in diameter and a tower 6 by 7 feet and 20 feet high, 10,500 g
water per hour were cooled from 104° F. to 68° F. The followin
was made at Mannheim, Germany: Vacuum in condenser, 28.1 inch
perature of condensing-water entering at top of tower, 104° to
temperature of water leaving the cooler, 66.2° to 71.6° F. The en
of the Sulzer compound type, of 120 horse-power. The amount
necessary for the arrangement amounts to about three per cent of
horse-power of the engine for the ventilator, and from one and on
three per cent for the lifting of the water to the top of the cooler,
being four and one half to six per cent.

A novel form of condenser has been used with considerable s
Germany and other parts of the Continent. The exhaust-steam
engine passes through a series of brass pipes immersed in water,
it gives up its heat. Between each section of tubes a number of g
disks are caused to rotate. These disks are cooled by a curre
supplied by a fan and pass down into the water, cooling it by
ing the heat given out by the exhaust-steam and carrying it up w
driven off by the air-current. The disks serve also to agitate the w
thus aid it in abstracting the heat from the steam. With 85
vacuum the temperature of the cooling water was about 130°
consumption of water for condensing is guaranteed to be less than
for each pound of steam condensed. For an engine 40 in. × 50 in.
lutions per minute, 90 lbs. pressure, there is about 1150 sq. ft. of co
surface. Another condenser, 1600 sq. ft. of condensing-surface, is
three engines, 32 in. × 48 in., 27 in. × 40 in., and 30 in. × 40 in., res
—*The Steamship.*

The Increase of Power that may be obtained by adding a c
giving a vacuum of 26 inches of mercury to a non-condensing engi
approximated by considering it to be equivalent to a net gain of
mean effective pressure per square inch of piston area. If A = area
in square inches, S = piston-speed in ft. per minute, then $\frac{12AS}{33,000}$ =
made available by the vacuum. If the vacuum = 13.2 lbs. per sq.
in. of mercury, then H.P. = $AS \div 2500$.

The saving of steam for a given horse-power will be represented
mately by the shortening of the cut-off when the engine is run
condenser. Clearance should be included in the calculation. To
effective pressure non-condensing, with a given actual cut-off,
considered, add 3 lbs. to obtain the approximate mean *total* pres
densing. From tables of expansion of steam find what actual c
give this mean total pressure. The difference between this and th
actual cut-off, divided by the latter and by 100, will give the per
saving.

The following diagram (from catalogue of H. R. Worthington)
percentage of power that may be gained by attaching a condense
condensing engine, assuming that the vacuum is 12 lbs. per sq. in
gram also shows the mean pressure in the cylinder for a given i
sure and cut-off, clearance and compression not considered.

The pressures given in the diagram are absolute pressures above

To find the mean effective pressure produced in an engine-cylin
lbs. gauge (= 105 lbs. absolute) pressure, cut-off at ¼ stroke: fin
left-hand or initial-pressure column, follow the horizontal line t
until it intersects the oblique line that corresponds to the ¼ cut-o
the mean total pressure from the row of figures directly above th
intersection, which in this case is 63 lbs. From this subtract the
lute back pressure (say 3 lbs. for a condensing engine and 15 lbs.
condensing engine exhausting into the atmosphere) to obtain t
fective pressure, which in this case, for a non-condensing en
lbs. To find the gain of power by the use of a condenser w
nd on [illegible] scale the figures that correspond in posit
[illegible] this case 25%. As the diagram does not ta
[illegible] compression, the results are only appr

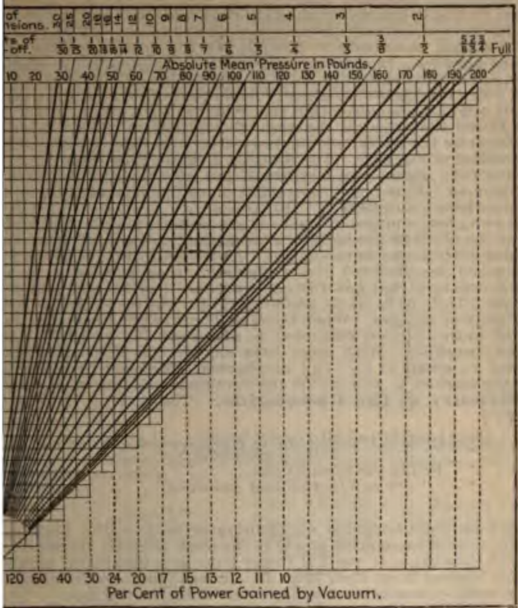


FIG. 151.

ators and Distillers are used with marine engines for the providing fresh water for the boilers or for drinking purposes. *vaporator* consists of a small horizontal boiler, contrived so as taken to pieces and cleaned. The water in it is evaporated by rom the main boilers passing through a set of tubes placed in its The steam generated in this boiler is admitted to the low-lve-box, so that there is no loss of energy, and the water con- is returned to the main boilers.

Feed-heater the feed-water before entering the boiler is heated arly to boiling-point by means of the waste water and steam w-pressure valve-box of a compound engine.

PETROLEUM, AND HOT-AIR ENGINES.

gines.—For theory of the gas-engine, see paper by Dugald . Inst. C. E. 1882, vol. lxix.; and Van Nostrand's Science Series, also Wood's Thermodynamics. For construction of gas-engines, on's Gas and Petroleum Engines; articles by Albert Spies in *fagazine*, 1893; also Appleton's Cyc. of Mechanics, and Modern

linary type of single-cylinder gas-engine (for example the Otto) *four-cycle engine one ignition* of gas takes place in one end of *r every two revolutions of* the fly-wheel, or every two double *e following sequence of operations* takes place during fo *kes; (a) inspiration* during an entire stroke; (b) com *cond (return) stroke; (c)* ignition at the dead-point, an *e third stroke; (d) expulsion* of the burnt gas during *Beau de Rochas* in 1862 laid down the law that four conditions necessary to realize the best results from the elast of gas: (1) The cylinders should have the greatest capacity with the circumferential surface; (2) the speed should be as high as possible cut-off should be as early as possible; (4) the initial pressure shou high as possible. In modern engines it is customary for ignition place, not at the dead point, as proposed by Beau de Rochas, but so later, when the piston has already made part of its forward stroke. sight it might be supposed that this would entail a loss of power, bu ence shows that though the area of the diagram is diminished, tl registered by the friction-brake is greater. Starting is also made this method of working. (The Simplex Engine, Proc. Inst. M. E. 18

In the Otto engine the mixture of gas and air is compressed to atmospheres. When explosion takes place the temperature sudde to somewhere about 2900° F. (Robinson.)

The two great sources of waste in gas-engines are: 1. The high ture of the rejected products of combustion; 2. Loss of heat thr cylinder walls to the water-jacket. As the temperature of the wat is increased the efficiency of the engine becomes higher.

With ordinary coal-gas the consumption may be taken at 20 c hour per I.H.P., or 24 cu. ft. per brake H.P. The consumption will the quality of the gas. When burning Dowson producer-gas the tion of anthracite (Welsh) coal is about 1.3 lbs. per I.H.P. per ordinary working. With large twin engines, 100 H.P., the consum reduced to about 1.1 lb. The mechanical efficiency or B.H.P. ÷ ordinary engines is about 85%; the friction loss is less in larger en

Efficiency of the Gas-engine. (Thurston on Heat as a Energy.)

Heat transferred into useful work....................	1
" " to the jacket-water................	52
" lost in the exhaust-gas........	16
" " by conduction and radiation	15
	— 8

This represents fairly the distribution of heat in the best form engine. The consumption of gas in the best engines ranges from mum of 18 to 20 cu. ft. per I.H.P. per hour to a maximum exceed smaller engines 25 cu. ft. or 30 cu. ft. In small engines the consum brake horse-power is one third greater than these figures.

The report of a test of a 170-H.P. Crossley (Otto) gas-engine in 1892, using producer-gas, shows a consumption of but .85 lb. of coa hour, or an absolute combined efficiency of 21.3% for the engine ducer. The efficiency of the engine alone is in the neighborhood

The Taylor gas-producer is used in connection with the Otto gas the works of Schleicher, Schumm & Co., of Philadelphia. The only to radiation through the walls of the producer and a small amou carried off in the water from the scrubber. Experiments on a engine show a consumption of 97/100 lb. of carbon per I.H.P. per l result is superior to any ever obtained on a steam-engine. (*From*

Tests of the Simplex Gas-engine. (Proc. Inst. M. E. Cylinder 7¾ × 15¾ in., speed 160 revs. per min. Trials were made gas of a heating value of 607 heat-units per cubic foot, and with gas, rich in CO, of about 150 heat-units per cubic foot.

	Town Gas.			Dowson	
	1.	2.	3.	1.	2.
Effective H.P................	6.70	8.67	9.28	7.12	3.6[illegible]
Gas per H.P. per hour, cu. ft..	21.55	20.12	20.73	88.08	114.8[illegible]
Water per H.P. per hour, lbs.	54.7	44.4	43.8	58.3	
Temp. water entering, F.......	51°	51°	51°	48°	
" " effluent..........	135°	144°	172°	144°	

The gas volume is reduced to 32° F. and 30 in barometer. A 50-H. working 35 to 40 effective H.P. with Dowson generator consum English anthracite per hour, equal to 1.48 to 1.3 lbs. per effective H. H.P. engine working [illegible] H.P. used 19.4 cu. ft. of gas per effective

320-H.P. [illegible]ine.—The flour-mills of M. Leblanc [illegible] with a 320-horse-power fuel-gas [illegible] the machine gives 450 horse-po[illegible] piston-stroke is 40 in.; and the

. Special arrangements have been devised in order to keep the parts of the machine at appropriate temperatures. The coal used lb. per indicated or 1.03 lb. per brake horse-power. The water used llons per brake horse-power per hour.

of an Otto Gas-engine. (*Jour. F. I.*, Feb. 1890, p. 115.)—En- I.P. nominal; working capacity of cylinder .2594 cu. ft.; clearance 796 cu. ft.

	° F.
ature of gas supplied..	62.2
" " exhaust...	774.3
" entering water	50.4
" exit water....	89.2
e of gas, in. of water..	3.06
ion per min., av'ge....	161.6
ons missed per min., ge....	6.8
ffective pressure, lbs. . in.	59.
ower, indicated.......	4.94
per explosion, foot- s	2204.
ons per minute........	74.
d per I.H.P. per hour, ...	23.4

Heat-units.	Per cent of Heat received.
Transferred into work..........	22.84
Taken by jacket-water.........	49.91
" " exhaust...............	27.22

Composition of the gas:

	By Volume.	By Weight.
CO_2	0.50%	1.923%
C_2H_4...........	4.32	10.520
O	1.00	2.797
CO	5.33	15.419
CH_4	27.18	38.042
H	51.57	9.021
N	9.06	22.273
	99.96	99.995

peratures and Pressures developed in a Gas-engine. n the Gas-engine.)—Mixtures of air and Oldham coal-gas. Temper- efore explosion, 17° C.

Mixture.		Max. Press above Atmos., lbs. per sq. in.	Temp. of Explosion calculated from observed Pressure.	Theoretical Temp. of Explosion if all Heat were evolved.
.	Air.			
1.	14 vols.	40.	806° C.	1786° C.
	13 "	51.5	1033	1912
	12 "	60.	1202	2058
	11 "	61.	1220	2228
	9 "	78.	1557	2670
	7 "	87.	1733	3334
	6 "	90.	1792	3808
	5 "	91.	1812	....
	4 "	80.	1595	....

of the Clerk Gas-engine. (Proc. Inst. C. E. 1882, vol. lxix.)— 6 × 12 in., 150 revs. per min.; mean available pressure 70.1 lbs., 9 maximum pressure, 220 lbs. per sq. in. above atmosphere; pressure gnition, 41 lbs. above atm.; temperature before compression 60° compression, 313° F.; temperature after ignition calculated from , 2800° F.; gas required per I.H.P. per hour, 22 cu. ft.

bustion of the Gas in the Otto Engine.—John Imray, in on of Mr. Clerk's paper on Theory of the Gas-engine, says: The which Mr. Otto introduced, and which rendered the engine a success, t, instead of burning in the cylinder an explosive mixture of gas and urned it in company with, and arranged in a certain way in respect ge volume of incombustible gas which was heated by it, and which ed the speed of combustion. W. R. Bousfield, in the same discus- ys: In the Otto engine the charge varied from a charge which was osive mixture at the point of ignition to a charge which was merely fluid near the piston. When ignition took place there was n explo- se to the point of ignition that was gradually communicated through mass of the cylinder. *As the ignition got farther away from the point of ignition the rate of transmission* became slower, and if the *ere not worked too fast the* ignition should gradually ce *n during its travel,* all the combustible gas being thus *y of slow combustion is*, however, disputed by Mr. *the whole quantity of* combustible gas is ignited in a

Carburetted Air in Gas-engines.—Air

gasoline or volatile petroleum spirit of low sp. gr., 0.65 to 0.70, [illegible] some of the gasoline, and the air thus saturated with vapor is equal in [illegible] ing or lighting power to ordinary coal-gas. It may therefore be used [illegible] fuel for gas-engines. Since the vapor is given off at ordinary temperat[illegible] gasoline is very explosive and dangerous, and should be kept in an [illegible] ground tank out of doors. A defect in the use of carburetted air for [illegible] engines is that the more volatile products are given off first, leaving an [illegible] residue which is often useless. Some of the substances in the oil that [illegible] taken up by the air are apt to form troublesome deposits and incrustati[illegible] when burned in the engine cylinder.

The Otto Gasoline-engine. (*Eng'g News*, May 4, 189[illegible]) [illegible] claimed that where but a small gasoline-engine is used and the gas[illegible] bought at retail the liquid fuel will be on a par with a steam-engine [illegible] lbs. of coal per horse-power per hour, and coal at $3.50 per ton, and [illegible] besides save all the handling of the solid fuel and ashes, as well as the [illegible] tendance for the boilers. As very few small steam-engines consume [illegible] than 6 lbs. of coal per hour, this is an exceptional showing for economy [illegible] 8 cts. per gallon for gasoline and 1/10 gal. required per H.P. per hour, [illegible] cost per H.P. per hour will be 0.8 cent.

The Priestman Petroleum-engine. (*Jour. Frank. Inst.*, [illegible] 1893.)—The following is a description of the operation of the engine: [illegible] ordinary high-test (usually 150° test) oil is forced under air-pressure [illegible] atomizer, where the oil is met by a current of air and broken up into [illegible] and sprayed into a mixer, where it is mixed with the proper proportio[illegible] supplementary air and sufficiently heated by the exhaust from the cyl[illegible] passing around this chamber. The mixture is then drawn by suction [illegible] the cylinder, where it is compressed by the piston and ignited by an ele[illegible] spark, a governor controlling the supply of oil and air proportionate [illegible] the work performed. The burnt products are discharged through an [illegible] haust-valve which is actuated by a cam. Part of the air supports the [illegible] bustion of the oil, and the heat generated by the combustion of [illegible] expands the air that remains and the products resulting from the expl[illegible] and thus develops its power from air that it takes in while runnin[illegible] other words, the engine exerts its power by inhaling air, heating tha[illegible] and expelling the products of combustion when done with. In the [illegible] engines only the 1/250 part of a pint of oil is used at any one time, [illegible] the smallest sizes the fuel is prepared in correct quantities varying [illegible] 1/7000 of a pint upward, according to whether the engine is running [illegible] or full duty. The cycle of operations is the same as that of the Ott[illegible] engine.

Trials of a 5-H.P. Priestman Petroleum-engine. [illegible] W. C. Unwin, Proc. Inst. C. E. 1892.)—Cylinder, 8½ × 12 in., making [illegible] 200 revs. per min. Two oils were used, Russian and American. Th[illegible] important results were given in the following table:

	Trial V. Full Power.	Trial I. Full Power.	Trial IV. Full Power.	Trial II. Half Power.
Oil used	Day-light.	Russo-lene.	Russo-lene.	Russo-lene.
Brake H.P.	7.722	6.765	6.882	3.62
I.H.P.	9.369	7.408	8.332	4.70
Mechanical efficiency	0.824	0.91	0.876	0.769
Oil used per brake H.P. hour, lb.	0.842	0.946	0.988	1.381
Oil used per indicated H.P. hour, lb.	0.694	0.864	0.816	1.063
Lb. of air per lb. of oil	33.4	31.7	43.2	21.7
Mean explosion pressure, lbs. per sq. in.	151.4	134.3	128.5	48.5
Mean compression pressure, lbs. per sq. in.	35.0	27.6	26.0	14.8
Mean terminal pressure, lbs. per sq. in.	35.4	23.7	25.5	15.5

To compa[illegible] [illegible] sumption with that of a steam-en[illegible] oil might be [illegible] valent to 1½ lbs. of coal. Then the [illegible] ngine was equivalent, in Trials I., IV., and V., to 1.42 lbs., 1.48 lbs., s. of coal per brake horse-power per hour. From Trial IV. the alues of the expenditure of heat were obtained:

	Per cent.
work at brake	13.31
friction	2.81
hown on indicator-diagram	16.12
ed in jacket-water	47.54
in exhaust-gases	26.72
ion and unaccounted for	9.61
tal	99.99

ha-engines are in use to some extent in small yachts and The naphtha is vaporized in a boiler, and the vapor is used ex- n the engine-cylinder, as steam is used; it is then condensed and the boiler. A portion of the naphtha vapor is used for fuel un- iler. According to the circular of the builders, the Gas Engine Co. of New York, a 2-H.P. engine requires from 3 to 4 quarts of er hour, and a 4-H.P. engine from 4 to 6 quarts. The chief advan- e naphtha-engine and boiler for launches are the saving of weight ickness of operation. A 2-H.P. engine weighs 200 lbs., a 4-H.P. 300 kes only about two minutes to get under headway. (Modern 1, p. 270.)

r (or Caloric) Engines.—Hot-air engines are used to some their bulk is enormous compared with their effective power. For t of the largest hot-air engine ever built (a total failure) see Life of Ericsson. For theoretical investigaton, see Rankine's ine and Rontgen's Thermodynamics. For description of con- see Appleton's Cyc. of Mechanics and Modern Mechanism, and 1 Substitutes for Steam, Trans. A. S. M. E., vii., p. 693.

f a Hot-air Engine (Robinson).—A vertical double-cylinder gine Co.'s) 12 nominal H.P. engine gave 20.19 I.H.P. in the work- r and 11.38 I.H.P. in the pump, leaving 8.81 net I.H.P.; while the ake H.P. was 5.9, giving a mechanical efficiency of 67%. Con- of coke, 3.7 lbs. per brake H.P. per hour. Mean pressure on 7 lbs. per square inch, and in pumps 15.9 lbs., the area of working eing twice that of the pumps. The hot air supplied was about d that rejected at end of stroke about 890° F.

result of Stirling's heat-engine was 2.7 lbs. per brake H.P. per ey's hot-air engine, 2 H.P. nominal, gave 4.2 I.H.P., 2.6 B.H.P.; l efficiency 62%; estimated temperature at highest pressure 1500° atmospheric pressure 700° F. Highest pressure, 14 lbs. per square e atmosphere. Consumption of fuel, 7 lbs. per hour per brake of cooling water, 30 lbs.

LOCOMOTIVES.

ncy of Locomotives and Resistance of Trains. Henderson, Proc. Engrs. Club of Phila. 1886.)—The efficiency of s can be divided into two principal parts: the first depending ze of the cylinders and wheels, the valve-gear, boiler and steam- f which the tractive power is a function; and the second upon grade, curvature, and friction, which combine to produce the

ive power may be determined as follows:

ractive power;
average effective pressure in cylinder;
troke of piston;
iameter of cylinders;
ameter of driving-wheels. Then

$$P = \frac{4\pi d^2 pS}{4\pi D} = \frac{d^2 pS}{D}.$$

The average effective pressure can be obtained from an indicator-diagram, or by calculation, when the initial pressure and ratio of expansion are known, together with the other properties of the valve-motion. The subjoined table from "Auchincloss" gives the proportion of mean effective pressure to boiler-pressure above atmosphere for various proportions of cut-off.

Stroke, Cut off at—	M.E.P. (Boiler-pres. = 1).	Stroke, Cut off at—	(M.E.P. Boiler-pres. = 1).	Stroke, Cut off at—	M.E.P. (Boiler-pres. = 1).
.1	.15	.333 = ⅓	.5 = ½	.625 = ⅝	.79
.125 = ⅛	.2	.375 = ⅜	.55	.666 = ⅔	.82
.15	.24	.4	.57	.7	.85
.175	.28	.45	.62	.75 = ¾	.89
.2	.32	.5 = ½	.67	.8	.93
.25 = ¼	.4	.55	.72	.875 = ⅞	.98
.3	.46				

These values were deduced from experiments with an English locomotive by Mr. Gooch. As diagrams vary so much from different causes, this table will only fairly represent practical cases. It is evident that the cut-off must be such that the boiler will be capable of supplying sufficient steam at the given speed.

In the following calculations it is assumed that the adhesion of the engine is at least equal to the tractive power, which is generally the case—if the engine be well designed—except when starting, or running at a very low rate of speed, with a small expansive ratio. When running faster, economy, and also the size of the boiler, necessitate a higher ratio of expansion, thus reducing the tractive power below the adhesion. If the adhesion be less than the tractive power, substitute it for the latter in the following formulæ.

The resistances can be computed in the following manner, first considering the train:

There is a resistance due to friction of the journals, pressure of wind, etc., which increases with the speed. Most of the experiments made with a view of determining the resistance of trains have been with European rolling-stock and on European railways. The few trials that have been made here seem to prove that with American systems this resistance is less.

The following table gives the resistance at different speeds, assumed for American practice:

Speed in miles per hour: $s =$	5	10	15	20	25	30	35	40	45	50	55	60
Resistance in pounds per ton of 2240 lbs.: $y =$	3.1	3.4	4.	4.8	5.8	7.1	8.6	10.2	12.1	14.3	16.8	19.2
Coefficient of resistance in terms of load: $l =$	.0015	.0017	.0020	.0024	.0029	.0035	.0043	.0051	.0060	.0071	.0084	.0096

$$l = .0015\left(1 + \frac{s^2}{650}\right).$$

The resistance due to curvature is about .5 lb. per ton per degree of curvature, or the coefficient = .00025c, where c = the curvature in degrees.

The effect of grades may be determined by the theory of the inclined plane.

Consider a load L on a grade of m feet per mile. The component of the weight L acting in the line of traction, or parallel to the track, is

$$L \sin \theta = \frac{Lm}{5280} = .00019Lm.$$

To combine these coefficients in one equation representing the resistance of the train:

Let L = weight of train, exclusive of engine, in pounds;
R = resistance of train, in pounds.
s, c, and m, as above. Then

$$R = L\left[.0015\left(1 + \frac{s^2}{650}\right) + .00025c \pm .00019m\right].$$

sign meaning that this coefficient is positive for ascending and nega- r descending grades.

nd a grade upon which a train would descend by itself, take the last ient minus and make $R = 0$, whence

$$m = 7.9\left(1+\frac{s^2}{650}\right) + 1.3c.$$

ocomotives usually have a long rigid wheel-base, the coefficient for ure had better be doubled. The resistance due to the friction of the g parts will be considered as being proportional to the tractive power, the effective tractive power will be represented by uP, the resistance $1-u)P$.

bining all these values, there results the equation between the trac- wer and the weight of the train and engine:

$$uP - W(.0005c \pm .00019m) = Ll + .00025c \pm .00019m,$$

g weight of engine and tender, and u being probably about .8. sforming, we have

$$L = \frac{uP - W(.0005c \pm .00019m)}{l + .00025c \pm .00019m},$$

$$P = \frac{L(l + .00025c \pm .00019m) + W(.0005c \pm .00019m)}{u}.$$

deductions, says Mr. Henderson, agree well with railroad practice. igures given above for resistances are very much less than those y the old formulæ (which were certainly wrong), but even Mr. Hen- s figures for high speed are too high, according to a diagram given by arnes in *Eng'g Mag.*, June, 1894, from which the following figures are :

ed, miles per hour..............	50	60	70	80	90	100
istance, pounds per gross ton..	12	12.4	13.5	15	17	20

News, March 8, 1894, gives a formula which for high speeds gives for resistance between those of Mr. Barnes and Mr. Henderson. See ported in *Eng'g News* of June 9, 1892. The formula is, resistance in per ton = ¼ velocity in miles per hour + 2. This gives for

.....	5	10	15	20	25	30	35	40	45	50	60	70	80	90	100
nce..	3¼	4.5	5¾	7	8¼	9.5	10¾	12	13¼	14.5	17	19.5	22	24.5	27

bles showing that the resistance varies with the area exposed to the ice and friction of the air per ton of load, see Dashiell, Trans. A. S. ol. xiii. p. 371.

tia and Resistances of Railroad Trains at Increasing s.—A series of tables and diagrams is given in *R. R. Gaz.*, Oct. 31, show the resistances due to inertia in starting trains and accelerat- r speeds.

echanical principles and formulæ from which these data were cal- are as follows:

peed in miles per hour to be acquired at the end of a mile.

$=$ average speed in miles per hour during the first mile run.

velocity in feet per second at the end of a mile; then $V \div 2 =$ aver- ocity in feet per second during the first mile run.

$\div V/2 =$ time in seconds required to run first mile $= 10560 \div V$.

$(10560 \div V) = V^2 \div 10560 = .0000947V^2 =$ Constant gain in velocity or ation in feet per second necessary to the acquirement of a velocity V nd of a mile.

cceleration due to the force of gravity, i.e., 32.2 feet per second.

orces required to accelerate a given mass in a given time to different es are in proportion to those velocities. The weight of a body is the e of the force which accelerates it in the case of gravity, and as we sidering 1 lb., or *the unit of* weight, as the mass to be accelerated, *g: $(V^2 \div 10560)$: : 1 is to the force required to accelerate 1 lb. to the V at the end of a mile run*, or, what is the same, to accelerate it at *of $V^2 \div 10560$ feet per second.*

his the pull on the drawbar—it is the same as the *d is properly termed* the inertia—in pounds per $V^2 \div (10560 \times 32.2)$, which equals $.00000294V^2$.

tractive efficiency of locomotives; With simple two-cyl
ing four wheels coupled, experiments have been made b
tive superintendent of the Eastern Railway of France, M
greatest possible care and with the best apparatus, and th
was that out of 100 I.H.P in the cylinders 43 H.P. only w
draw-bar. The loss of 57% was rather a high price to pa
of the engine. How much of that loss was due to co
could yet say; but a considerable amount of it must be
cause it was known that large engines with a single pai
not coupled were doing their work more economically, w
motive engineers who had not yet gone in for compoundi
going back to the single pair of driving-wheels. Moreov
loss of 57% had been confirmed independently on the Pen
trials made with an engine having $18\frac{1}{4} \times 24$-in. cylind
wheels four-coupled; by taking indicator diagrams up t
which were professed to be taken correctly, the powe
was found to be only 42% of that in the cylinders, or onl
French experiments.

The Size of Locomotive Cylinders is usuall
that the engine will just overcome the adhesion of its wh
der favorable circumstances.

The adhesion of the wheel is about one third the weig
clean and sanded, but is usually assumed at 0.25. (Thurs

A committee of the American Association of Maste
studying the performance reports of the best engines, p
ing formula for weight on driving-wheels: $W = \frac{0.85C\ldots}{D}$
mean pressure in the cylinder is taken at 0.85 of the
starting, C is a numerical coefficient of adhesion, d the d
in inches, D that of the drivers in inches, P the pressu
pounds per square inch, S the stroke of piston in inches
for passenger engines, 0.24 for freight, and 0.22 for "swi

The common builder's rule for determining the size o
locomotive is the following, in which we accept Mr. Fo
that the steam-pressure at the engine may be taken as

'on Borries's rule for the diameter of the low-pressure cylinder of a com-
ind locomotive is $d^2 = \frac{2ZD}{ph}$,

ere d = diameter of l.p. cylinder in inches;
D = diameter of driving-wheel in inches;
p = mean effective pressure per sq. in., after deducting internal machine friction;
h = stroke of piston in inches;
Z = tractive force required, usually 0.14 to 0.16 of the adhesion.

he value of p depends on the relative volume of the two cylinders, and n indicator experiments may be taken as follows:

ass of Engine.	Ratio of Cylinder Volumes.	p in percentage of Boiler-pressure.	p for Boiler-pressure of 176 lbs.
ge-tender eng's	1 : 2 or 1 : 2.05	42	74
k-engines......	1 : 2 or 1 : 2.2	40	71

he Size of Locomotive Boilers. (Forney's Catechism of the omotive.)—They should be proportioned to the amount of adhesive ght and to the speed at which the locomotive is intended to work. Thus comotive with a great deal of weight on the driving-wheels could pull a vier load, would have a greater cylinder capacity than one with little ad- ve weight, would consume more steam, and therefore should have a er boiler.

ie weight and dimensions of locomotive boilers are in nearly all cases rmined by the limits of weight and space to which they are necessarily ined. It may be stated generally that *within these limits a locomotive er cannot be made too large.* In other words, boilers for locomotives ıld always be made as large as is possible under the conditions that de- ıine the weight and dimensions of the locomotives.

'ootten's Locomotive. (Clark's Steam-engine; see also Jour. nk. Inst. 1891, and Modern Mechanism, p. 485.)—J. E. Wootten designed constructed a locomotive boiler for the combustion of anthracite and ite, though specially for the utilization as fuel of the waste produced in mining and preparation of anthracite. The special feature of the engine ie fire-box, which is made of great length and breadth, extending clear the wheels, giving a grate-area of from 64 to 85 sq. ft. The draught ısed over these large areas is so gentle as not to lift the fine pa-ticles of fuel. A number of express-engines having this type of boiler are engaged he fast trains between Philadelphia and Jersey City. The fire-box shell ft. 8 in. wide and 10 ft. 5 in. long; the fire-box is 8×9½ ft., making 76 sq. f grate-area. The grate is composed of bars and water-tubes alternately, regular types of cast-iron shaking grates are also used. The height of fire-box is only 2 ft. 5 in. above the grate. The grate is terminated by idge of fire-brick, beyond which a combustion-chamber, 27 in. long, s to the flue-tubes, about 184 in number, 1¾ in. diam. The cylinders are ı. diam., with a stroke of 22 inches. The driving-wheels, four-coupled, 5 ft. 8 in. diam. The engine weighs 44 tons, of which 29 tons are on driv- wheels. The heating-surface of the fire-box is 135 sq. ft., that of the tubes is 982 sq. ft.; together, 1117 sq. ft., or 14.7 times the grate-area. ling 15 passenger-cars, weighing with passengers 360 tons, at an average d of 42 miles per hour, over ruling gradients of 1 in 89, the engine con- es 62 lbs. of fuel per mile, or 34¼ lbs. per sq. ft. of grate per hour.

ualities Essential for a Free-steaming Locomotive. m a paper by A. E. Mitchell, read before the N. Y. Railroad Club; *g News*, Jan. 24, 1891.)—Square feet of boiler-heating surface for bitu- ous coal should not be less than 4 times the square of the diameter in es of a cylinder 1 inch larger than the cylinder to be used. One tenth is should be in the fire-box. On anthracite locomotives more heating- ice is required in the fire-box, on account of the larger grate-area ired, but the heating-surface of the flues should not be materially eased.

'ate-surface, Smoke-stacks, and Exhaust-nozzles for omotives. (*Am. Mach.*, Jan. 8, 1891.)—For grate-surface for anthr *oal: Multiply the displacement in cubic feet of one piston during by 8.5; the product will be the area of the grate in square fee bituminous coal: Multiply the displacement in feet of a stroke by 6½; the product will be the grate-area in squ with cylinders 12 in. in diameter and upwards.* For smaller cylinders the ratio of grate-area to piston-displacement should be 7½ to 1, or even more, if the design of the engine will admit this proportion.

The grate-areas in the following table have been found by the foregoing rules, and agree very closely with the average practice:

Smoke-stacks.—The internal area of the smallest cross-section of the stack should be 1/17 of the area of the grate in soft-coal-burning engines.

A. E. Mitchell, Supt. of Motive Power of the N. Y. L. E. & W. R. R., says that recent practice varies from this rule. Some roads use the same size of stack, 13½ in. diam. at throat, for all engines up to 20 in. diam. of cylinder.

The area of the orifices in the exhaust-nozzles depends on the quantity and quality of the coal burnt, size of cylinder, construction of stack, and the condition of the outer atmosphere. It is therefore impossible to give rules for computing the exact diameter of the orifices. All that can be done is to give a rule by which an approximate diameter can be found. The exact diameter can only be found by trial. Our experience leads us to believe that the area of each orifice in a double exhaust-nozzle should be equal to 1/400 part of the grate-surface, and for single nozzles 1/200 of the grate-surface. These ratios have been used in finding the diameters of the nozzles given in the following table. The same sizes are often used for either hard or soft coal-burners.

Size of Cylinders, in inches.	Grate-area for Anthracite Coal, in sq. in.	Grate-area for Bituminous Coal, in sq. in.	Diameter of Stacks, in inches.	Double Nozzles. Diam. of Orifices, in inches.	Single Nozzles. Diam. of Orifices, in inches.
12 × 20	1591	1217	9½	2	2 13/16
13 × 20	1873	1432	10½	2⅛	3
14 × 20	2179	1666	11¼	2 5/16	3¼
15 × 22	2742	2097	12½	2 9/16	3 11/16
16 × 24	3415	2611	14	2⅞	4 1/16
17 × 24	3856	2948	15	3 1/16	4 5/16
18 × 24	4321	3304	15¾	3¼	4⅝
19 × 24	4810	3678	16½	3 7/16	4 13/16
20 × 24	5337	4081	17½	3⅝	5 1/16

Exhaust-nozzles in Locomotive Boilers.—A committee of the Am. Ry. Master Mechanics' Assn. in 1890 reported that they had, after two years of experiment and research, come to the conclusion that, owing to the great diversity in the relative proportions of cylinders and boilers, together with the difference in the quality of fuel, any rule which does not recognize each and all of these factors would be worthless.

The committee was unable to devise any plan to determine the size of the exhaust-nozzle in proportion to any other part of the engine or boiler, and believes that the best practice is for each user of locomotives to adopt a nozzle that will make steam freely and fill the other desired conditions, best determined by an intelligent use of the indicator and a check on the fuel account. The conditions desirable are: That it must create draught enough on the fire to make steam, and at the same time impose the least possible amount of work on the pistons in the shape of back pressure. It should be large enough to produce a nearly uniform blast without lifting or tearing the fire, and be economical in its use of fuel.

Fire-brick Arches in Locomotive Fire-boxes.—A committee of the Am. Ry. Master Mechanics' Assn. in 1890 reported strongly in favor of the use of brick arches in locomotive fire-boxes. They say: It is the unanimous opinion of all who use bituminous coal and brick arch, that it is most efficient in consuming the various gases composing black smoke, and by impeding and delaying their passage through the tubes, and mingling and subjecting them to the heat of the furnace, greatly lessens the volume ejected, and intensifies combustion, and does not in the least check but rather augments draught, with the consequent saving of fuel and increased steaming capacity that might be expected from such results. This *in particular* when used in connection with extension front.

Size, Weight, Tractive Power, etc., of Different Sizes of Locomotives. (J. G. A. Meyer, Modern Locomotive Construction, &c.

Aug. 8, 1885.)—The tractive power should not be more or less than esion. In column 3 of each table the adhesion is given, and since the n and tractive power are expressed by the same number of pounds, gures are obtained by finding the tractive power of each engine, for pose always using the small diameter of driving-wheels given in 2. The weight on drivers is shown in column 4, which is obtained by ying the adhesion by 5 for all classes of engines. Column 5 gives the on the trucks, and these are based upon observations. Thus, the on the truck for an eight-wheeled engine is about one half of that on the drivers.

ogul engines we multiply the total weight on drivers by the decimal the product will be the weight on the truck.

n-wheeled engines the total weight on the drivers, multiplied by the .32, will be equal to the weight on the truck.

astly, for consolidation engines, the total weight on drivers multi- the decimal .16, will determine the weight on the truck.

umn 6 the total weight of each engine is given, which is obtained by the weight on the drivers to the weight on the truck. Dividing the n given in column 1 by 7½ will give the number of tons of 2000 lbs. engine is capable of hauling on a straight and level track, column 7. eight of engines given in these tables will be found to agree gen- vith the actual weights of locomotives recently built, although it ot be expected that these weights will agree in every case with the veights, because the different builders do not build the engines alike. ctual weight on trucks for eight-wheeled or ten-wheeled engines will r much from those given in the tables, because these weights depend on the difference between the total and rigid wheel-base, and these often changed by the different builders. The proportion between l and total wheel-base is generally the same.

ule for finding the tractive power is :

$$\frac{\left\{\begin{matrix}\text{e of dia. of}\\ \text{in inches}\end{matrix}\right\} \times \left\{\begin{matrix}\text{Mean effect. steam}\\ \text{press. per sq. in.}\end{matrix}\right\} \times \left\{\begin{matrix}\text{stroke}\\ \text{in feet}\end{matrix}\right\}}{\text{Diameter of wheel in feet.}} = \text{tractive power.}$$

HT-WHEELED LOCOMOTIVES.

wheels.	Adhesion.	Weight on Drivers.	Weight on Truck.	Total Weight.	Hauling Capacity on Level Track in tons of 2000 lbs., including Tender.
2	3	4	5	6	7
n.	lbs.	lbs.	lbs.	lbs.	
-51	4000	20000	10000	30000	533
-51	5324	26620	13310	39930	709
-54	5940	29700	14850	44550	792
-57	6828	34140	17070	51210	910
-61	7697	38485	19242	57727	1026
-66	8836	44180	22090	66270	1178
-66	9533	47665	23832	71497	1271
-66	10404	52020	26010	78030	1387
-66	11472	57360	28680	86040	1529

TEN-WHEELED ENGINES.

Cylinders—Diameter, Stroke.	Diameter of Driving-wheels.	Adhesion.	Weight on Drivers.	Weight on Truck.	Total Weight, with Water and Fuel.	Hauling Capacity on Level Track in tons of 2000 lbs., including Tender.
1	2	3	4	5	6	7
in.	in.	lbs.	lbs.	lbs.	lbs.	
12×18	39-43	5981	29907	9570	39477	797
13×18	41-45	6677	33387	10683	44070	890
14×20	43-47	8205	41023	13127	54150	1093
15×22	45-50	9900	49500	15840	65340	1320
16×24	48-54	11520	57600	18432	76032	1536
17×24	51-56	12240	61200	19584	80784	1632
18×24	51-56	13722	68611	21955	90566	1829
19×24	54-60	14440	72200	23104	95304	1925

MOGUL ENGINES.

n.	lbs.	lbs.	lbs.	lbs.	
-40	4978	24891	4978	29869	663
-41	6480	32400	6480	38880	864
-42	7399	36997	7399	44396	986
-43	9046	45230	9046	54276	1206
47	10607	53035	10607	63642	1414
51	12288	61440	12288	73738	1638
4	12739	63697	12739	76436	1698
[illegible]	13722	68611	13722	82333	1829
[illegible]	14440	72200	14440	86640	1925

CONSOLIDATION ENGINES.

in.	in.	lbs.	lbs.	lbs.	lbs.	
14×16	36-38	7840	39200	6272	45472	10
15×18	36-38	10125	50625	8100	58725	13
20×24	48-50	18000	90000	14400	104400	[illegible]
22×24	50-52	20909	104544	16727	121271	[illegible]

Leading American Types of Locomotive for Freight and Passenger Service.

1. The eight-wheel or "American" passenger type, having four coupled driving-wheels and a four-wheeled truck in front.
2. The "ten-wheel" type, for mixed traffic, having six coupled drivers and a leading four-wheel truck.
3. The "Mogul" freight type, having six coupled driving-wheels and a pony or two-wheel truck in front.
4. The "Consolidation" type, for heavy freight service, having eight coupled driving-wheels and a pony truck in front.

Besides these there is a great variety of types for special conditions of service, as four-wheel and six-wheel switching-engines, without trucks; the Forney type used on elevated railroads, with four coupled wheels under the engine and a four-wheeled rear truck carrying the water-tank and fuel; locomotives for local and suburban service with four coupled driving-wheels with a two-wheel truck front and rear, or a two-wheel truck front and a four-wheel truck rear, etc. "Decapod" engines for heavy freight service have ten coupled driving-wheels and a two-wheel truck in front.

Steam-distribution for High-speed Locomotives.

(C. H. Quereau, *Eng'g News*, March 8, 1894.)

Balanced Valves.—Mr. Philip Wallis, in 1886, when Engineer of Tests for the C., B. & Q. R. R., reported that while 6 H.P. was required to work unbalanced valves at 40 miles per hour, for the balanced valves 2.2 H.P. only was necessary.

Effect of Speed on Average Cylinder-pressure.—Assume that a locomotive has a train in motion, the reverse lever is placed in the running notch, and the track is level; by what is the maximum speed limited? The resistance of the train and the load increase, and the power of the locomotive decreases with increasing speed till the resistance and power are equal, when the speed becomes uniform. The power of the engine depends on the average pressure in the cylinders. Even though the cut-off and boiler pressure remain the same, this pressure decreases as the speed increases because of the higher piston-speed and more rapid valve-travel the steam has a shorter time in which to enter the cylinders at the higher speed. The following table, from indicator-cards taken from a locomotive at varying speeds, shows the decrease of average pressure with increasing speed:

Miles per hour	46	51	51	53	54	57	60	[illegible]
Speed, revolutions	224	248	248	258	263	277	292	[illegible]
Average pressure per sq. in.:								
Actual	51.5	44.0	47.3	43.0	41.3	42.5	37.3	[illegible]
Calculated	....	46.5	46.5	44.7	43.8	41.6	39.5	[illegible]

The "average pressure calculated" was figured on the assumption that the mean effective pressure would decrease in the same ratio that the speed increased. The main difference lies in the higher steam-line at the lower speeds, and consequent higher expansion-line, showing that more steam entered the cylinder. The back pressure and compression-lines agree quite closely for all the cards, though they are slightly better for the slower speeds. That the difference is not greater may safely be attributed to the large exhaust-ports, passages, and exhaust tip, which is 5 in. diameter. These are matters of great importance for high speeds.

Boiler-pressure.—The increase of train resistance with increased speed is not as the square of the velocity, as is commonly supposed. It is more likely that it increases as the speed after about 20 miles an hour is reached. Assuming that the latter is true, and that an average of 50 lbs. per square inch is the greatest that can be realized in the cylinders of a given engine at 40 miles an hour, and that this pressure furnishes just sufficient power to keep the train at this speed, it follows that, to increase the speed to 50 miles, the mean effective pressure must be increased in the same proportion. To increase the capacity for speed of any locomotive its power must be increased at least by as much as the speed is to be increased. One way to accomplish this is to increase the boiler-pressure. That this is generally realized is shown by the increase in boiler-pressure in the last ten years. Ten... ...ree single-expansion locomotives described in the railway journals ... steam-pressures are as follows: 3, 160 lbs.; 4, 165 lbs.; 2, 170 ... 1, 190 lbs.

Valve-travel. — An increased average cylinder-pressure may also be
btained by increasing the valve-travel without raising the boiler-pressure
nd better results will be obtained by increasing both. The longer trave
ives a higher steam-pressure in the cylinders, a later exhaust-opening
ter exhaust-closure, and a larger exhaust-opening—all necessary for high
eeds and economy. I believe that a 20-in. port and 6½-in. (or even 7-in.)
avel could be successfully used for high-speed engines, and that frequently
so doing the cylinders could be economically reduced and the counter-
lance lightened. Or, better still, the diameter of the drivers increased,
curing lighter counterbalance and better steam-distribution.

Size of Drivers.—Economy will increase with increasing diameter of
ivers, provided the work at average speed does not necessitate a cut-off
nger than one fourth the stroke. The piston-speed of a locomotive with
in. drivers at 55 miles per hour is the same as that of one with 68-in.
ivers at 61 miles per hour.

Steam-ports.—The length of steam-ports ranges from 15 in. to 23 in., and
s considerable influence on the power, speed, and economy of the loco-
otive. In cards from similar engines the steam-line of the card from the
gine with 23-in. ports is considerably nearer boiler-pressure than that of
card from the engine with 17¼-in. ports. That the higher steam-line is
e to the greater length of steam-port there is little room for doubt. The
in. port produced 531 H.P. in an 18½-in. cylinder at a cost of 23.5 lbs. of
licated water per I.H.P. per hour. The 17¼-in. port, 424 H.P., at the rate
22.9 lbs. of water, in a 19-in. cylinder.

Allen Valves.—There is considerable difference of opinion as to the advan-
ge of the Allen ported-valve. (See *Eng. News*, July 6, 1893.)

Speed of Railway Trains.—In 1834 the average speed of trains on
Liverpool and Manchester Railway was twenty miles an hour; in 1838 it
s twenty-five miles an hour. But by 1840 there were engines on the Great
stern Railway capable of running fifty miles an hour with a train, and
hty miles an hour without. A speed of 86 miles per hour was made in
gland with the T. W. Worsdell compound locomotive. The total weight
the engine, tender, and train was 695,000 lbs.; indicator-cards were taken
owing 1068.6 H.P. on the level. At a speed of 75 miles per hour on a
el, and the same train, the indicator-cards showed 1040 H.P. developed.
ans. A. S. M. E., vol. xiii., 363.)

he limitation to the increase of speed of heavy locomotives seems at
esent to be the difficulty of counterbalancing the reciprocating parts. The
balanced vertical component of the reciprocating parts causes the pres-
e of the driver on the rail to vary with every revolution. Whenever the
eed is high, it is of considerable magnitude, and its change in direction is
rapid that the resulting effect upon the rail is not inappropriately called
hammer blow." Heavy rails have been kinked, and bridges have been
ken to their fall under the action of heavily balanced drivers revolving
high speeds. The means by which the evil is to be overcome has not yet
en made clear. See paper by W. F. M. Goss, Trans. A. S. M. E., vol. xvi.

Engine No. 999 of the New York Central Railroad ran a mile in 32 seconds,
ual to 112 miles per hour, May 11, 1893.

$$\left.\begin{array}{l}\textit{eed in miles}\\ \textit{per hour}\end{array}\right\} = \frac{\text{circum. of driving-wheels in in.} \times \text{no. of rev. per min.} \times 60}{63{,}360}$$

$$= \text{diam. of driving-wheels in in.} \times \text{no. of rev. per min.} \times .003$$

(approximate, giving result 8/10 of 1 per cent too great).

DIMENSIONS OF SOME LARGE AMERICAN LOCOMOTIVES, 1893.

he four locomotives described below were exhibited at the Chicago
position in 1893. The dimensions are from *Engineering News*, June, 1893.
e first, or Decapod engine, has ten-coupled driving-wheels. It is one of
heaviest and most powerful engines ever built for freight service. The
iladelphia & Reading engine is a new type for passenger service, with four-
pled drivers. The Rhode Island engine has six drivers, with a 4-whe
ing truck and a 2-wheel trailing truck. These three engines have
pound cylinders. The fourth is a simple engine, of the standard Am
8-wheel type, 4 driving-wheels, and a 4-wheel truck in front.
e holds the world's record for speed (1893) for short distances, b
mile in 32 seconds.

	Baldwin. N. Y., L. E. & W. R. R. Decapod Freight.	Baldwin. Phila. & Read. R. R. Express Passenger.	Rhode Isl. Locomoti'e Works. Heavy Express.
Running-gear:			
Driving-wheels, diam	4 ft. 2 in.	6 ft. 6 in.	6 ft. 6 in.
Truck " "	2 " 6 "	4 " 0 "	2 " 9 "
Journals, driving-axles...	9 × 10 in.	8½ × 12 in.	8 × 8¾ in.
" truck- " ...	5 × 10 "	6½ × 10 "	5½ × 10 "
" tender- " ...	4½ × 9 "	4½ × 8 "	4¼ × 8 "
Wheel-base:			
Driving....................	18 ft. 10 in.	6 ft. 10 in.	13 ft. 6 in.
Total engine..............	27 " 3 "	23 " 4 "	29 " 9¼ "
" tender....	16 " 8 "	16 " 0 "	15 " 0 "
" engine and tender...	53 " 4 "	47 " 3 "	50 " 6¾ "
Wt. in working-order:			
On drivers.........	170,000 lbs.	82,700 lbs.	88,500 lbs.
On truck-wheels......... .	29,500 "	47,000 "	54,500 "
Engine, total........... .	192,500 "	129,700 "	143,000 "
Tender "	117,500 "	80,573 "	75,000 "
Engine and tender, loaded	310,000 "	210,273 "	218,000 "
Cylinders:			
h.p. (2)...............	16 × 28 in.	13 × 24 in.	one 21 × 26
l.p. (2)....................	27 × 28 "	22 × 24 "	one 31 × 26
Distance centre to centre.	7 ft. 5 "	7 ft. 4½ in.	7 ft. 1 in.
Piston-rod, diam..... ...	4 in.	3½ in.	3½ in.
Connecting-rod, length...	9′ 8 7/16″	8 ft. 0½ in.	10 ft. 3½ in.
Steam-ports....	28½ × 2 in.	24 × 1½ in.	1½ × 20 and 1½ × 25
Exhaust-ports....	28½ × 8 "	24 × 4½ "	3 × 20 in.
Slide-valves, out. lap, h.p.	⅞ in.	⅞ in.	1¼ in.
" " out. lap, l.p..	⅝ "	⅝ "	1 in.
" " in. lap, h.p...	.. ..	(neg.) ⅛ in.	....
" " in. lap, l.p...	....	None	....
" " max. travel .	6 in.	5 in.	6¼ in.
" " lead, h.p.....	1/16 in.	⅛ "	3/32 "
" " lead, l.p.....	5/16 "	⅜ "	....
Boiler—Type..........	Straight	Straight	Wagon top
Diam. of barrel inside....	6 ft. 2½ in.	4 ft. 8¼ in.	5 ft. 2 in.
Thickness of barrel-plates	¾ in.	⅝ in.	⅝ in.
Height from rail to centre line	8 ft. 0 in.	....	8 ft. 11 in.
Length of smoke-box.....	5 " 7⅞ "	....	6 " 1 "
Working steam-pressure..	180 lbs.	180 lbs.	200 lbs.
Firebox—type............	Wootten	Wootten	Radial stay
Length inside.............	10′ 11 9/16″	9 ft. 6 in.	10 ft. 0 in.
Width "	8 ft. 2⅛ in.	8 " 0⅛ "	2 " 9⅝ "
Depth at front...........	4 " 6 "	3 " 2¾ "	6 " 10¾ "
Thickness of side plates..	5/16 in.	5/16 in.	5/16 in.
" " back plate...	5/16 "	5/16 "	⅜ "
Thickness of crown-sheet.	⅜ "	5/16 "	⅜ "
" " tube " .	½ "	½ "	½ "
Grate-area...	89.6 sq. ft.	76.8 sq. ft.	28 sq. ft.
Stay-bolts, diam., 1⅛ in..	pitch, 4¼ in.	... .	4 in.
Tubes—iron................	354	324	272
Pitch....	2¾ in.	2 1/16 in.	2¾ in.
Diam., outside...........	2 "	1½ in.	2 "
Length betw'n tube-plates	11 ft. 11 in.	10 ft. 0 in.	12 ft. 8⅝ in.
Heating-surface:			
Tubes, exterior	2,208.8 ft.	1,262 sq. ft.	....
Fire-box.................	234.3 "	173 " "	....
Miscellaneous:			
Exhaust-nozzle, diam....	5 in.	5½ in.	....
Smokestack, smal'st diam.	1 ft. 6 "	1 ft. 6 in.	1 ft. [illegible]
" height from rail	15 " 6½ "	14 ft. 0¾ in.	15 " [illegible]

Name of Railroad.	Passenger Freight En[illegible]	No. of Driv[illegible]	No. of Fr[illegible] Truck-wh[illegible]	Diam. of Dr[illegible] ing-wheels	Size of Cylinders, Inches.	Total Weig[illegible] of Engine, [illegible]	Total Weig[illegible] on Driving wheels, lb[illegible]	Area of Gra[illegible] sq. ft.	Firebox He[illegible] ing-surfac[illegible] sq. ft.	Tube Heati[illegible] surface, sq.	Steam-pres[illegible] ure per sq. [illegible] Atmospheri[illegible] lbs.	Length of Tubes, ft. and in.		Diam. of Tub[illegible] in.	Ratio of Cy[illegible] inder-power Weight, Ava[illegible] able for Ad[illegible] hesion.
C. M. & St. P.	P.	4	4	62	16 × 24	86,000	54,000	15.5	115	801	160	11	0	2	0.461
C. R. R. of N. J.	"	4	4	68	19 × 24	105,900	75,900	37	......	....	160	11	9¾	2	0.421
B. & O.	"	6	4	62	21 × 26	133,000	103,300	28.2	188.5	1846.8	160	13	2¾	2¼	0.450
C., C., C. & St. L.	"	6	4	66	18½ × 24	131,400	102,800	18.2	147	1883.4	160	13	4⅛	2	0.304
Penn. R. R.	"	4	4	78	19 × 24	123,000	81,000	26.2	144.3	1672.2	180	12	0	2	0.387
N. Y. C. & H. R.	"	4	4	84½	19 × 24	123,000	82,200	27.3	......	....	180	12	0	2	0.352
C. B. & Q.	"	4	4	68⅞	18 × 24	....	66,600	24.5	111	1301	160	....	....	...	0.426
N. Y. C. & H. R.	"	4	4	78	19 × 24	126,150	81,400	27.3	147.7	1670.7	180	12	0	2	0.385
C., C., C. & St. L.	"	4	4	68	18½ × 24	129,280	86,500	31.3	143.7	1426.3	180	11	4	2	0.395
N. Y., L. E. & W.	"	6	4	68	20 × 24	127,600	100,000	39.5	186	1885	160	13	2	2	0.355
M. C.	"	6	4	74	20 and 29 × 24	135,000	99,000	28.2	141.2	1839.5	180	13	0	....	0.389
Penn. R. R.	"	6	4	74	20 and 30 × 24	138,000	102,000	26.2	141.7	1811.5	180	13	0	2	0.404
C. R. R. of N. J.	"	4	4	78	13 and 22 × 24	123,800	88,400	38.5	......	....	180	11	10	2	0.476
Penn. R. R.	"	6	4	72	14 and 24 × 24	133,000	100,000	28.3	......	....	180	14	0	2	0.542
Philadelphia & Reading	"	4	2	78	13 and 22 × 24	129,000	83,000	76	128.9	1261.7	170	10	0	1½	0.423
C. B. & Q.	"	6	2	62	20 and 29 × 24	115,000	98,000	27.1	126	1380	160	....	...	...	0.417
C., M. & St. P.	"	6	4	78	21 and 31 × 26	143,000	88,500	......	......	....	200	12	8⅝	2	0.554
[illegible] & Pa.	F.	8	2	50¼	19 × 26	126,870	110,650	29	......	....	160	12	1⅛	2	0.424
[illegible]	"	10	0	50	22 × 28	150,300	150,300	37.5	180	2171	160	14	6	2½	0.458
[illegible] & St. L.	"	6	4	56	19 × 24	128,200	99,600	18.2	147	1883.4	160	13	4⅛	2	0.391
[illegible]	"	8	2	50	21 × 26	125,000	113,400	28.7	155	1584	160	13	7	2¼	0.507
[illegible]	"	6	4	60	20 × 26	127,500	101,500	32.5	146.2	1608	180	13	0	2¼	0.482
[illegible]	"	8	4	54	19 × 24	129,000	96,000	34.5	189.7	2212.6	180	13	6	2¼	0.472
[illegible] & Pa.	"	8	2	50¼	13 and 21 × 26	129,000	116,550	29	......	....	175	12	11½	2	0.538
[illegible]	"	6	4	55	20 and 28 × 26	126,000	99,500	28.6	141.2	1788.3	180	13	0	2¼	0.525
[illegible] & W.	"	10	2	50	16 and 27 × 28	200,550	173,700	89.6	182.5	2208	180	12	0	2	0.660
[illegible] Lebanon	"	8	2	50	14 and 24 × 28	150,000	135,000	35.3	......	....	180	13	6	2¼	0.675
[illegible]	"	4	4	62	12 and 20 × 26	122,400	87,970	18.2	......	....	175	13	7½	2¼	0.525
[illegible] Central	"	6	2	56½	20 and 29 × 26	128,500	107,300	26.1	......	....	180	11	0	2	0.512

Dimensions of Some American Locomotives.—The [illegible] page 861 is condensed from one given by D. L. Barnes, in his paper [illegible] "Distinctive Features and Advantages of American Locomotive [illegible] Trans. A.S.C.E., 1893. The formula from which column marked "[illegible] cylinder-power to weight available for adhesion" is calculated as follows:

$$\frac{2 \times \text{cylinder area} \times \text{boiler-pressure} \times \text{stroke}}{\text{Weight on drivers} \times \text{diameter of driving-wheel}}$$

(Ratio of cylinder-power of compound engines cannot be compared [illegible] that of the single-expansion engines.)

Where the boiler-pressure could not be determined from the [illegible] of the locomotives, as given by the builders and operators of the [illegible] it has been assumed to be 160 lbs. per sq. in. above the atmosphere.

For compound locomotives the figures in the last column of [illegible] based on the capacity of the low-pressure cylinders only, the volume of [illegible] high-pressure being omitted. This has been done for the purpose of [illegible] parison, and because there is no accurate simple way of comparing [illegible] cylinder-power of single-expansion and compound locomotives.

Dimensions of Standard Locomotives on the N. Y. C. [illegible] H. R. R. and Penna. R. R., 1882 and 1893.

C. H. Quereau, *Eng'g News*, March 8, 1894.

	N. Y. C. & H. R. R.				Pennsylvania R. R.			
	Through Passenger.		Through Freight.		Through Passenger.		Through Freight.	
	1882.	1893.	1882.	1893.	1882.	1893.	1882.	[illegible]
Grate surface, sq. ft....	17.87	27.3	17.87	29.8	17.6	33.2	[illegible]	[illegible]
Heating surface, sq. ft..	1353	1821	1353	1763	1057	1583	[illegible]	[illegible]
Boiler, diam., in.........	50	58	50	58	50	57	[illegible]	[illegible]
Driver, diam., in	70	78, 86	64	67	62	78	[illegible]	[illegible]
Steam-pressure, lbs. ..	150	180	150	160	125	175	[illegible]	[illegible]
Cylin., diam. and stroke.	17×24	19×24	17×24	19×26	17×24	[illegible]	[illegible]	[illegible]
Valve-travel, ins	5¼	5½	5¼	5¾	5	[illegible]	[illegible]	[illegible]
Lead at full gear, ins....	1/16	1/16	1/16	1/16	1/16	0	[illegible]	[illegible]
Outside lap..............	⅞	1	⅞	⅞	¾	1	[illegible]	[illegible]
Inside lap or clearance..	0	0	1/16*l*	3/32*l*	0	[illegible]	[illegible]	[illegible]
Steam-ports, length.....	15½	18	15½	18	16	17¼	[illegible]	[illegible]
" " width......	1¼	1¼	1¼	1¼	1¼	1¼	[illegible]	[illegible]
Type of engine...... ...	Am.	Am.	Am.	Mog.	Am.	Am.	[illegible]	[illegible]

Indicated Water Consumption of Single and Comp[illegible] Locomotive Engines at Varying Speeds.

C. H. Quereau, *Eng'g News*, March 8, 1894.

Two-cylinder Compound.			Single-expansion.		
Revolutions.	Speed, miles per hour.	Water per I.H.P. per hour.	Revolutions.	Miles per Hour.	W[illegible]
100 to 150	21 to 31	18.33 lbs.	151	31	[illegible]
150 " 200	31 " 41	18.9 "	219	45	[illegible]
200 " 250	41 " 51	19.7 "	253	52	[illegible]
250 " 275	51 " 56	21.4 "	307	63	[illegible]
			321	66	[illegible]

It appears that the compound engine is the more economical at lo[illegible] the economy decreasing as the speed increases, and that the sing[illegible] increases in economy with increase of speed within ordinary limit[illegible] ing more economical than the compound at speeds of more tha[illegible] per hour.

The C., B. & Q. two-cylinder compound, which was shown [illegible] omical than simple engines of the same class when tested, [illegible] rvice [illegible] been shown to be 15% more economical in t[illegible] gle-expansion engine, and 29% more economical than the 40 simple engines of the same class on the same division.

sts of a Locomotive at High Speed. (*Locomo-* 1893.)—Cards were taken by Mr. Angus Sinclair on the ng the Empire State Express.

RESULTS OF INDICATOR-DIAGRAMS.

Miles per hour.	I.H.P.	Card No.	Revs.	Miles. per hour.	I.H.P.
37.1	648.3	7	304	70.5	977
60.8	728	8	296	68.6	972
44	551	9	300	69.6	1,045
58	891	10	304	70.5	1,059
60	960	11	340	78.9	1,120
69	983	12	310	71.9	1,026

was of the eight-wheel type, built by the Schenectady ks, with 19 × 24 in. cylinders, 78-in. drivers, and a large ox. Details of important dimensions are as follows: of fire-box, 150.8 sq. ft.; of tubes, 1670.7 sq. ft.; of boiler. te area, 27.3 sq. ft. Fire-box: length, 8 ft.; width, 3 ft 4¾ utside diameter, 2 in. Ports: steam, 18 × 1¼ in.; exhaust, e-travel, 5½ in. Outside lap, 1 in.; inside lap, 1/64 in. -axle, 8½ × 10½ in.; truck-axle, 6 × 10 in.

ited of four coaches, weighing, with estimated load, 340,000 tive and tender weighed in working order 200,000 lbs., weight of the train about 270 tons. During the time that st lifting the train into speed diagram No. 1 was taken. It ylinder-pressure of 59 lbs. According to this, the power ils to move the train is 6553 lbs., or 24 lbs. per ton. The in hour. When a speed of nearly 60 miles an hour was rage cylinder-pressure is 40.7 lbs., representing a total 4520 lbs., without making deductions for internal friction. or friction, it leaves 15 lbs. per ton to keep the train going ed. Cards 6, 7, and 8 represent the work of keeping the miles an hour. They were taken three miles apart, when ost uniform. The average cylinder-pressure for the three Deducting 10% again for friction, this leaves 17.6 lbs. per exerted in keeping the train up to a velocity of 70 miles. rip 7 lbs. of water were evaporated per lb. of coal. The e train from New York to Albany was done on a coal con- t 3½ lbs. per H.P. per hour. The highest power recorded 1120 H.P.

-testing Apparatus at the Laboratory of ersity. (W. F. M. Goss, Trans. A. S. M. E., vol. xiv. 826.)— mounted with its drivers upon supporting wheels which afts turning in fixed bearings, thus allowing the engine to hanging its position as a whole. Load is supplied by four ted to the supporting shafts and offering resistance to the pporting wheels. Traction is measured by a dynamometer draw-bar. The boiler is fired in the usual way, and an bove the engine, but not in pipe connection with it, carries e given out at the stack.

thod of Conducting Locomotive-tests is given in a report f the A. S. M. E. in vol. xiv. of the Transactions, page 1312.

el in Locomotives.—In American practice economy arily sacrificed to obtain greater economy due to heavy Barnes, in *Eng. Mag.*, June, 1894, gives a diagram showing fficiency of boilers due to high rates of combustion, from ng figures are taken:

q. ft. of grate per hour......	12	40	80	120	160	200
y of boiler.......	80	75	67	59	51	43

is given as representing stationary-boiler practice, 40 lbs. tive practice, 120 lbs. average American, and 200 lbs. max- *locomotive practice.*

of Compounding.—Report of a Committee of the *Master Mechanics' Association on Compound Locomotives 1890) gives the following summary of the advanta- iding: (a) It has achieved a saving in the fuel by sonable boiler-pressures, with encouraging possibil* of further improvement in pressure and in fuel and water economy has lessened the amount of water (dead weight) to be hauled, so tha tender and its load are materially reduced in weight. (*d*) It has in the possibilities of speed far beyond 60 miles per hour, without straining the motion, frames, axles, or axle-boxes of the engine. (*e*) increased the haulage-power at full speed, or, in other words, has in the continuous H.P. developed, per given weight of engine and boiler some classes has increased the starting-power. (*g*) It has materially l the slide-valve friction per H.P. developed. (*h*) It has equalized or uted the turning force on the crank-pin, over a longer portion of it which, of course, tends to lengthen the repair life of the engine. (*i*) two-cylinder type it has decreased the oil consumption, and has eve so in the Woolf four-cylinder engine. (*j*) Its smoother and steadier d on the fire is favorable to the combustion of all kinds of soft coal; a sparks thrown being smaller and less in number, it lessens the risk t erty from destruction by fire. (*k*) These advantages and econom gained without having to improve the man handling the engine, less left to his discretion (or careless indifference) than in the simple eng Valve-motion, of every locomotive type, can be used in its best work most effective position. (*m*) A wider elasticity in locomotive design mitted; as, if desired, side-rods can be dispensed with, or articulated e of 100 tons weight, with independent trucks, used for sharp curves on tain service, as suggested by Mallet and Brunner.

Of 27 compound locomotives in use on the Phila. and Reading Rail 1892), 12 are in use on heavy mountain grades, and are designed to equivalent of 22 × 24 in. simple consolidations; 10 are in somewhat service and correspond to 20 × 24 in. consolidations; 5 are in fast pas service. The monthly coal record shows:

Class of Engine.	No.	Gain in F Economy
Mountain locomotives.	12	28% to 3[illegible]
Heavy freight service	10	12% to 17[illegible]
Fast passenger	5	9% to 11[illegible]

(Report of Com. A. R. M. M. Assn. 1892.) For a description of the t types of compound locomotive, with discussion of their relative mer paper by A. Von Borries, of Germany, The Development of the Com Locomotive, Trans. A. S. M. E. 1893, vol. xiv., p. 1172.

Counterbalancing Locomotives.—The following rules, a by different locomotive-builders, are quoted in a paper by Prof. (Trans. A. S. M. E., x. 302):

A. "For the main drivers, place opposite the crank-pin a weight e one half the weight of the back end of the connecting-rod plus one ha weight of the front end of the connecting-rod, piston, piston-rod, and head. For balancing the coupled wheels, place a weight opposite the pin equal to one half the parallel rod plus one half of the weights front end of the main-rod, piston, piston-rod, and cross-head. The c of gravity of the above weights must be at the same distance fro axles as the crank-pin."

B. The rule given by D. K. Clark: "Find the separate revolving w of crank-pin boss, coupling-rods, and connecting-rods for each whe the reciprocating weight of the piston and appendages, and one ha connecting-rod, divide the reciprocating weight equally between each and add the part so allotted to the revolving weight on each whe sums thus obtained are the weights to be placed opposite the crank-p at the same distance from the axis. To find the counterweight to b when the distance of its centre of gravity is known, multiply the weight by the length of the crank in inches and divide by the gr tance." This rule differs from the preceding in that the same wei placed in each wheel.

C. "$W = \dfrac{S \times \left(w - \frac{w}{f}\right)}{G}$, in which S = one half the stroke, G = di from centre of wheel to centre of gravity in counterbalance, w = wei crank-pin to be balanced, W = weight in counterbalance, f = coeffi friction so called, = 5 in ordinary practice. The reciprocating w ound by adding together the weights of the piston, piston-rod, c nd one half of the main rod. The revolving weight for the m ound [illegible] together the weights of the crank-pin boss, c od, and one half of each parallel-rod connecting to this d the reciprocating weight divided by the number of lving weight for the remainder of the wheels is found in as for the main wheel, except one half of the main rod is reight of the crank-pin hub and the counterbalance does ight of the spokes, but of the metal inclosing them. This ed for one cylinder and its corresponding wheels."

s nearly as possible the weights of crank-pin, additional boss for the same, add side rod, and main connections, ad, with cross-head on one side: the sum of these multi-nce in inches of the centre of the crank-pin from the centre l divided by the distance from the centre of the wheel to e of gravity of the counterweights, is taken for the total that side of the locomotive which is to be divided among t side."

e wheels of the locomotive with a weight equal to the pin, crank-pin hub, main and parallel rods, brasses, etc., the weight of the reciprocating parts (cross-head, piston ng)."

weights of the revolving parts which are attached to xactness, and divide equally two thirds of the weights of parts between all the wheels. One half of the main rod is rocating, and the other as revolving weight."

on Counterbalancing Locomotives, in *R. R. & Eng. Jour.*, 1890, and a paper by W. F. M. Goss, in Trans. A. S. M. E.,

Safe Load for Steel Tires on Steel Rails.). 786.)—Mr. Chanute's experiments led to the deduction uld be the limit of load for any one driving-wheel. Mr. jects to Mr. Chanute's figure of 12,000 lbs., and says that which has a light load on it is more injurious to the rail as a heavy load. In English practice 8 and 10 tons are Oberlin Smith has used steel castings for cam-rollers 4 in. ce, which stood well under loads of from 10,000 to 20,000 r Smith proposed a formula for the rolls of a pivot-bridge luced to the form: Load = $1760 \times$ face $\times \sqrt{\text{diam.}}$, all in

of some large American locomotives on pages 860 and 861. " the load on each driving-wheel is 17,000 lbs., and on bs.

ge Railways in Manufacturing Works.— inches gauge, several miles in length, is in the works of d Yorkshire Railway. Curves of 13 feet radius are used. sed have the following dimensions (Proc. Inst. M. E., July, rs were 5 in. diameter with 6 in. stroke, and 2 ft. 3¼ in. The wheels were 16¼ in. diameter, the wheel-base ne 7 ft. 4¼ in. long, and the extreme width of the engine of steel, 2 ft. 3 in. outside diameter and 2 ft. long between ining 55 tubes of 1⅜ in. outside diameter; the fire-box, of al, 2 ft. 3 in. long and 17 in. inside diameter. The heating-t. in the fire-box and 36.12 in the tubes, total 46.54 sq. ft.; 8 sq. ft.; capacity of tank, 26½ gallons; working-pressure, tractive power, say, 1412 lbs., or 9.22 lbs. per lb. of effec- sq. in. on the piston. Weight, when empty, 2.80 tons; orking order, 3.19 tons.

of a system of narrow-gauge railways for manufactories, C. W. Hunt Co., New York.

notives.—For dimensions of light ocomotives used for. or much valuable information concerning them, see cata- ter & Co., Pittsburgh.

urning Locomotives. (From Clark's Steam-en- stion of petroleum refuse in locomotives has been success Mr. *Thos. Urquhart*, on the Grazi and Tsaritsin Railway, *Since November, 1884, the whole stock of 143 loco ndence has been fired with petroleum refuse. zle through a tubular opening in the back of steam, with an induced current of air. or "regenerative or accumulative comb fire-box, into which the combined curr* spray against the rugged brickwork slope. In this arrangement the work is maintained at a white heat, and combustion is complete and less. The form, mass, and dimensions of the brickwork are the m portant elements in such a combination.

Compressed air was tried instead of steam for injection, but no appr reduction in consumption of fuel was noticed.

The heating-power of petroleum refuse is given as 19,832 heat equivalent to the evaporation of 20.53 lbs. of water from and at 212° F 17.1 lbs. at 8½ atmospheres, or 125 lbs. per sq. in., effective pressure highest evaporative duty was 14 lbs. of water under 8½ atmospheres of the fuel, or nearly 82% efficiency.

There is no probability of any extensive use of petroleum as fuel fo motives in the United States, on account of the unlimited supply of c the comparatively limited supply of petroleum.

Fireless Locomotive.—The principle of the Francq locom that it depends for the supply of steam on its spontaneous generatio a body of heated water in a reservoir. As steam is generated and off the pressure falls; but by providing a sufficiently large volume o heated to a high temperature, at a pressure correspondingly high, a of surplus pressure may be secured, and means may thus be provi supplying the required quantity of steam for the trip.

The fireless locomotive designed for the service of the Metropolita way of Paris has a cylindrical reservoir having segmental ends, ab 7 in. in diameter, 26¼ ft. in length, with a capacity of about 620 cub Four fifths of the capacity is occupied by water, which is heated by of a powerful jet of steam supplied from stationary boilers. The w heated until equilibrium is established between the boilers and the voir. The temperature is raised to about 390° F., corresponding to per sq. in. The steam from the reservoir is passed through a re valve, by which the steam is reduced to the required pressure. It passed through a tubular superheater situated within the receiver upper part, and thence through the ordinary regulator to the cy The exhaust-steam is expanded to a low pressure, in order to obvia of escape. In certain cases the exhaust-steam is condensed in vessels, which are only in part filled with water. In the upper free pipe is placed, into which the steam is exhausted. Within this pipe pipe is fixed, perforated, from which cold water is projected into t rounding steam, so as to effect the condensation as completely as The heated water falls on an inclined plane, and flows off without with the cold water. The condensing water is circulated by mea centrifugal pump driven by a small three-cylinder engine.

In working off the steam from a pressure of 225 lbs. to 67 lbs., 53 feet of water at 390° F., is sufficient for the traction of the trains, for the circulating-pump for the condensers, for the brakes, and for lighting of the train. At the stations the locomotive takes from 220 lbs. of steam—nearly the same as the weight of steam consumed du run between two consecutive charging stations. There is 210 cubic condensing water. Taking the initial temperature at 60° F., the te ture rises to about 180° F. after the longest runs underground.

The locomotive has ten wheels, on a base 24 ft. long, of which coupled, 4½ ft. in diameter. The extreme wheels are on radial axle cylinders are 23½ in. in diameter, with a stroke of 23½ in.

The engine weighs, in working order, 53 tons, of which 36 tons are coupled wheels. The speed varies from 15 miles to 25 miles per hou trains weigh about 140 tons.

Compressed-air Locomotives.—For an account of the M system of compressed-air locomotives see page 509, *ante*.

SHAFTING.

also TORSIONAL STRENGTH; also SHAFTS OF STEAM-ENGINES.)

neters of shafts to resist torsional strains only, Molesworth gives
$\overline{l}$
:, in which d = diameter in inches, P = twisting force in pounds

the end of a lever-arm whose length is l in inches, K = a coeffi-
se values are, for cast iron 1500, wrought iron 1700, cast steel 3200,
e 460, brass 425, copper 380, tin 220, lead 170. The value given for
probably applies only to high-carbon steel.
1 gives:

d shafts well against:

$$\text{H.P.} = \frac{d^3R}{125};\ d = \sqrt[3]{\frac{125\ \text{H.P.}}{R}},\ \text{for iron;}$$

$$\text{H.P.} = \frac{d^3R}{75};\ d = \sqrt[3]{\frac{75\ \text{H.P.}}{R}},\ \text{for cold-rolled iron.}$$

ne shafting, ft. apart:

$$\text{H.P.} = \frac{d^3R}{90};\ d = \sqrt[3]{\frac{90\ \text{H.P.}}{R}},\ \text{for iron;}$$

$$\text{H.P.} = \frac{d^3R}{55};\ d = \sqrt[3]{\frac{55\ \text{H.P.}}{R}},\ \text{for cold-rolled iron.}$$

smission sim- lleys:

$$\text{H.P.} = \frac{d^3R}{62.5};\ d = \sqrt[3]{\frac{62.5\ \text{H.P.}}{R}},\ \text{for iron;}$$

$$\text{H.P.} = \frac{d^3R}{35};\ d = \sqrt[3]{\frac{35\ \text{H.P.}}{R}},\ \text{for cold-rolled iron.}$$

rse-power transmitted, d = diameter of shaft in inches, R = rev-
r minute.

ncis gives for turned-iron shafting $d = \sqrt[3]{\frac{100\ \text{H.P.}}{R}}$.

d Laughlins give the same formulæ as Prof. Thurston, with the
xceptions: For line shafting, hangers 8 ft. apart:

$$\text{cold-rolled iron, H.P.} = \frac{d^3R}{50},\ d = \sqrt[3]{\frac{50\ \text{H.P.}}{R}}.$$

ly transmitting power and short counters:

$$\text{turned iron, H.P.} = \frac{d^3R}{50},\ d = \sqrt[3]{\frac{50\ \text{H.P.}}{R}};$$

$$\text{cold-rolled iron, H.P.} = \frac{d^3R}{30},\ d = \sqrt[3]{\frac{30\ \text{H.P.}}{R}}.$$

o give the following notes: Receiving and transmitting pulleys
ays be placed as close to bearings as possible; and it is g
e short "headers" between the main tie-beams of a
he main receivers, carried by the head shafts, with
side as is contemplated in the formulæ. But if it
for the shaft to span the full width of the "bay"

termediate bearings, or for the pulley to be placed away from towards or at the middle of the bay, the size of the shaft mu increased to secure the *stiffness* necessary to support the load due deflection. Shafts may not deflect more than 1/80 of an foot of clear length with safety.

To find the diameter of shaft necessary to carry safely the n the centre of a bay: Multiply the fourth power of the diamete above formulæ by the length of the "bay," and divide this pr distance from centre to centre of the bearings when the shaft as required by the formula. The fourth root of this quotie diameter required.

The following table, computed by this rule, is practically cor

Diameter of Shaft given by the Formulæ for Head Shafts.	Diameter of Shaft necessary to carry the Load at t a Bay, which is from Centre to Centre of Bea						
	2½ ft.	3 ft.	3½ ft.	4 ft.	5 ft.	6 ft.	8
in.	in.	in.	in.	in.	in.	in.	
2	2⅛	2¼	2⅜	2½	2⅝	2¾	
2½	2½	2⅝	2¾	2⅞	3	3⅛	
3	3	3⅛	3¼	3⅜	3½	3¾	
3½	....	3½	3⅝	3¾	4	4¼	
4	....	4	4⅛	4¼	4½	4¾	
4½	....	....	4½	4⅝	4⅞	5⅛	
5	....	....	5	5⅛	5⅜	5⅝	
5½	....	....	....	5½	5¾	6	
6	....	....	....	6	6⅜	6⅝	

As the strain upon a shaft from a load upon it is propo product of the parts of the shaft multiplied into each oth should the load be applied near one end of the span or bay ins centre, multiply the fourth power of the diameter of the shaf carry the load at the centre of the span or bay by the produ parts of the shaft when the load is near one end, and divide th the product of the two parts of the shaft when the load is c centre. The fourth root of this quotient will be the diameter r

The shaft in a line which carries a receiving-pulley, or wh transmitting-pulley to drive another line, should always be head-shaft, and should be of the size given by the rules for sh main pulleys or gears.

Deflection of Shafting. (Pencoyd Iron Works.)—As of steel and iron is practically alike under similar conditions o and loads, and as shafting is usually determined by its transv rather than its ultimate strength, nearly the same dimensio used for steel as for iron.

For continuous line-shafting it is considered good practic deflection to a maximum of 1/100 of an inch per foot of length. of bare shafting in pounds $= 2.6d^2L = W$, or when as fully pulleys as is customary in practice, and allowing 40 lbs. per i for the vertical pull of the belts, experience shows the load in about $13d^2L = W$. Taking the modulus of transverse elasticit lbs., we derive from authoritative formulæ the following:

$$L = \sqrt[3]{873d^2},\ d = \sqrt{\frac{L^3}{873}},\ \text{for bare shafting;}$$

$$L = \sqrt[3]{175d^2},\ d = \sqrt{\frac{L^3}{175}},\ \text{for shafting carrying pulleys}$$

L being the maximum distance in feet between bearings shafting subjected to bending stress alone, d = diam. in inch

The [illegible] stress is inversely proportional to the velo [illegible] stress will not be reduced in the same r [illegible] write a formula covering the whole

simple for practical application, but the following rules are correct the range of velocities usual in practice.
continuous shafting so proportioned as to deflect not more than 1/100 inch per foot of length, allowance being made for the weakening of key-seats,

$$l = \sqrt[3]{\frac{50\,\text{H.P.}}{R}},\quad L = \sqrt[3]{720d^2},\ \text{for bare shafts;}$$

$$l = \sqrt[3]{\frac{70\,\text{H.P.}}{R}},\quad L = \sqrt[3]{140d^2},\ \text{for shafts carrying pulleys, etc.}$$

am. in inches, L = length in feet, R = revs. per min.
following table (by J. B. Francis) gives the greatest admissible dis- between the bearings of continuous shafts subject to no transverse except from their own weight, as would be the case were the power off from the shaft equal on all sides, and at an equal distance from nger-bearings.

of Shaft, nches.	Distance between Bearings, in ft. Wrought-iron Shafts.	Steel Shafts.	Diam. of Shaft, in inches.	Distance between Bearings, in ft. Wrought-iron Shafts.	Steel Shafts.
[illegible]	15.46	15.89	6	22.30	22.92
[illegible]	17.70	18.19	7	23.48	24.13
[illegible]	19.48	20.02	8	24.55	25.23
5	20.99	21.57	9	25.58	26.24

se conditions, however, do not usually obtain in the transmission of by belts and pulleys, and the varying circumstances of each case it impracticable to give any rule which would be of value for univer- plication.
example, the theoretical requirements would demand that the bear- be nearer together on those sections of shafting where most power ivered from the shaft, while considerations as to the location and d contiguity of the driven machines may render it impracticable to te the driving-pulleys by the intervention of a hanger at the theo- lly required location. (Joshua Rose.)

e-power Transmitted by Turned Iron Shafting at Different Speeds.

IME MOVER OR HEAD SHAFT CARRYING MAIN DRIVING-PULLEY OR GEAR, WELL SUPPORTED BY BEARINGS. Formula: H.P. = $d^3R \div 125$.

Number of Revolutions per Minute.										
60	80	100	125	150	175	200	225	250	275	300
H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.
2.6	3.4	4.3	5.4	6.4	7.5	8.6	9.7	10.7	11.8	12.9
3.8	5.1	6.4	8	9.6	11.2	12.8	14.4	16	17.6	19.2
5.4	7.3	8.1	10	12	14	16	18	20	22	24
7.5	10	12.5	15	18	22	25	28	31	34	37
10	13	16	20	24	28	32	36	40	44	48
13	17	20	25	30	35	40	45	50	55	60
16	22	27	34	40	47	54	61	67	74	81
20	27	34	42	51	59	68	76	85	93	102
25	33	42	52	63	73	84	94	105	115	126
30	41	51	64	76	89	102	115	127	140	153
43	58	72	90	108	126	144	162	180	198	216
[illegible]	80	100	125	150	175	200	225	250	275	300
[illegible]	106	133	166	199	233	266	299	333	366	[illegible]

AS SECOND MOVERS OR LINE-SHAFTING, BEARINGS 8 FT. APART.
Formula: H.P. $= d^3R \div 90$.

Diam. of Shaft.	Number of Revolutions per Minute.										
	100	125	150	175	200	225	250	275	300	325	[illegible]
Ins.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.
1¾	6	7.4	8.9	10.4	11.9	13.4	14.9	16.4	17.9	19.4	[illegible]
1⅞	7.3	9.1	10.9	12.7	14.5	16.3	18.2	20	21.8	23.6	[illegible]
2	8.9	11.1	13.3	15.5	17.7	20	22.2	24.4	26.6	28.8	[illegible]
2⅛	10.6	13.2	15.9	18.5	21.2	23.8	26.5	29.1	31.8	34.4	[illegible]
2¼	12.6	15.8	19	22	25	28	31	35	38	41	[illegible]
2⅜	15	18	22	26	29	33	37	41	44	48	[illegible]
2½	17	21	26	30	34	39	43	47	52	56	[illegible]
2¾	23	29	34	40	46	52	58	64	69	75	[illegible]
3	30	37	45	52	60	67	75	82	90	97	[illegible]
3¼	38	47	57	66	76	85	95	104	114	123	[illegible]
3½	47	59	71	83	95	107	119	131	143	155	[illegible]
3¾	58	73	88	102	117	132	146	162	176	190	[illegible]
4	71	89	107	125	142	160	178	196	213	231	[illegible]

FOR SIMPLY TRANSMITTING POWER.
Formula: H.P. $= d^3R \div 50$.

Diam. of Shaft.	Number of Revolutions per Minute.										
	100	125	150	175	200	233	267	300	333	367	[illegible]
Ins.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.
1½	6.7	8.4	10.1	11.8	13.5	15.7	17.9	20.3	22.5	24.8	[illegible]
1⅝	8.6	10.7	12.8	15	17.1	20	22.8	25.8	28.6	31.5	[illegible]
1¾	10.7	13.4	16	18.7	21.5	25	28	32	36	39	[illegible]
1⅞	13.2	16.5	19.7	23	26.4	31	35	39	44	48	[illegible]
2	16	20	24	28	32	37	42	48	53	58	[illegible]
2⅛	19	24	29	33	38	44	51	57	63	70	[illegible]
2¼	22	28	34	39	45	52	60	68	75	83	[illegible]
2⅜	27	33	40	47	53	62	70	79	88	96	[illegible]
2½	31	39	47	54	62	73	83	93	104	114	[illegible]
2¾	41	52	62	73	83	97	111	125	139	153	[illegible]
3	54	67	81	94	108	126	144	162	180	198	[illegible]
3¼	68	86	103	120	137	160	182	205	228	250	[illegible]
3½	85	107	128	150	171	200	228	257	285	313	[illegible]

Horse-power Transmitted by Cold-rolled Iron Shafting at Different Speeds.

AS PRIME MOVER OR HEAD SHAFT CARRYING MAIN DRIVING-PULLEY OR GEAR, WELL SUPPORTED BY BEARINGS. Formula: H.P. $= d^3R \div 75$.

Diam. of Shaft.	Number of Revolutions per Minute.										
	60	80	100	125	150	175	200	225	250	275	[illegible]
Ins.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.
1½	2.7	3.6	4.5	5.6	6.7	7.9	9.0	10	11	12	[illegible]
1¾	4.3	5.6	7.1	8.9	10.6	12.4	14.2	16	18	19	[illegible]
2	6.4	8.5	10.7	13	16	19	21	24	26	29	[illegible]
2¼	9	12	15	19	23	26	30	34	38	42	[illegible]
2½	12	17	21	26	31	36	41	47	52	57	[illegible]
2¾	16	22	27	35	41	48	55	62	70	76	[illegible]
3	21	29	36	45	54	63	72	81	90	99	[illegible]
3¼	27	36	45	57	68	80	91	103	114	[illegible]	[illegible]
3½	34	45	57	71	86	100	114	129	143	[illegible]	[illegible]
3¾	42	56	70	87	105	123	140	158	174	[illegible]	[illegible]
4	51	69	85	106	128	149	170	192	213	[illegible]	[illegible]
4½	73	[illegible]	121	151	182	212	243	273	303	[illegible]	[illegible]

SECOND MOVERS OR LINE-SHAFTING, BEARINGS 8 FT. APART.

Formula: H.P. = $d^3R \div 50$.

Number of Revolutions per Minute.										
00	125	150	175	200	225	250	275	300	325	350
P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.
.7	8.4	10.1	11.8	13.5	15.2	16.8	18.5	20.2	21.9	23.6
.6	10.7	12.8	15	17.1	19.3	21.5	23.6	25.7	28.9	31
.7	13.4	16	18.7	21.5	24.2	26.8	29.5	32.1	34.8	39
.2	16.5	19.7	23	26.4	29.6	32.9	36.2	39.5	42.8	46
	20	24	28	32	36	40	44	48	52	56
	24	29	33	38	43	48	52	57	62	67
	28	34	39	45	50	56	61	68	74	80
	33	40	47	53	60	67	73	80	86	94
	39	47	54	62	69	78	86	93	101	109
	52	62	73	83	93	104	114	125	135	145
	67	81	94	108	121	134	148	162	175	189
	86	103	120	137	154	172	188	205	222	240
	107	128	150	171	192	214	235	257	278	300

FOR SIMPLY TRANSMITTING POWER AND SHORT COUNTERS.

Formula: H.P. = $d^3R \div 30$.

Number of Revolutions per Minute.										
00	125	150	175	200	233	267	300	333	367	400
P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.
.5	8.1	9.7	11.3	13	15.2	17.4	19.5	21.7	23.9	26
5	10.7	12.8	15	17	19.8	22.7	25.5	28.4	31	34
2	14	16.8	19.6	22.5	26	30	33	37	41	45
.2	17.7	21.2	24.8	28.4	33	38	42	47	52	57
	22	27	31	35	41	47	53	59	65	71
	27	33	38	44	51	58	65	72	79	87
	33	40	46	53	62	71	80	88	97	106
	40	47	55	63	73	84	95	105	116	127
	47	57	66	76	89	101	114	127	139	152
	55	66	77	88	103	118	133	148	163	178
	65	78	91	104	121	138	155	172	190	207
	84	99	113	138	161	184	207	231	254	277
	112	135	157	180	210	240	270	300	330	360

EED OF SHAFTING.—Machine shops .... 120 to 180
Wood-working................ 250 to 300
Cotton and woollen mills...... 300 to 400

re in some factories lines 1000 ft. long, the power being applied at e.

w Shafts.—Let d be the diameter of a solid shaft, and $d_1 d_2$ the and internal diameters of a hollow shaft of the same material, shafts will be of equal torsional strength when $d^3 = \frac{d_1^4 - d_2^4}{d_1}$. hollow shaft with internal diameter of 4 inches will weigh 16% less id 10-inch shaft, but its strength will be only 2.56% less. If the hole ased to 5 inches diameter the weight would be 25% less than that d shaft, and the strength 4.25% less.

for Laying Out Shafting.—The table on the opposite page *Stevens Indicator*, April, 1892) is used by Wm. Sellers & ... *he laying out of shafting.*

-cuts at the head of this table show the position of ... *of couplings, either for the case of extension in bo*... *al head-shaft or extension in one direction from the*

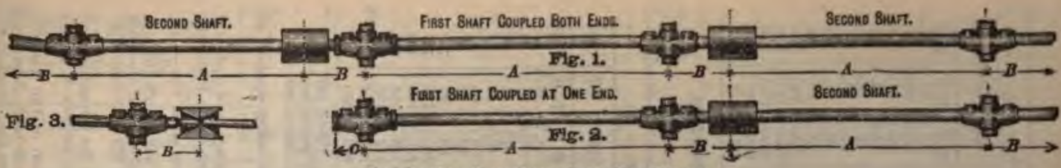


Table for Laying Out Shafting.

Distance from Centre of Bearing to End of Shaft for Coupling. See B, Figs. 1, 2, and 3.

Nominal Size of 1st Shaft, ins. \ Nominal Size of 2d Shaft.	1½″	1¾″	2″	2¼″	2½″	2¾″	3″	3¼″	3½″	4″	4½″	5″	5½″	6″	6½″	7″	7½″	8″
1½	8½																	
1¾	9	9½																
2	9½	10½	11															
2¼		11	11½	12														
2½		11½	12	12½	13													
2¾		12	12½	13	13½	14												
3			13	13½	14	14½	15											
3¼			13½	14	14½	15	15½	15½										
3½			14	14½	15	15½	16	16	16½									
4			15	15½	16	16½	17	17	17½	18								
4½				16½	17	17½	18	18	18½	19½	20							
5					18½	18½	19	19½	20	20½	21½	22						
5½						20	20½	21	21	22	22½	23½	24					

USE OF TABLE.—Look for size of first shaft in left-hand column, under the head of Size of first shaft, and in the top line of table, marked Size of second shaft, find the size of the shaft to be coupled to it. The intersection gives the length B; this added to the length A, or distance from centre to centre of bearing, and in cases similar to Fig. 2, to the length C, gives the length of the first shaft, thus: as in Fig. 1, $B + A + B$ = length; Fig. 2, $C + A + B$ = length.

Make bearings at equal distances from each other, when practicable, and always put two bearings on the first, which is the collared shaft. See Figs. 1 and 2.

Length of Collared End for Fast Coll., ins.	Length of Bearing, or Box, ins.	Double Convise Coupling: Length, inches.	Double Convise Coupling: Diameter, inches.
3½	6	5½	3¾
4	7	6½	4⅝
4½	8	7⅞	5⅜
5	9	8¼	5⅞
5⅝	10	9½	6 9/16
6¼	11	10	7
6⅝	12	11	7¾
7¼	13	12	8¼
7⅝	14	13	9 1/16
8 15/16	16	14¼	10
10	18	16	11⅝
11⅛	20	18	12 9/16
12 3/16	22	19	13½
	24	21	14½

PULLEYS.

rtions of Pulleys. (See also Fly-wheels, pages 820 to 823.)—umber of arms, D = diameter of pulley, S = thickness of belt, t = of rim at edge, T = thickness in middle, B = width of rim, β = belt, h = breadth of arm at hub, h_1 = breadth of arm at rim, e = of arm at hub e_1 = thickness of arm at rim, c = amount of crown-nsions in inches.

	Unwin.	Reuleaux.
of rim....................	$9/8(\beta + 0.4)$	$9/8\beta$ to $5/4\beta$
ness at edge of rim.......	$0.7S + .005D$	(thick. of rim.) $1/5h$ to $\frac{1}{4}h$
" middle of rim.....	$2t + c$	....
dth of arm at hub.....	For single belts = $.6337\sqrt[3]{\frac{BD}{n}}$; For double belts = $.798\sqrt[3]{\frac{BD}{n}}$	$\frac{\frac{1}{4}''}{4} + \frac{B}{4} + \frac{D}{20n}$
" " " rim.........	$\frac{2}{3}h$	$0.8h$
ness of arm at hub.......	$0.4h$	$0.5h$
" " " rim.......	$0.4h_1$	$0.5h_1$
ber of arms, for a le set, ...	$3 + \frac{BD}{150}$	$\frac{1}{2}\left(5 \times \frac{D}{2B}\right)$
h of hub	not less than $2.5S$, is often $\frac{2}{3}B$.	B for sin.-arm pulleys. $2B$ " double-arm "
ness of metal in hub.......	....	h to $\frac{3}{4}h$
ning of pulley.............	$1/24B$	....

nber of arms is really arbitrary, and may be altered if necessary.

with two or three sets of arms may be considered as two or three pulleys combined in one, except that the proportions of the arms 0.8 or 0.7 time that of single-arm pulleys. (Reuleaux.)

.E.—Dimensions of a pulley 60″ diam., 16″ face, for double belt ½″

on by....	n	h	h_1	e	e_1	t	T	L	M	c
n.........	9	3.79	2.53	1.52	1.01	.65	1.97	10.7	3.8	.67
aux.. ...	4	5.0	4.0	2.5	2.0	1.25		16	5	

owing proportions are given in an article in the *Amer. Machinist*, not stated:

$5D + .5$ in., $h_1 = .04D + 3125$ in., $e = .025D + .2$ in., $e_1 = .016D +$

ive for the above example: $h = 4.25$ in., $h_1 = 2.71$ in., $e = 1.7$ in., in. The section of the arms in all cases is taken as elliptical.

lowing solution for breadth of arm is proposed by the author: belt pull of 45 lbs. per inch of width of a single belt, that the ain is taken in equal proportions on one half of the arms, and that a beam loaded at one end and fixed at the other. We have the or a beam of elliptical section $fP = .0982 \frac{Rbd^2}{l}$, in which P = the the modulus of rupture of the cast iron, b = breadth, d = depth, ngth of the beam, and f = factor of safety. Assume a modulus e of 36,000 lbs., a factor of safety of 10, and an additional allow-safety in taking l = ½ the diameter of the pulley instead of ½D dius of the hub.

$= h$, the breadth of the arm at the hub, and $b = e = 0.4h$, the

We then have $fP = 10 \times \frac{45B}{n+2} = 900\frac{B}{n} = \frac{3535 \times 0.4h^3}{\frac{1}{2}D}$, whence

$\frac{BD}{...} = .633\sqrt[3]{\frac{BD}{n}}$, which is practically the same ...

nwin from a different set of assumptions.

Convexity of Pulleys.—Authorities differ. Morin gives a rise eq
to 1/10 of the face; Molesworth, 1/24; others from 1/8 to 1/96. Scott
Smith says the crown should not be over 1/8 inch for a 24-inch face. Pul
for shifting belts should be "straight," that is, without crowning.

CONE OR STEP PULLEYS.

To find the diameters for the several steps of a pair of cone-pulleys:
1. **Crossed Belts.**—Let D and d be the diameters of two pulleys
nected by a crossed belt, L = the distance between their centres, an
the angle either half of the belt makes with a line joining the centres
pulleys: then total length of belt $= (D+d)\frac{\pi}{2} + (D+d)\frac{\pi\beta}{180} + 2L$ c

β = angle whose sine is $\frac{D+d}{2L}$. $\operatorname{Cos}\beta = \sqrt{L^2 - \left(\frac{D+d}{2}\right)}$. The len
the belt is constant when $D+d$ is constant; that is, in a pair o
pulleys the belt tension will be uniform when the sum of the diame
each opposite pair of steps is constant. Crossed belts are seldom u
cone-pulleys, on account of the friction between the rubbing parts
belt.

To design a pair of tapering speed-cones, so that the belt
equally tight in all positions: When the belt is crossed, use a pair o
and similar cones tapering opposite ways.

2. **Open Belts.**—When the belt is uncrossed, use a pair of equ
similar conoids tapering opposite ways, and bulging in the middle,
ing to the following formula: Let L denote the distance between th
of the conoids; R the radius of the larger end of each; r the radius
smaller end; then the radius in the middle, r_0, is found as follows:

$$r_0 = \frac{R+r}{2} + \frac{(R-r)^2}{6.28L}. \quad \text{(Rankine.)}$$

If D_0 = the diameter of equal steps of a pair of cone-pulleys, D a
the diamêters of unequal opposite steps, and L = distance betw
axes, $D_0 = \frac{D+d}{2} + \frac{(D-d)^2}{12.566L}$.

If a series of differences of radii of the steps, $R - r$, be assume
for each pair of steps $\frac{R+r}{2} = r_0 - \frac{(R-r)^2}{6.28L}$, and the radii of each
computed from their half sum and half difference, as follows:

$$R = \frac{R+r}{2} + \frac{R-r}{2}; \quad r = \frac{R+r}{2} - \frac{R-r}{2}.$$

A. J. Frith (Trans. A. S. M. E., x. 298) shows the following applica
Rankine's method: If we had a set of cones to design, the extreme
ters of which, including thickness of belt, were 40″ and 10″, and th
desired 4, 3, 2, and 1, we would make a table as follows, L being 100″

Trial Sum of $D+d$.	Ratio.	Trial Diameters.		Values of $\frac{(D-d)^2}{12.56L}$	Amount to be Added.	Corrected
		D	d			D
50	4	40	10	.7165	.0000	40
50	3	37.5	12.5	.4975	.2190	37.7190
50	2	33.333	16.666	.2212	.4953	33.8282
50	1	25	25	.0000	.7165	25.7165

The above formulæ are approximate, and they do not give satisf
results when the difference of diameters of opposite steps is large a
the axes of the pulleys are near together, giving a large belt-angle
following more accurate solution of the problem is given by C. A
(Trans. A. S. M. E., x. 269) (Fig 152):

La centre distance C or EF, and draw the circles D, and
r of pulleys, which are always previously dete
s. Draw HI tangent to the circles D, and d,
E and F, erect the perpendicular BG, makin

BG = .314C. With G as a centre, draw a circle tangent to HI. Ge[nerally] this circle will be outside of the belt-line, as in the cut, but when C i[s small] and the first pulleys D_1 and d_1 are large, it will fall on the inside of th[e belt-]line. The belt-line of any other pair of pulleys must be tangent to t[he cir]cle G; hence any line, as JK or LM, drawn tangent to the circle G, w[ill give]

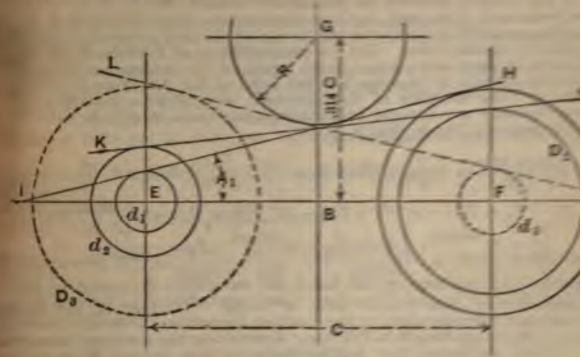


FIG. 182.

the diameters D_2, d_2 or D_3, d_3 of the pulleys drawn tangent to the[m] from the centres E and F.

The above method is to be used when the belt-angle A does not [exceed] 18°. When it is between 18° and 30° a slight modification is made. [In this] case, in addition to the point G, locate another point *u* on the line B[G] above B. Draw a tangent line to the circle G, making an angle of 18[° with the] line of centres EF, and from the point *u* draw an arc tangent to th[is tan]gent line. All belt-lines with angles greater than 18° are tangent to t[his arc.] The following is the summary of Mr. Smith's mathematical method:

A = angle in degrees between the centre line and the belt of any [pair of] pulleys;
a = .314 for belt-angles less than 18°, and .298 for angles betw[een 18°] and 30°;
$B°$ = an angle depending on the velocity ratio;
C = the centre distance of the two pulleys;
D, d = diameters of the larger and smaller of the pair of pulleys;
$E°$ = an angle depending on $B°$;
L = the length of the belt when drawn tight around the pulleys;
r = $D \div d$, or the velocity ratio (larger divided by smaller).

$$(1)\ \sin A = \frac{D-d}{2C}; \quad (2)\ \tan B° = \frac{2a(r-1)}{r+1};$$

$$(3)\ \sin E° = \sin B°\left(\cos A - \frac{D+d}{4aC}\right);$$

(4) $A = B° - E°$ when $\sin E°$ is positive; $= B° + E°$ when $\sin E°$ is ne[gative];

(5) $d = \dfrac{2C \sin A}{r-1}$; $= .3183(L - 2C)$ when $A = 0$ and $r = 1$;

(6) $D = rd$;

(7) $L = 2C \cos A + .01745d[180 + (r-1)(90 + A)]$.

Equation (1) is used *only once* for any pair of cones to obtai[n ...] cos A, *by the aid of tables of sines* and cosines, for use in eq[uations ...]

BELTING.

Theory of Belts and Bands.—A pulley is driven by a belt by means of the friction between the surfaces in contact. Let T_1 be the tension on the driving side of the belt, T_2 the tension on the loose side; then $S = T_1 - T_2$, is the total friction between the band and the pulley, which is equal to the tractive or driving force. Let f = the coefficient of friction, θ the ratio of the length of the arc of contact to the length of the radius, a = the angle of the arc of contact in degrees, e = the base of the Naperian logarithms = 2.71828, m = the modulus of the common logarithms = 0.434295. The following formulæ are derived by calculus (Rankine's Mach'y & Millwork, p. 351; Carpenter's Exper. Eng'g, p. 173):

$$\frac{T_1}{T_2} = e^{f\theta};\quad T_2 = \frac{T_1}{e^{f\theta}};\quad T_1 - T_2 = T_1 - \frac{T_1}{e^{f\theta}} = T_1(1 - e^{-f\theta}).$$

$$T_1 - T_2 = T_1(1 - e^{-f\theta}) = T_1(1 - 10^{-f\theta m}) = T_1(1 - 10^{-.00758fa});$$

$$\frac{T_1}{T_2} = 10^{.00758fa};\quad T_1 = T_2 \times 10^{.00758fa};\quad T_2 = \frac{T_1}{10^{.00758fa}}.$$

If the arc of contact between the band and the pulley expressed in turns and fractions of a turn $= n$, $\theta = 2\pi n$; $e^{f\theta} = 10^{2.7288fn}$; that is, $e^{f\theta}$ is the natural number corresponding to the common logarithm $2.7288fn$.

The value of the coefficient of friction f depends on the state and material of the rubbing surfaces. For leather belts on iron pulleys, Morin found $f = .56$ when dry, .36 when wet, .23 when greasy, and .15 when oily. In calculating the proper mean tension for a belt, the smallest value, $f = .15$, is to be taken if there is a probability of the belt becoming wet with oil. The experiments of Henry R. Towne and Robert Briggs, however (Jour. Frank. Inst., 1868), show that such a state of lubrication is not of ordinary occurrence; and that in designing machinery we may in most cases safely take $f = 0.42$. Reuleaux takes $f = 0.25$. The following table shows the values of the coefficient $2.7288f$, by which n is multiplied in the last equation, corresponding to different values of f; also the corresponding values of various ratios among the forces, when the arc of contact is half a circumference:

$f =$	0.15	0.25	0.42	0.56
$2.7288f =$	0.41	0.68	1.15	1.53

Let $\theta = \pi$ and $n = \frac{1}{2}$, then

$T_1 \div T_2 =$	1.603	2.188	3.758	5.821
$T_1 \div S =$	2.66	1.84	1.36	1.21
$T_1 + T_2 \div 2S =$	2.16	1.34	0.86	0.71

In ordinary practice it is usual to assume $T_2 = S$; $T_1 = 2S$; $T_1 + T_2 \div 2S = 1.5$. This corresponds to $f = 0.22$ nearly.

For a wire rope on cast iron f may be taken as 0.15 nearly; and if the groove of the pulley is bottomed with gutta percha, 0.25. (Rankine.)

Centrifugal Tension of Belts.—When a belt or band runs at a high velocity, centrifugal force produces a tension in addition to that existing when the belt is at rest or moving at a low velocity. This centrifugal tension diminishes the effective driving force.

Rankine says: If an endless band, of any figure whatsoever, runs at a given speed, the centrifugal force produces a uniform tension at each cross-section of the band, equal to the weight of a piece of the band whose length is twice the height from which a heavy body must fall, in order to acquire the velocity of the band. (See Cooper on Belting, p. 101.)

If T_c = centrifugal tension;
V = velocity in feet per second;
g = acceleration due to gravity = 32.2;
W = weight of a piece of the belt 1 ft. long and 1 sq. in. sectional area.

Leather weighing 56 lbs. per cubic foot gives $W = 56 \div 144 = .388$.

$$T_c = \frac{WV^2}{g} = \frac{.388V^2}{32.2} = .012V^2.$$

ting Practice. Handy Formulæ for Belting. — Since practical application of the above formulæ the value of the coefficient ion must be assumed, its actual value varying within wide limits (15%), and since the values of T_1 and T_2 also are fixed arbitrarily, it is cus- in practice to substitute for these theoretical formulæ more simple cal formulæ and rules, some of which are given below.

d = diam. of pulley in inches; πd = circumference;
V = velocity of belt in ft. per second; v = vel. in ft. per minute;
a = angle of the arc of contact;
L = length of arc of contact in feet $= \pi d a \div (12 \times 360)$;
F = tractive force per square inch of sectional area of belt;
w = width in inches; t = thickness;
S = tractive force per inch of width $= F \div t$;
. = revs. per minute; rps. = revs. per second = rpm. ÷ 60.

$$V = \frac{\pi d}{12} \times \text{rps.} = \frac{\pi d}{12} \times \frac{\text{rpm.}}{60} = .004363d \times \text{rpm.} = \frac{d \times \text{rpm.}}{229.2};$$

$$v = \frac{\pi d}{12} \times \text{rpm.}; = .2618d \times \text{rpm.}$$

$$\text{power, H.P.} = \frac{Svw}{33000} = \frac{SVw}{550} = \frac{Swd \times \text{rpm.}}{126050} = .000007933Swd \times \text{rpm.}$$

= working tension per square inch = 275 lbs., and $t = 7/32$ inch, $S =$ nearly, then

$$\text{H.P.} = \frac{vw}{550} = .109Vw = .000476wd \times \text{rpm.} = \frac{wd \times \text{rpm.}}{2101} \quad (1)$$

= 180 lbs. per square inch, and $t = 1/6$ inch, $S = 30$ lbs., then

$$\text{H.P.} = \frac{vw}{1100} = .055Vw = .000238wd \times \text{rpm.} = \frac{wd \times \text{rpm.}}{4202} \quad (2)$$

e working strain is 60 lbs. per inch of width, a belt 1 inch wide travel- 0 ft. per minute will transmit 1 horse-power. If the working strain is per inch of width, a belt 1 inch wide, travelling 1100 ft. per minute, ansmit 1 horse-power. Numerous rules are given by different writers ing which vary between these extremes. A rule commonly used is: wide travelling 1000 ft. per min. = I.H.P.

$$\text{H.P.} = \frac{vw}{1000} = .06Vw = .000262wd \times \text{rpm.} = \frac{wd \times \text{rpm.}}{3820} \quad (3)$$

orresponds to a working strain of 33 lbs. per inch of width.
writers give as safe practice for single belts in good condition a g tension of 45 lbs. per inch of width. This gives

$$\text{H.P.} = \frac{wv}{733} = .0818Vw = .000357wd \times \text{rpm.} = \frac{wd \times \text{rpm.}}{2800} \quad (4)$$

double belts of average thickness, some writers say that the trans- efficiency is to that of single belts as 10 to 7, which would give

$$\text{double belts} = \frac{wv}{513} = .1169Vw = .00051wd \times \text{rpm.} = \frac{wd \times \text{rpm.}}{1960} \quad (5)$$

authorities, however, make the transmitting-power of double belts hat of single belts, on the assumption that the thickness of a double- twice that of a single belt.

s for horse-power of belts are sometimes based on the number of feet of surface of the belt which pass over the pulley in a minute. per min. $= wv \div 12$. The above formulæ translated into this form

For $S = 60$ lbs. per inch wide; H.P. = 46 sq. ft. per minute.
" $S = 30$ " " " H.P. = 92 " "
" $S = 33$ " " " H.P. = 83 " "
" $S = 45$ " " " H.P. = 61 " "
" $S = 64.8$ " " " H.P. = 43 " " (double be

The above formulæ are all based on the supposition that the arc of con- tact is 180° For other arcs, the transmitting power is approximately pro- portional to the ratio of the degrees of arc to 180°.

Some rules base the horse-power on the length of the arc of contact in feet. Since $L = \frac{\pi d a}{12 \times 360}$ and H. P. $= \frac{Srw}{33000} = \frac{Sw}{33000} \times \frac{\pi d}{12} \times$ rpm. $\times$ ... we obtain by substitution H.P. $= \frac{Sw}{16500} \times L \times$ rpm., and the five formulæ take the following form for the several values of S:

$$\text{H.P} = \frac{wL \times \text{rpm.}}{275}\ (1);\quad \frac{wL \times \text{rpm.}}{550}\ (2);\quad \frac{wL \times \text{rpm.}}{500}\ (3);\quad \frac{wL \times \text{rpm.}}{367}$$

$$\text{H.P. (double belt)} = \frac{wL \times \text{rpm.}}{257}\ (5).$$

None of the handy formulæ take into consideration the centrifugal ten- sion of belts at high velocities. When the velocity is over 3000 ft. per min- ute the effect of this tension becomes appreciable, and it should be taken account of as in Mr. Nagle's formula, which is given below.

Horse-power of a Leather Belt One Inch wide.

Formula: H.P. $= CVtw(S - .012V^2) \div 550$.

For $f = .40$, $a = 180°$, $C = .715$, $w = 1$.

Velocity in ft. per sec.	Laced Belts, $S = 275$. Thickness in inches $= t$. 1/7 .143	1/6 .167	3/16 .187	7/32 .219	1/4 .250	5/16 .312	1/3 .333
10	.51	.59	.63	.73	.84	1.05	1.18
15	.75	.88	1.00	1.16	1.32	1.66	1.77
20	1.00	1.17	1.32	1.54	1.75	2.19	2.31
25	1.23	1.43	1.61	1.88	2.16	2.69	2.86
30	1.47	1.72	1.93	2.25	2.58	3.22	3.44
35	1.69	1.97	2.22	2.59	2.96	3.70	3.94
40	1.90	2.22	2.49	2.90	3.32	4.15	4.44
45	2.09	2.45	2.75	3.21	3.67	4.58	4.89
50	2.27	2.65	2.98	3.48	3.98	4.97	5.30
55	2.44	2.84	3.19	3.72	4.26	5.32	5.69
60	2.58	3.01	3.38	3.95	4.51	5.64	6.02
65	2.71	3.16	3.55	4.14	4.74	5.92	6.32
70	2.81	3.27	3.68	4.29	4.91	6.14	6.54
75	2.89	3.37	3.79	4.42	5.05	6.31	6.73
80	2.94	3.43	3.86	4.50	5.15	6.44	6.86
85	2.97	3.47	3.90	4.55	5.20	6.50	6.93
90	2.97	3.47	3.90	4.55	5.20	6.50	6.93

The H.P. becomes a maximum at 87.41 ft. per sec, = 5245 ft. p. min.

Velocity in ft. per sec.	Riveted Belts, $S =$ 40... Thickness in inches = ... 7/32 219	1/4 .250	5/16 .312	1/3 .333	3/8 .375
15	1.69	1.94	2.42	2.58	2.91
20	2.24	2.57	3.21	3.42	3.82
25	2.79	3.19	3.98	4.25	4.78
30	3.31	3.79	4.74	5.05	5.67
35	3.82	4.37	5.46	5.83	6.56
40	4.33	4.95	6.19	6.60	7.42
45	4.85	5.49	6.86	7.32	8.43
50	5.26	6.01	7.51	8.02	9.02
55	5.68	6.50	8.12	8.66	9.74
60	6.09	6.96	8.70	9.28	10.43
65	6.45	7.37	9.22	9.83	11.06
70	6.78	7.75	9.69	10.33	11.62
75	7.09	8.11	10.13	10.84	12.16
80	7.36	8.41	10.51	11.21	12.61
85	7.58	8.66	10.82	11.55	13.00
90	7.74	8.85	11.06	11.80	13.27
100	7.96	9.10	11.37	12.13	13.65

The H.P. becomes a maxi... 105.4 ft. per sec. = 6324 ft. per ...

In the above table the angle of subtension, a, is taken at 180°.

Should it be..	90°	100°	110°	120°	130°	140°	150°	160°	170°
Multiply above values by	.65	.70	.75	.79	.83	.87	.91	.94	.97

A. F. Nagle's Formula (Trans. A. S. M. E., vol. ii., 1881. Tables published in 1882.)

$$\text{H.P.} = CVtw\left(\frac{S - .012V^2}{550}\right);$$

$C = 1 - 10^{-.00758fa}$;
a = degrees of belt contact;
f = co... of friction;
w = ...;

t = thickness in inches;
V = velocity in feet per se...;
S = stress upon belt per s...

Taking S at 275 lbs. per sq. in. for laced belts and 400 lbs. per sq. in. for lapped and riveted belts, the formula becomes

$$\text{H.P.} = CVtw(.50 - .0000218V^2) \text{ for laced belts;}$$
$$\text{H.P.} = CVtw(.727 - .0000218V^2) \text{ for riveted belts.}$$

VALUES OF $C = 1 - 10^{-.00758fa}$. (NAGLE.)

f = coefficient of friction.	Degrees of contact = a.										
	90°	100°	110°	120°	130°	140°	150°	160°	170°	180°	200°
.15	.210	.230	.250	.270	.288	.307	.325	.342	.359	.376	.408
.20	.270	.295	.319	.342	.364	.386	.408	.428	.448	.467	.503
.25	.325	.354	.381	.407	.432	.457	.480	.503	.524	.544	.582
.30	.376	.408	.438	.467	.494	.520	.544	.567	.590	.610	.649
.35	.423	.457	.489	.520	.548	.575	.600	.624	.646	.667	.705
.40	.467	.502	.536	.567	.597	.624	.649	.673	.695	.715	.753
.45	.507	.544	.579	.610	.640	.667	.692	.715	.737	.757	.792
.55	.578	.617	.652	.684	.713	.739	.763	.785	.805	.822	.853
.60	.610	.649	.684	.715	.744	.769	.792	.813	.832	.848	.877
.00	.792	.825	.853	.877	.897	.913	.927	.937	.947	.956	.969

The following table gives a comparison of the formulæ already given for e case of a belt one inch wide, with arc of contact 180°.

orse-power of a Belt One Inch wide, Arc of Contact 180°.

COMPARISON OF DIFFERENT FORMULÆ.

Velocity in ft. p. min.	Sq. ft. of Belt p. min.	Form. 1 H.P. = $\frac{wv}{550}$.	Form. 2 H.P. = $\frac{wv}{1100}$.	Form. 3 H.P. = $\frac{wv}{1000}$.	Form. 4 H.P. = $\frac{wv}{733}$.	Form. 5 dbl. belt H.P. = $\frac{wv}{513}$.	Nagle's Form. 7/32″ single belt Laced.	Nagle's Form. 7/32″ single belt Riveted
600	50	1.09	.55	.60	.82	1.17	.73	1.14
1200	100	2.18	1.09	1.20	1.64	2.34	1.54	2.24
1800	150	3.27	1.64	1.80	2.46	3.51	2.25	3.31
2400	200	4.36	2.18	2.40	3.27	4.68	2.90	4.33
3000	250	5.45	2.73	3.00	4.09	5.85	3.48	5.26
3600	300	6.55	3.27	3.60	4.91	7.02	3.95	6.09
4200	350	7.63	3.82	4.20	5.73	8.19	4.29	6.78
4800	400	8.73	4.36	4.80	6.55	9.36	4.50	7.36
5400	450	9.82	4.91	5.40	7.37	10.53	4.55	7.74
6000	500	10.91	5.45	6.00	8.18	11.70	4.41	7.96
6600	550	....	....	....	.. ....	....	4.05	7.97
7200	600	....	....	....	... .	....	3 49	7.75

dth of Belt for a Given Horse-power.—The width of belt red for any given horse-power may be obtained by transposing the for- r horse-power so as to give the value of w. Thus:

ormula (1), $w = \frac{550 \text{ H.P.}}{v} = \frac{9.17 \text{ H.P.}}{V} = \frac{2101 \text{ H.P.}}{d \times \text{rpm.}} = \frac{275 \text{ H.P.}}{L \times \text{rpm.}}$.

ormula (2), $w = \frac{1100 \text{ H.P.}}{v} = \frac{18.33 \text{ H.P.}}{V} = \frac{4202 \text{ H.P.}}{d \times \text{rpm.}} = \frac{530 \text{ H.P.}}{L \times \text{rpm.}}$.

ormula (3), $w = \frac{1000 \text{ H.P.}}{v} = \frac{16.67 \text{ H.P.}}{V} = \frac{3820 \text{ H.P.}}{d \times \text{rpm.}} = \frac{500 \text{ H.P.}}{L \times \text{rpm.}}$.

ormula (4), $w = \frac{733 \text{ H.P.}}{v} = \frac{12.22 \text{ H.P.}}{V} = \frac{2800 \text{ H.P.}}{d \times \text{rpm.}} = \frac{360 \text{ H.P.}}{L \times \text{rpm.}}$.

ormula (5),* $w = \frac{513 \text{ H.P.}}{v} = \frac{8.56 \text{ H.P.}}{V} = \frac{1960 \text{ H.P.}}{d \times \text{rpm.}} = \frac{257 \text{ H.P.}}{L \times \text{rpm.}}$.

double belts.

Many authorities use formula (1) for double belts and formula (2) or (
single belts.

To obtain the width by Nagle's formula, $w = \frac{550 \text{ H.P.}}{CVt(S - .012V^2)}$, or d
the given horse-power by the figure in the table corresponding to the
thickness of belt and velocity in feet per second.

The formula to be used in any particular case is larg
matter of judgment. A single belt proportioned according to form
if tightly stretched, and if the surface is in good condition, will trans
horse-power calculated by the formula, but one so proportioned is
tionable, first, because it requires so great an initial tension that it is
stretch, slip, and require frequent restretching and relacing; and
because this tension will cause an undue pressure on the pulley-shaf
therefore an undue loss of power by friction. To avoid these diff
formula (2), (3), or (4,) or Mr. Nagle's table, should be used; the lat
cially in cases in which the velocity exceeds 4000 ft. per min.

Taylor's Rules for Belting.—F. W. Taylor (Trans. A. S.
xv. 204) describes a nine years' experiment on belting in a machin
giving results of tests of 42 belts running night and day. Some o
belts were run on cone pulleys and others on shifting, or fast-and-loo
leys. The average net working load on the shifting belts was only
that of the cone belts.

The shifting belts varied in dimensions from 39 ft. 7 in. long, 3.5 i
.25 in. thick, to 51 ft. 5 in. long, 6.5 in. wide, .37 in. thick. The co
varied in dimensions from 24 ft. 7 in. long, 2 in. wide, .25 in. thick,
10 in. long, 4 in. wide, .37 in. thick.

Belt-clamps were used having spring-balances between the two
clamps, so that the exact tension to which the belt was subje
accurately weighed when the belt was first put on, and each tim
tightened.

The tension under which each belt was spliced was carefully figu
to place it under an initial strain—while the belt was at rest imm
after tightening—of 71 lbs. per inch of width of double belts. This
alent, in the case of

Oak tanned and fulled belts,	to 192 lbs.	per sq. in. section
Oak tanned, not fulled belts,	to 229 "	" " " "
Semi-raw-hide belts,	to 253 "	" " " "
Raw-hide belts,	to 284 "	" " " "

From the nine years' experiment Mr. Taylor draws a number o
sions, some of which are given in an abridged form below.

In using belting so as to obtain the greatest economy and the m
factory results, the following rules should be observed:

	Oak Tanned and Fulled Leather Belts.	Other Leat and Rubb
A double belt, having an arc of contact of 180°, will give an effective pull on the face of a pulley per inch of width of belt of....	35 lbs.	30
Or, a different form of same rule: The number of sq. ft. of double Belt passing around a pulley per minute required to transmit one horse-power is...............	80 sq. ft.	90
Or: The number of lineal feet of double-belting 1 in. wide passing around a pulley per minute required to transmit one horse-power is............................	950 ft.	11
Or: A double belt 6 in. wide, running 4000 to 5000 ft. per min., will transmit...........	30 H.P.	2

The terms "initial tension," "effective pull," etc., are thus ex
Mr. *Taylor*: When pulleys upon which belts are tightened are
strands of the belt (the upper and lower) are under the same
of [illegible] tension," "initial tension," or "tension whil

e stress per in. of width, or sq. in. of section, to which one of the of the belt is tightened, when at rest. After the belts are in motion ismitting power, the stress on the slack side, or strand, of the belt less, while that on the tight side—or the side which does the pull-omes greater than when the belt was at rest. By the term "total e mean the total stress per in. of width, or sq. in. of section, on the e of belt while in motion.

fference between the stress on the tight side of the belt and its slack ile in motion, represents the effective force or pull which is trans-rom one pulley to another. By the terms "working load," "net load," or "effective pull," we mean the difference in the tension ight and slack sides of the belt per in. of width, or sq. in. section, motion, or the net effective force that is transmitted from one pul-other per in. of width or sq. in. of section.

scovery of Messrs. Lewis and Bancroft (Trans. A. S. M. E., vii. 749) "sum of the tension on both sides of the belt does not remain ," upsets all previous theoretical belting formulæ.

lt speed for maximum economy should be from 4000 to 4500 ft. per

st distance from centre to centre of shafts is from 20 to 25 ft.

ulleys work most satisfactorily when located on the slack side of about one quarter way from the driving-pulley.

are more durable and work more satisfactorily made narrow and ther than wide and thin.

fe and advisable to use: a double belt on a pulley 12 in. diameter or triple belt on a pulley 20 in. diameter or larger; a quadruple belt ley 30 in. diameter or larger.

ts increase in width they should also be made thicker.

ds of the belt should be fastened together by splicing and cement-ead of lacing, wiring, or using hooks or clamps of any kind.

lice should be used on triple and quadruple belts and when idlers . Stepped splice, coated with rubber and vulcanized in place, is best er belts.

uble belting the rule works well of making the splice for all belts in. wide, 10 in. long; from 10 in. to 18 in. wide the splice should be width as the belt, 18 in. being the greatest length of splice required le belting.

hould be cleaned and greased every five to six months.

leather belts will last well when repeatedly tightened under a hen at rest) of 71 lbs. per in. of width, or 240 lbs. per sq. in. section. l not maintain this tension for any length of time, however.

amps having spring-balances between the two pairs of clamps e used for weighing the tension of the belt accurately each time it ned.

retch, durability, cost of maintenance, etc., of belts proportioned rding to the ordinary rules of a total load of 111 lbs. per inch of rresponding to an effective pull of 65 lbs. per inch of width, and (B) g to a more economical rule of a total load of 54 lbs., corresponding ective pull of 26 lbs. per inch of width, are found to be as follows:

t is impracticable to accurately weigh the tension of a belt in tight-it is safe to shorten a double belt one half inch for every 10 ft. of r (A) and one inch for every 10 ft. for (B), if it requires tightening.

leather belts, when treated with great care and run night and day ate speed, should last for 7 years (A); 18 years (B).

st of all labor and materials used in the maintenance and repairs of elts, added to the cost of renewals as they give out, through a term will amount on an average per year to 37% of the original cost of (A); 14% or less (B).

ring the total expense of belting, and the manufacturing cost le to this account, by far the largest item is the time lost on the s while belts are being relaced and repaired.

tal stretch of leather belting exceeds 6% of the original length.

retch during *the first six* months of the life of belts is 38% of their *etch (A); 15% (B).*

e belt will stretch 47/100 of 1% of its length before requiring to be *(A); 81/100 of 1% (B).*

t important consideration in making up tables and rules for th *re of belting is how* to secure the minimum of interruptions *e from this source.*

The average double belt (A), when running night and day in a machine shop, will cause at least 26 interruptions to manufacture during its life, or interruptions per year, but with (B) interruptions to manufacture will not average oftener for each belt than one in sixteen months.

The oak-tanned and fulled belts showed themselves to be superior in all respects except the coefficient of friction to either the oak-tanned and filled, the semi-raw-hide, or raw-hide with tanned face.

Belts of any width can be successfully shifted backward and forward on tight and loose pulleys. Belts running between 5000 and 6000 ft. per min. and driving 300 H.P. are now being daily shifted on tight and loose pulleys to throw lines of shafting in and out of use.

The best form of belt-shifter for wide belts is a pair of rollers twice the width of belt, either of which can be pressed onto the flat surface of the belt on its slack side close to the driven pulley, the axis of the roller making an angle of 75° with the centre line of the belt.

Remarks on Mr. Taylor's Rules. (Trans. A. S. M. E., xv., [illegible]) —The most notable feature in Mr. Taylor's paper is the great difference between his rules for proper proportioning of belts and those given by other writers. A very commonly used rule is, one horse-power may be transmitted by a single belt 1 in. wide running x ft. per min., substituting for x various values, according to the ideas of different engineers, ranging usually from 550 to 1100.

The practical mechanic of the old school is apt to swear by the figure 600 as being thoroughly reliable, while the modern engineer is more apt to use the figure 1000. Mr. Taylor, however, instead of using a figure from 550 to 1100 for a single belt, uses 950 to 1100 for double belts. If we assume that a double belt is twice as strong, or will carry twice as much power as a single belt, then he uses a figure at least twice as large as that used in modern practice, and would make the cost of belting for a given shop twice as large as if the belting were proportioned according to the most liberal of the customary rules.

This great difference is to some extent explained by the fact that the problem which Mr. Taylor undertakes to solve is quite a different one from that which is solved by the ordinary rules with their variations. The problem of the latter generally is, "How wide a belt must be used, or how narrow a belt may be used, to transmit a given horse-power?" Mr. Taylor's problem is: "How wide a belt must be used so that a given horse-power may be transmitted with the minimum cost for belt repairs, the longest life to the belt, and the smallest loss and inconvenience from stopping the machine while the belt is being tightened or repaired?"

The difference between the old practical mechanic's rule of a 1-in.-wide single belt, 600 ft. per min., transmits one horse-power, and the rule commonly used by engineers, in which 1000 is substituted for 600, is due to the belief of the engineers, not that a horse-power could not be transmitted by the belt proportioned by the older rule, but that such a proportion involved undue strain from overtightening to prevent slipping, which strain entailed too much journal friction, necessitated frequent tightening, and decreased the length of the life of the belt.

Mr. Taylor's rule substituting 1100 ft. per min. and doubling the belt is a further step, and a long one, in the same direction. Whether it will be taken in any case by engineers will depend upon whether they appreciate the extent of the losses due to slippage of belts slackened by use under overstrain, and the loss of time in tightening and repairing belts, to such a degree as to induce them to allow the first cost of the belts to be doubled in order to avoid these losses.

It should be noted that Mr. Taylor's experiments were made on rather narrow belts, used for transmitting power from shafting to machinery, and his conclusions may not be applicable to heavy and wide belts, such as engine fly-wheel belts.

MISCELLANEOUS NOTES ON BELTING.

Formulæ are useful for proportioning belts and pulleys, but they furnish no *means* of estimating how much power a particular belt may be transmitting at any given time, any more than the size of the engine is a measure of the load it is actually drawing, or the known strength of a horse a measure of the load on the wagon. The only reliable means of determining the power [illegible] transmitted is some form of dynamometer. (See [illegible], p. 707.)

hickness, the power transmitted ought to increase in ouble belts we should have half the width required for e same conditions. With large pulleys and moderate probable that this holds good. With small pulleys, uble belt is used, there is not such perfect contact ce and the belt, due to the rigidity of the latter, and ary to bend the belt-fibres than when a thinner and sed. The centrifugal force tending to throw the belt ncreases with the thickness, and for these reasons the t required to transmit a given horse-power when used s generally assumed not less than seven tenths the to transmit the same power. (Flather on "Dynamoment of Power.")

ver, finds that great pliability is objectionable, and n for small pulleys. The power consumed in bending ulley he considers inappreciable. According to Rantrifugal tension, this tension is proportional to the belt, and hence it does not increase with increase of idth is decreased in the same proportion, the sectional ant.

ns. A. S. M. E., x. 765) says: The best belts are made eather, and curried with the use of cod oil and tallow, quality. Such belts have continued in use thirty to d as simple driving-belts, driving a proper amount of ad suitable care. The flesh side should not be run to he reason that the wear from contact with the pulley rain side, as that surface of the belt is much weaker than the flesh side; also as the grain is hard it is more r of attrition; further, if the grain is actually worn off, t suffer in its integrity from a ready tendency of the ck.

contact of a belt with a pulley comes, first, in the y-face, including freedom from ridges and hollows left ond, in the smoothness of the surface and evenness in a belt; third, in having the crown of the driving and rely alike,—as nearly so as is practicable in a commercial ng the crown of pulleys not over 1/8″ for a 24″ face, that ley is not to be over 1/4″ larger in diameter in its centre; crown other than two planes meeting at the centre; naterial on or in a belt, in addition to those necessarily process, to keep them pliable or increase their tractive y depend upon the exigencies arising in the use of r than over-use; seventh, with reference to the lacing e a good practice to cut the ends to a convex shape by at there may be a nearly uniform stress on the lacing s compared with the edges. For a belt 10″ wide, the ould recede 1/10″.

s.—In punching a belt for lacing, use an oval punch, of the punch being parallel with the sides of the belt. oles in each end, placed zigzag. In a 3-in. belt there in each end—two in each row. In a 6-inch belt, seven ow nearest the end. A 10-inch belt should have nine e holes should not come nearer than 3/4 of an inch from inch from the ends of the belt. The second row should from the end. On wide belts these distances should r.

e centre of the belt and take care to keep the ends to lace both sides with equal tightness. The lacing l on the side of the belt that runs next the pulley. In rve the same rules as putting on new ones.

n Quarter-twist.—A belt must run squarely on to ect with a belt two horizontal shafts at right angles an engine-shaft near the floor with a line attached to *re a quarter-turn.* First, ascertain the central point *lley at the extremity of the horizontal diameter where pulley,* and then set that point on the driven pulley *ponding point on the driver.* This will cause the belt *ch pulley,* and it will leave at an angle greater o *ze of the pulleys and their distance from each othe*

is wide a partial vacuum is formed between the be
high velocity. The pressure is then greater than th
tensions in the belt, and the resistance to slipping is
advantage of permitting a greater power to be trans
and of diminishing the strain on the shafting.

On the other hand, some writers claim that the be
itself and the pulley, which tends to diminish the fr
tractive force. On this theory some manufacturers p
numerous holes to let the air escape.

Care of Belts.—Leather belts should be well p
and even loose steam and other moisture.

Belts of coarse, loose leather will do better service in
wet or moist situations the finest and firmest leather s
Hoyt & Co.)

Do not allow oil to drip upon the belts. It destroys

Leather belting cannot safely stand above 110° of he

Strength of Belting.—The ultimate tensile st
not generally enter as a factor in calculations of pow

The strength of the solid leather in belts is from 200
inch; at the lacings, even if well put together, only
riveted, the joint should have half the strength of the
ing strain on the driving side is generally taken at no
strength of the lacing, or from one eighth to one six
of the solid belt. Dr. Hartig found that the tension i
30 to 532 lbs. per square inch, averaging 273 lbs.

Adhesion Independent of Diameter. (S
1. The adhesion of the belt to the pulley is the same—
degrees of contact, aggregate tension or weight bei
reference to width of belt or diameter of pulley.

2. A belt will slip just as readily on a pulley four fee
on a pulley two feet in diameter, provided the conditi
pulleys, the arc of contact, the tension, and the nu
travels per minute are the same in both cases.

3. A belt of a given width, and making any given
minute, will transmit as much power running on pul
eter as it will on pulleys four feet in diameter, provi
tension, and conditions of pulley faces are the same i

4. To obtain a greater amount of power from belt

ed belts this belt would be designed for 332 H.P. By Mr. Taylor's be used to transmit only 123 H.P.
may be taken as examples of what a belt may be made to do, d not be used as precedents in designing. It is not stated how was lost by the journal friction due to over-tightening of these

ssings.—We advise, when the belt is pliable, and only dry and pplication of blood-warm tallow. This applied, and dried in by r sun, will tend to keep the leather in good working condition. e tallow passes into the tallow of the leather, serving to soften earine is left on the outside, to fill the pores and leave a smooth e addition of resin to the tallow for belts, if used in wet or damp e of service and help preserve their strength. Belts which have l and dry should have an application of neat's-foot or liver oil, small quantity of resin. This prevents the oil from injuring the s to preserve it. There should not be so much resin as to leave ky. (J. B. Hoyt & Company.)
ld not be soaked in water before oiling, and penetrating oils eldom be used, except occasionally when a belt gets very dry rom neglect. It may then be moistened a little, and have neat's- ied. Frequent applications of such oils to a new belt render the and flabby, thus causing it to stretch, and making it liable to ne. A composition of tallow and oil, with a little resin or bees- er to use. Prepared castor-oil dressing is good, and may be a brush or rag while the belt is running. (Alexander Bros.)
for Cloth or Leather. (Molesworth.)—16 parts gutta- dia-rubber, 2 pitch, 1 shellac, 2 linseed-oil, cut small, melted to- well mixed.
Belting.—The advantages claimed for rubber belting are ormity in width and thickness; it will endure a great degree of ld without injury; it is also specially adapted for use in damp or or where exposed to the action of steam; it is very durable, and nsile strength, and when adjusted for service it has the most per- the pulleys, hence is less liable to slip than leather.
animal oil or grease on rubber belts, as it will greatly injure and y them.
lts will be improved, and their durability increased, by putting ainter's brush, and letting it dry, a composition made of equal lead, black lead, French yellow, and litharge, mixed with boiled nd japan enough to make it dry quickly. The effect of this will ce a finely polished surface. If, from dust or other cause, the slip, it should be lightly moistened on the side next the pulley linseed-oil. (From circulars of manufacturers.)

GEARING.

TOOTHED-WHEEL GEARING.

Pitch-circle, etc.—If two cylinders with parallel axes are ether and one of them is rotated on its axis, it will drive the other f the friction between the surfaces. The cylinders may be con- pair of spur-wheels with an infinite number of very small teeth. eth are formed upon the cylinders, making alternate elevations sions in the cylindrical surfaces, the distance between the axes he same, we have a pair of gear-wheels which will drive one an- ressure upon the faces of the teeth, if the teeth are properly making the teeth the cylindrical surface may entirely disap- he position it occupied may still be considered as a cylindrical ich is called the "pitch-surface," and its trace on the end of the n a plane cutting the wheel at right angles to its axis, is called circle" or "pitch-line." The diameter of this circle is called the *eter, and the distance* from the face of one tooth to the corre- *e of the next tooth* on the same wheel, measured on an arc of *le, is called the* "pitch of the tooth," or the circular pitch. *ls having teeth* of the same pitch are geared together so that *cles touch, it is a* property of the pitch-circles that their *rtional to the* number of teeth in the wheels, and vice

Chordal pitch = diam. of pitch-circle × sine of $\frac{180°}{\text{No. of teeth}}$. Ch
pitch of a wheel of 10 in. pitch diameter and 10 teeth, $10 \times \sin 18° = 3$
in. Circular pitch of same wheel = 3.1416. Chordal pitch is used wi
or sprocket wheels, to conform to the pitch of the chain.

Formulæ for Determining the Dimensions of Small Ge

(Brown & Sharpe Mfg. Co.)

P = diametral pitch, or the number of teeth to one inch of diame pitch-circle;

D' = diameter of pitch circle....... D = whole diameter....... N = number of teeth....... V = velocity.......	Larger Wheel.	These wh run togethe
d' = diameter of pitch-circle....... d = whole diameter....... n = number of teeth....... v = velocity.......	Smaller Wheel.	

a = distance between the centres of the two wheels;
b = number of teeth in both wheels;
t = thickness of tooth or cutter on pitch-circle;
s = addendum;
D'' = working depth of tooth;
f = amount added to depth of tooth for rounding the corners and clearance;
$D''+f$ = whole depth of tooth;
π = 3.1416.
P' = circular pitch, or the distance from the centre of one tooth to centre of the next measured on the pitch-circle.

Formulæ for a single wheel:

$$P = \frac{N+2}{D};\quad D' = \frac{D \times N}{N+2};\quad D'' = \frac{2}{P} = 2s;\quad s = \frac{1}{P} = \frac{P'}{\pi} = .318$$

$$P = \frac{N}{D'};\quad D' = \frac{N}{P};\quad N = PD';\quad N = PD - 2;\quad s = \frac{D'}{N} = \frac{D}{N+2};$$

$$P' = \frac{\pi}{P};\quad D = \frac{N+2}{P};\quad f = \frac{t}{10};\quad s+f = \frac{1}{P}\left(1+\frac{\pi}{20}\right) = .3$$

$$P = \frac{\pi}{P'};\quad D = D' + \frac{2}{P};\quad t = \frac{1.57}{P} = \tfrac{1}{2}P'.$$

Formulæ for a pair of wheels:

$$b = 2aP;\qquad n = \frac{PD'V}{v};\qquad D = \frac{2a(N+2)}{b};$$

$$N = \frac{nv}{V};\qquad v = \frac{PD'V}{n};\qquad d = \frac{2a(n+2)}{b};$$

$$n = \frac{NV}{v};\qquad v = \frac{NV}{n};\qquad a = \frac{b}{2P};$$

$$N = \frac{bv}{v+V};\qquad V = \frac{nv}{N};\qquad a = \frac{D'+d'}{2};$$

$$n = \frac{bV}{v+V};\qquad D' = \frac{2aV}{v+V};\qquad d' = \frac{2aV}{v+V}.$$

The following proportions of gear wheels are recommended by
(Stevens Indicator, April, 1892.)

Proportions of Gear-wheels.

Circular Pitch.	Outside of Pitch-line. $P \times .3$	Inside of Pitch-line. For Cast or Cut Bevels or for Cast Spurs. $P \times .4$	Inside of Pitch-line. For Cut Spurs. $P \times .35$	Width of Space. For Cast Spurs or Bevels. $P \times .525$	Width of Space. For Cut Bevels or Spurs. $P \times .51$
1/4	.075	.100	.088	.131	.128
.2618	.079	.105	.092	.137	.134
.31416	.094	.126	.11	.165	.16
3/8	.113	.150	.131	.197	.191
.3927	.118	.157	.137	.206	.2
.4477	.134	.179	.157	.235	.228
1/2	.15	.20	.175	.263	.255
.5236	.157	.209	.183	.275	.267
9/16	.169	.225	.197	.295	.287
5/8	.188	.25	.219	.328	.319
.62832	.188	.251	.22	.33	.32
3/4	.225	.3	.263	.394	.383
.7854	.236	.314	.275	.412	.401
7/8	.263	.35	.307	.459	.446
1	.3	.4	.35	.525	.51
1.0472	.314	.419	.364	.55	.534
1 1/8	.338	.45	.394	.591	.574
1.1424	.343	.457	.40	.6	.583
1 1/4	.375	.5	.438	.656	.638
1.25664	.377	.503	.44	.66	.641
1 3/8	.413	.55	.481	.722	.701
1 1/2	.45	.6	.525	.788	.765
1.5708	.471	.628	.55	.825	.801
1 3/4	.525	.7	.613	.919	.893
2	.6	.8	.7	1.05	1.02
2.0944	.628	.838	.733	1.1	1.068
2 1/4	.675	.9	.788	1.181	1.148
2 1/2	.75	1.0	.875	1.313	1.275
2 3/4	.825	1.1	.963	1.444	1.403
3	.9	1.2	1.05	1.575	1.53
3.1416	.942	1.257	1.1	1.649	1.602
3 1/4	.975	1.3	1.138	1.706	1.657
3 1/2	1.05	1.4	1.225	1.838	1.785

Thickness of rim below root = depth of tooth.

idth of Teeth.—The width of the faces of teeth is generally made 2 to 3 times the circular pitch — from 6.28 to 9.42 divided by the diam- pitch. There is no standard rule for width.

e following sizes are given in a stock list of cut gears in "Grant's s:"

etral pitch.....	3	4	6	8	12	16
inches.....	... 3 and 4	2 1/2	1 3/4 and 2	1 1/4 and 1 1/2	3/4 and 1	1/2 and 5/8

e Walker Mfg. Co. give:

lar pitch, in..	1/2	5/8	3/4	7/8	1	1 1/2	2	2 1/2	3	4	5	6
in............	1 1/4	1 1/2	1 3/4	2	2 1/2	4 1/2	6	7 1/2	9	12	16	20

ules for Calculating the Speed of Gears and Pulleys.—relations of the size and speed of driving and driven gear wheels are same as those of belt pulleys. In calculating for gears, multiply or e by the diameter of the pitch-circle or by the number of teeth, as be required. In calculating for pulleys, multiply or divide by their eter *in inches.*

D = diam. of driving wheel, d = diam. of driven, R = revolutions per *e of driver. r = revs. per min. of driven.*

$R = rd \div D$; $r = RD \div d$; $D = dr \div R$; $d = DR \div r$.

number of teeth of driver and n = number of teeth of driven,

$N = nr \div R$; $n = NR \div r$; $R = rn \div N$; $r = RN \div n$.

The rack in the cycloidal system is equivalent to a wheel with an in[finite] number of teeth. The pitch is equal to the circular pitch of the m[ating] gear. Both faces and flanks are cycloids formed by rolling the gen[erating] circle of the mating gear-wheel on each side of the straight pitch-li[ne of] the rack.

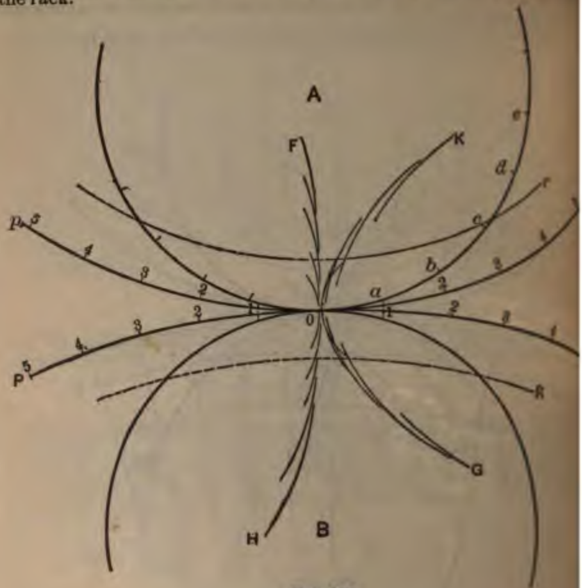


Fig. 155.

Another method of drawing the cycloidal curves is shown in Fig. [155. It] is known as the method of tangent arcs. The generating-[circ]les, as [B], are drawn with equal radii, the length of the radius being less tha[n] [the] radius of pl, the smaller pitch-circle. Equal divisions 1, 2, 3, 4, etc., [are] marked off on the pitch-circles and divisions of the same length step[ped] on one of the generating-circles, as $oabc$, etc. From the points 1, 2, 3, [on] the line po, with radii successively equal to the chord distances oa, [ob, oc,] od, oe, draw the five small arcs F. A line drawn through the outer e[dges of] these small arcs, tangent to them all, will be the hypocycloidal curve [for the] flank of a tooth below the pitch-line pl. From the points 1, 2, 3, etc., [on the] line ol, with radii as before, draw the small arcs G. A line tangent t[o these] arcs will be the epicycloid for the face of the same tooth for which th[e flank] curve has already been drawn. In the same way, from centres on t[he lines] P_0, and oL, with the same radii, the tangent arcs H and K may be [drawn,] which will give the tooth for the gear whose pitch-circle is PL.

If the generating-circle had a radius just one half of the radius of [pl, the] hypocycloid F would be a straight line, and the flank of the tooth [would] have been radial.

The Involute Tooth.—In drawing the involute tooth cur[ve, the] *angle of obliquity*, or the angle which a common tangent to the teeth [makes when] they are in contact at the pitch-point, makes with a line joining th[e centres] of the wheels, is first arbitrarily determined. It is customary to tak[e] ... The pitch-lines pl and PL being drawn in contact at O, the line ... AB is drawn through O normal to a common tangent to the too[th] at th[e an]gle of obliquity to a common tangent to the pit[ch] ...

ngle is 20°. From the centres of the pitch-circles draw circles c t to the line AB. These circles are called base-lines or base- which the involutes F and K are drawn. By laying off conven- s 0, 1, 2, 3, which should each be less than 1/10 of the diameter circle, small arcs can be drawn with successively increasing will form the involute. The involute extends from the points F

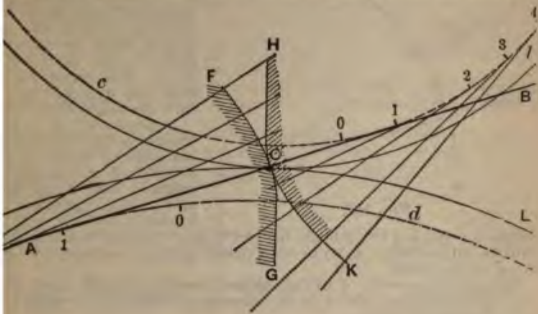


FIG. 156.

to their respective base-circles, where a tangent to the invo- a radius of the circle, and the remainders of the tooth curves, are radial straight lines.

olute system the customary standard form of tooth is one gle of obliquity of 15° (Brown and Sharpe use 14½°), an adden- it one third the circular pitch, and a clearance of about one addendum. In this system the smallest gear of a set has 12 ing the smallest number of teeth that will gear together when his angle of obliquity. In gears with less than 30 teeth the teeth must be slightly rounded over to avoid interference (see h of Gears). All involute teeth of the same pitch and with the f obliquity work smoothly together. The rack to gear with an hed wheel has straight faces on its teeth, which make an angle dle line of the tooth equal to the angle of obliquity, or in the n the faces are inclined at an angle of 30° with each other.

e teeth of a rack which is to gear with an involute wheel (Fig B be the pitch-line of the rack and $AI = II'$ = the pitch. Through

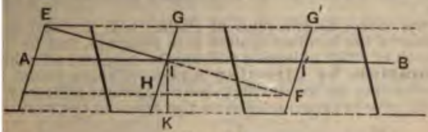


FIG. 157.

int I draw EF at the given angle of obliquity. Draw AE and icular to EF. Through E and F draw lines EGG' and FH par- itch-line. EGG' will be the addendum-line and HF the flank- draw IK perpendicular to AB equal to the greatest addendum wheels of the given pitch and obliquity plus an allowance for ual to ⅙ of the addendum. Through K, parallel to AB, draw -line. The fronts of the teeth are planes perpendicular to EF are planes inclined at the same angle to AB in the con outer half of the working face AE may be slightly c es it a circular arc drawn from a centre on the pi with a radius = 2.1 inches divided by the diametral pitch, or .67 in. × circular pitch.

To Draw an Angle of 15° without using a Protractor.—From C on line AC, with radius AC, describe an arc AB, and from A with the same radius, cut the arc at B. Bisect the arc BA by drawing small arcs at D from A and B as centres, with the same radius, which must be greater than half of AB. Join DC, cutting at E. The angle ECA is 30°. Bisect the arc AE in like manner and the angle FCA will be 15°.

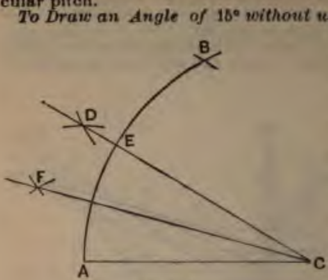


Fig. 158.

A property of involute-toothed wheels is that the distance between the axes of a pair of gears may be altered to a considerable extent without interfering with their action. The backlash is therefore variable at will, and may be adjusted by moving the wheels farther from or nearer to each other, and may thus be adjusted so as to be no greater than is necessary to prevent jamming of the teeth.

The relative merits of cycloidal and involute-shaped teeth are still a subject of dispute, but there is an increasing tendency to adopt the involute tooth for all purposes.

Clark (R. T. D., p. 734) says: Involute teeth have the disadvantage of being too much inclined to the radial line, by which an undue pressure is exerted on the bearings.

Unwin (Elements of Machine Design, 8th ed., p. 265) says: The obliquity of action is ordinarily alleged as a serious objection to involute wheels. Its importance has perhaps been overrated.

George B. Grant (*Am. Mach.*, Dec. 26, 1885) says:

1. The work done by the friction of an involute tooth is always less than the same work for any possible epicycloidal tooth.

2. With respect to work done by friction, a change of the base from a gear of 12 teeth to one of 15 teeth makes an improvement for the epicycloid of less than one half of one per cent.

3. For the 12-tooth system the involute has an advantage of 1½ per cent, and for the 15-tooth system an advantage of ⅜ per cent.

4. That a maximum improvement of about one per cent can be accomplished by the adoption of any possible non-interchangeable radial flank tooth in preference to the 12-tooth interchangeable system.

5. That for gears of very few teeth the involute has a decided advantage.

6. That the common opinion among millwrights and the mechanical public in general in favor of the epicycloid is a prejudice that is founded on long continued custom, and not on an intimate knowledge of the properties of that curve.

Wilfred Lewis (Proc. Engrs. Club of Phila., vol. x., 1893) says a slow reaction in favor of the involute system is in progress, and he believes that an involute tooth of 22½° obliquity will finally supplant all other forms.

Approximation by Circular Arcs.—Having found the form of the actual tooth-curve on the drawing-board, circular arcs may be found by trial which will give approximations to the true curves, and these may

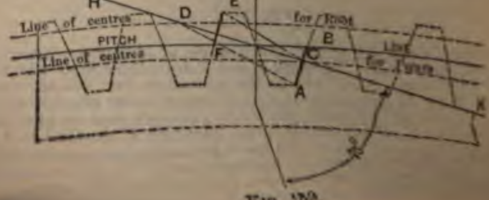


Fig. 159.

mpleting the drawing and the pattern of the gear-wheels. The e curve is connected to the clearance by a fillet, which should be possible to give increased strength to the tooth, provided it is not gh to cause interference.

rth gives the following method of construction by circular arcs: e radial line at the edge of the tooth on the pitch-line, lay off the an angle of 75° with the radial line; on this line will be the cen- root AB and the point EF. The lines struck from these centres in thick lines. Circles drawn through centres thus found will nes in which the remaining centres will be. The radius DA for he root AB is = pitch + the thickness of the tooth. The radius iking the point of the tooth EF = the pitch.

B. Grant says: It is sometimes attempted to construct the curve andy method or empirical rule, but such methods are generally

d Gears.—Two gears of the same pitch and diameter mounted le on the same shaft will act as a single gear. If one gear is keyed aft so that the teeth of the two wheels are not in line, but the ne wheel slightly in advance of the other, the two gears form a ear. If mated with a similar stepped gear on a parallel shaft the teeth in contact will be twice as great as in an ordinary gear, l increase the strength of the gear and its smoothness of action.

ed Teeth.—If a great number of very thin gears were placed one slightly in advance of the other, they would still act as a gear. Continuing the subdivision until the of each separate gear is infinitesimal, the the teeth instead of being in steps take the spiral or twisted surface, and we have a ar. The twist may take any shape, and if it is ection for half the width of the gear and in the direction for the other half, we have what is the herring-bone or double helical tooth. The of the twisted tooth if twisted in one direction end thrust on the shaft, but if the herring- t is used, the opposite obliquities neutralize r. This form of tooth is much used in heavy l practice, where great strength and resistance are necessary. They are frequently made of ngs (Fig. 160). The angle of the tooth with a el to the axis of the gear is usually 30°.

FIG. 160.

Gears.—If a twisted gear has a uniform twist it becomes a r. The line in which the pitch-surface intersects the face of the rt of a helix drawn on the pitch-surface. A spiral wheel may be h only one helical tooth wrapped around the cylinder several which it becomes a screw or worm. If it has two or three teeth d, it is a double- or triple-threaded screw or worm. A spiral-gear nto a rack is used to drive the table of some forms of planing-

-gearing.—When the axes of two spiral gears are at right d a wheel of one, two, or three threads works with a larger wheel threads, it becomes a worm-gear, or endless screw, the smaller

FIG. 161.

ler being called the worm, and the larger, or driven wheel. . With this arrangement a high velocity ratio may be obt pair of wheels. For a one-threaded wheel the velocity r the number of teeth in the worm-wheel. The worm and wheel [...] monly so constructed that the worm will drive the wheel, but the w[...] not drive the worm.

To find the diameter of a worm-wheel at the throat, number of [...] pitch of the worm being given: Add 2 to the number of teeth, mul[...] sum by 0.3183, and by the pitch of the worm in inches.

To find the number of teeth, diameter at throat and pitch of wo[...] given: Divide 3.1416 times the diameter by the pitch, and subtract [...] the quotient.

In Fig. 161 *ab* is the diam. of the pitch-circle, *cd* is the diam. at th[...]

EXAMPLE.—Pitch of worm ¼ in., number of teeth 70, required th[...] at the throat. $(70 + 2) \times .3183 \times .25 = 5.73$ in.

Teeth of Bevel-wheels. (Rankine's Machinery and Mil[...] The teeth of a bevel-wheel have acting surfaces of the conical ki[...] erated by the motion of a line traversing the apex of the conic[...] surface, while a point in it is carried round the traces of the teeth [...] spherical surface described about that apex.

The operations of drawing the traces of the teeth of bevel-wheels [...] whether by involutes or by rolling curves, are in every respect anal[...] those for drawing the traces of the teeth of spur-wheels; except th[...] case of bevel-wheels all those operations are to be performed on the [...] of a sphere described about the apex, instead of on a plane, subs[...] poles for centres and great circles for straight lines.

In consideration of the practical difficulty, especially in the case [...] wheels, of obtaining an accurate spherical surface, and of drawing [...] when obtained, the following approximate method, proposed orig[...] Tredgold, is generally used:

Let *O*, Fig. 162, be the common apex of the pitch-cones, *OBI*, *O*[...] pair of bevel-wheels; *OC*, *OC'*, the axes of those cones; *OI* their line [...] tact. Perpendicular to *O*[...] *AIA'*, cutting the axes in [...] make the outer rims of the [...] and of the wheels portions [...] cones *ABI*, *A'B'I*, of which [...] row zones occupied by the te[...] be sufficiently near for practi[...] poses to a spherical surface d[...] about *O*. As the cones *AB*[...] cut the pitch-cones at right a[...] the outer pitch-circles *IB*, *I*[...] may be called the normal co[...] find the traces of the teeth [...] normal cones, draw on a flat [...] circular arcs, *ID*, *ID'*, with [...] *AI*, *A'I*; those arcs will be [...] velopments of arcs of th[...] circles *IB*, *IB'* when the co[...] faces *ABI*, *A'B'I* are spread out flat. Describe the traces of teet[...] developed arcs as for a pair of spur-wheels, then wrap the devel[...] on the normal cones, so as to make them coincide with the pitch-ci[...] trace the teeth on the conical surfaces.

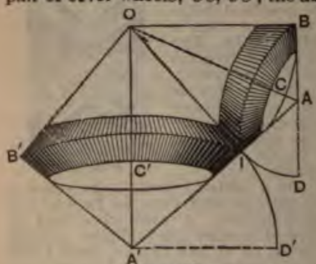


FIG. 162.

For formulæ and instructions for designing bevel-gears, and for m[...] valuable information on the subject of gearing, see "Practical T[...] Gearing," and "Formulas in Gearing," published by Brown & Sh[...] Co.; and "Teeth of Gears," by George B. Grant, Lexington, M[...] student may also consult Rankine's Machinery and Millwork, R[...] Constructor, and Unwin's Elements of Machine Design. See also [...] Gearing, by C. W. MacCord in App. Cyc. Mech., vol. ii.

Annular and Differential Gearing. (S. W. Balch., A[...] Aug. 24, 1893.)—In internal gears the sum of the diameters of the d[...] circles for faces and flanks should not exceed the difference in [...] diameters of the pinion and its internal gear. The sum may be equ[...] difference or it may be less; if it is equal, the faces of the teet[...] *wheel* will drive the faces as well as the flanks of the teeth of [...] *wheel*. The teeth will therefore make contact with each other at [...] at the same time.

Cycloidal tooth-curves for interchangeable gears are formed [...] *ing circles* of about ⅝ the pitch diameter of the smallest gear [...] *to admit two* such circles between the pitch-circles of the pinio[...]

nber of teeth in the internal gear should exceed the number in 12 or more, if the teeth are of the customary proportions and ed in interchangeable gearing.

a less difference is desirable, and the teeth may be modified in to make this possible.

e tooth curves resulting from smaller describing circles may be These will give teeth which are more rounding and narrower at nd therefore not as desirable as the regular forms.

The tips of the teeth may be rounded until they clear. This is a method which aims at modifying the teeth to such outlines as ribing circles would give.

ne of the describing circles may be omitted and one only used. be equal to the difference between the pitch-circles. This will meshing of gears differing by six teeth. It will usually prove to put wheels in inside gears that differ by much less than 12

r diametral pitch and standard tooth forms are determined on, r to which the internal gear-blank is to be bored is calculated by 2 from the number of teeth, and dividing the remainder by the tch.

outlines are the match of a spur-gear of the same number of ametral pitch, so that the spur-gear will fit the internal gear as its die, except that the teeth of each should fail to bottom in aces of the other by the customary clearance of one tenth the the tooth.

earing is particularly valuable when employed in differential is is a mechanical movement in which one of the wheels is a crank so that its centre can move in a circle about the centre wheel. Means are added to the device which restrain the wheel from turning over and confine it to the revolution of the crank. of the number of teeth in the revolving wheel compared with ce between the two will represent the ratio between the revolv- nd the crank-shaft by which the other is carried. The advan- omplishing the change of speed with such an arrangement, as ith ordinary spur-gearing, lies in the almost entire absence of consequent wear of the teeth.

e limitation that the difference between the wheels must not be e possible ratio of speed might be increased almost indefinitely. r of differential gears made to do the service of a whole train of the problem is properly worked out with bevel-gears this limita- e completely set aside, and external and internal bevel-gears, but a single tooth if need be, made to mesh perfectly with each

al bevel-gears have been used with advantage in mowing-ma- description of their construction and operation is given by Mr. article from which the above extracts are taken.

EFFICIENCY OF GEARING.

sive series of experiments on the efficiency of gearing, chiefly piral gearing, is described by Wilfred Lewis in Trans. A. S. M. E., e average results are shown in a diagram, from which the fol- oximate average figures are taken :

EFFICIENCY OF SPUR, SPIRAL, AND WORM GEARING.

aring.	Pitch.	Velocity at Pitch line in feet per min.				
		3	10	40	100	200
....		.90	.935	.97	.98	.985
on..............	45°	.81	.87	.93	.955	.965
....	30	.75	.815	.89	.93	.945
....	20	.67	.75	.845	.90	.92
....	15	.61	.70	.805	.87	.90
or worm.....	10	.51	.615	.74	.82	.86
"	7	.43	.53	.72	.765	.8[illegible]
"	5	.34	.43	.60	.70	[illegible]

The values of s in the above table are given by Mr. Lewis tentatively, in the absence of sufficient data upon which to base more definite values, but they have been found to give satisfactory results in practice.

Mr. Lewis gives the following example to illustrate the use of the table: Let it be required to find the working strength of a 12-toothed pinion of 1-inch pitch, 2½-inch face, driving a wheel of 60 teeth at 100 feet or less per minute, and let the teeth be of the 20-degree involute form. In the formula $W = spfy$ we have for a cast-iron pinion $s = 8000$, $pf = 2.5$, and $y = .078$; and multiplying these values together, we have $W = 1560$ pounds. For the wheel we have $y = .134$ and $W = 2680$ pounds.

FIG. 163.

The cast-iron pinion is, therefore, the measure of strength; but if a steel pinion be substituted we have $s = 20,000$ and $W = 3900$ pounds, in which combination the wheel is the weaker, and it therefore becomes the measure of strength.

For bevel-wheels Mr. Lewis gives the following, referring to Fig. 163: D = large diameter of bevel; d = small diameter of bevel; p = pitch at large diameter; n = actual number of teeth; f = face of bevel; N = formative number of teeth = $n \times$ secant a, or the number corresponding to radius R; y = factor depending on shape of teeth and formative number N; W = working load on teeth.

$$W = spfy\frac{D^3 - d^3}{3D^2(D-d)}; \text{ or, more simply, } W = spfy\frac{d}{D},$$

which gives almost identical results when d is not less than ⅔ D, as is the case in good practice.

In *Am. Mach.*, June 22, 1893, Mr. Lewis gives the following formulæ for the working strength of the three systems of gearing, which agree closely with those obtained by use of the table:

For involute, 20° obliquity, $W = spf\left(.154 - \frac{.912}{n}\right)$;

For involute 15°, and cycloidal, $W = spf\left(.124 - \frac{.684}{n}\right)$;

For radial flank system, $W = spf\left(.075 - \frac{.276}{n}\right)$;

in which the factor within the parenthesis corresponds to y in the original formula. For the horse-power transmitted, Mr. Lewis's general formula $W = spfy, = \frac{33,000 \text{ H.P.}}{v}$, may take the form $\text{H.P.} = \frac{spfyv}{33,000}$, in which v = velocity in feet per minute; or since $v = d\pi \times \text{rpm.} \div 12 = .2618d \times \text{rpm.}$, in which d = diameter in inches and rpm. = revolutions per minute,

$$\text{H.P.} = \frac{Wv}{33,000} = \frac{spfy \times d \times \text{rpm.}}{126,050} = .000007933dspfy \times \text{rpm.}$$

It must be borne in mind, however, that in the case of machines which consume power intermittently, such as punching and shearing machines, the gearing should be designed with reference to the maximum load which can be brought upon the teeth at any time, and not upon the average horse-power transmitted.

Comparison of the Harkness and Lewis Formulæ. Take an average case in which the safe working strength of the metal $s = 6000$, $v = 200$ ft. per min., and $y = .100$, the value in Mr. Lewis's table for an involute tooth of 15° obliquity, or a cycloidal tooth, the number of teeth in the wheel being 27.

$$\text{H.P.} = \frac{spfyv}{33,000} = \frac{6000pfv \times .100}{33,000} = \frac{pfv}{55} = 1.09pfV,$$

if V is taken in feet per second.

[illegible] gives $\text{H.P.} = \frac{0.910Vpf}{\sqrt{1 + 0.65V}}$. If the V in the [illegible]

n at $200 \div 60 = 3\frac{1}{3}$ feet per second. $\sqrt{1+0.65V} = \sqrt{3.167} = 1.78$,
P.$= \frac{.910}{1.78}Vpf = .571pfV$, or about 52% of the result given by Mr. Lewis's
a. This is probably as close an agreement as can be expected, since arkness derived his formula from an investigation of ancient prece- and rule-of-thumb practice, largely with common cast gears, while wis's formula was derived from considerations of modern practice achine moulded and cut gears.

Lewis takes into consideration the reduction in working strength of a lue to increase in velocity by the figures in his table of the values of c working stress s for different speeds. Prof. Harkness gives expres- the same reduction by means of the denominator of his formula, $\overline{0.65V}$. The decrease in strength as computed by this formula is hat less than that given in Mr. Lewis's table, and as the figures given table are not based on accurate data, a mean between the values given formula and the table is probably as near to the true value as may ained from our present knowledge. The following table gives the for different speeds according to Mr. Lewis's table and Prof. Hark- formula, taking for a basis a working stress s, for cast-iron 8000, and el 20,000 lbs. at speeds of 100 ft. per minute and less:

peed of teeth, ft. per min..	100	200	300	600	900	1200	1800	2400
" " ft. per sec..	$1\frac{2}{3}$	$3\frac{1}{3}$	5	10	15	20	30	40
ress s, cast-iron, Lewis...	8000	6000	4800	4000	3000	2400	2000	1700
e do., $s \div 8000$............	1	.75	.6	.5	.375	.3	.25	.2125
$\div \sqrt{1+0.65V}$..............	.6930	.5621	.4850	.3650	.3050	.2672	.2208	.1924
e val. $c \div .693$............	1	.811	.700	.526	.439	.385	.318	.277
00 × ($c \div .693$)............	8000	6488	5600	4208	3512	3080	2544	2216
f s and s_1, cast-iron = s_2.	8000	6200	5200	4100	3300	2700	2300	2000
" " for steel = s_3.	20000	15500	13000	10300	8100	6800	5700	4900
ress for steel, Lewis......	20000	15000	12000	10000	7500	6000	5000	4300

paring the two formulæ for the case of $s = 8000$, corresponding to a f 100 ft. per min., we have

ss: H.P. $= 1 \div \sqrt{1+0.65V} \times .910Vpf = .695 \times .91 \times 1\frac{2}{3}pf = 1.051pf$

$$\text{H.P.} = \frac{spfyv}{33,000} = \frac{spfyV}{550} = \frac{8000 \times 1\frac{2}{3}pfy}{550} = 24.24pfy,$$

h y varies according to the shape and number of the teeth.

adial-flank gear with 12 teeth $y = .052$; $24.24pfy = 1.260pf$;
0° involute, 19 teeth, or 15° inv., 27 teeth $y = .100$; $24.24pfy = 2.424pf$;
5° involute, 300 teeth $y = .150$; $24.24pfy = 3.636pf$.

the weakest-shaped tooth, according to Mr. Lewis, will transmit 20 t more horse-power than is given by Prof. Harkness's formula, in the shape of the tooth is not considered, and the average-shaped according to Mr. Lewis, will transmit more than double the horse- given by Prof. Harkness's formula.

parison of Other Formulæ.—Mr. Cooper, in summing up mination, selected an old English rule, which Mr. Lewis considers as bly correct expression of good general averages, viz.: $X = 2000pf$, eaking load of tooth in pounds, p = pitch, f = face. If a factor of of 10 be taken, this would give for safe working load $W = 200pf$.

ge B. Grant, in his Teeth of Gears, page 33, takes the breaking load f, and, with a factor of safety of 10, gives $W = 350pf$.

om's Pocket-Book, 20th ed., 1891, says: "The strength and durability -iron teeth require that they shall transmit a force of 80 lbs. per inch h and *per inch breadth of face*." This is equivalent to $W = 80pf$, or *of that given by the English rule.*

Halsey (Clark's Pocket Book) gives a table calculated from
H.P. $= pfd \times$ rpm. $\div 850$.
& Laughlins give H.P. $= pfd \times$ rpm. $\div 550$.
rmulæ transformed give $W = 128pf$ and $W = 218pf$, respe

Unwin, on the assumption that the load acts on the corners of the te[eth] derives a formula $p = K\sqrt{W}$, in which K is a coefficient derived from [ex]isting wheels, its values being: for slowly moving gearing not subje[ct to] much vibration or shock $K = .04$; in ordinary mill-gearing, runn[ing at] greater speed and subject to considerable vibration, $K = .05$; and in w[heels] subjected to excessive vibration and shock, and in mortise gearing, K [=] Reduced to the form $W = Cpf$, assuming that $f = 2p$, these values of K [give] $W = 262pf$, $200pf$, and $139pf$, respectively.

Unwin also gives the following formula, based on the assumption th[at the] pressure is distributed along the edge of the tooth: $p = K_1\sqrt{\frac{P}{f}}$

where K_1 = about .0707 for iron wheels and .0848 for mortise wheels [if] the breadth of face is not less than twice the pitch. For the case of f [= 2p] and the given values of K_1 this reduces to $W = 200pf$ and $W =$ [...] respectively.

Box, in his Treatise on Mill Gearing, gives H.P. $= \frac{12p^2f\sqrt{dn}}{1000}$, in wh[ich n] = number of revolutions per minute. This formula differs from the modern formulæ in making the H.P. vary as p^2f, instead of as pf, [and in] this respect it is no doubt incorrect.

Making the H.P. vary as $\sqrt{dn}$ or as $\sqrt{v}$, instead of directly as v, [makes] the velocity a factor of the working strength as in the Harkness and [...] formulæ, the relative strength varying as $\frac{\sqrt{v}}{v}$, or as $\frac{1}{\sqrt{v}}$, which for dif[ferent] velocities is as follows:

Speed of teeth in ft. per min., v =	100	200	300	600	900	1200	180[0]
Relative strength =	1	.707	.574	.408	.333	.289	.2[..]

Showing a somewhat more rapid reduction than is given by Mr. Lew[is].

For the purpose of comparing different formulæ they may in gen[eral be] reduced to either of the following forms:

$$\text{H.P.} = Cpfv, \quad \text{H.P.} = C_1pfd \times \text{rpm.}, \quad W = cpf,$$

in which p = pitch, f = face, d = diameter, all in inches; v = vel[ocity in] feet per minute, rpm. revolutions per minute, and C, C_1 and c coef[ficients]. The formulæ for transformation are as follows:

$$\text{H.P.} = \frac{Wv}{33000} = \frac{W \times d \times \text{rpm.}}{126,050};$$

$$W = \frac{33,000\ \text{H.P.}}{v} = \frac{126,050\ \text{H.P.}}{d \times \text{rpm.}} = 33,000Cpf; \; pf = \frac{\text{H.P.}}{Cv} = \frac{\text{H.P.}}{C_1d \times \text{rpm.}}$$

$$C_1 = .2618C; \quad c = 33,000C; \quad C = 3.82C_1, = \frac{c}{33,000}; \quad c = 126,050C$$

In the Lewis formula C varies with the form of the tooth and [the] speed, and is equal to $sy \div 33,000$, in which y and s are the values tak[en from] the table, and $c = sy$.

In the Harkness formula C varies with the speed and is equal to [...] (V being in feet per second), $= \frac{.01517}{\sqrt{1 + .011v}}$.

In the Box formula C varies with the pitch and also with the [...] and equals $\frac{12p\sqrt{d \times \text{rpm.}}}{1000v} = .02345\frac{p}{\sqrt{v}}$. $c = 33,000C = 774\frac{p}{\sqrt{v}}$

For $v = 100$ ft. per min. $C = 77.4p$; for $v = 600$ ft. per minute [...]

In the other formulæ considered C, C_1, and c are constants. [Reducing] the several formulæ to the form $W = cpf$, we have the followin[g]:

ISON OF DIFFERENT FORMULÆ FOR STRENGTH OF GEAR-TEETH.

king pressure per inch pitch and per inch of face, or value of c in $= cpf$:

	$v = 100$ ft. per min.	$v = 600$ ft. per min.
eak form of tooth, radial flank, 12 teeth...	$c = 416$	208
dium tooth, inv. 15°, or cycloid, 27 teeth..	$c = 800$	400
rong form of tooth, or cycloid, 300 teeth...	$c = 1200$	600
Average tooth..............................	$c = 347$	184
h of 1 inch pitch..........................	$c = 77.4$	31.6
" 3 inches pitch...........................	$c = 232$	95

in which c is independent of form and speed: Old English 00; Grant, $c = 350$; Nystrom, $c = 80$; Halsey, $c = 128$; Jones & , $c = 218$; Unwin, $c = 262$, 200, or 139, according to speed, shock, tion.

e given by Nystrom and those given by Box for teeth of small so much smaller than those given by the other authorities that they jected as having an entirely unnecessary surplus of strength. The en by Mr. Lewis seem to rest on the most logical basis, the form of as well as the velocity being considered; and since they are said to en satisfactory in an extended machine practice, they may be con- eliable for gears that are so well made that the pressure bears face of the teeth instead of upon the corners. For rough ordi- the old English rule $W = 200pf$ is probably as good as any, ex the figure 200 may be too high for weak forms of tooth and for ds.

mula $W = 200pf$ is equivalent to H.P. $= \frac{pfd \times \text{rpm.}}{630} = \frac{pfv}{165}$, or

)15873$pfd \times$ rpm. $= .006063pfv$.

num Speed of Gearing.—A. Towler, *Eng'g*, April 19, 1889, es the maximum speeds at which it was possible under favorable to run toothed gearing safely as follows:

	Ft. per min.
ary cast-iron wheels........................	1800
l " " "	2400
e " " "	2400
ary cast-steel wheels.......................	2600
l " " "	3000
l cast-iron machine-cut wheels..............	3000

oleman Sellers (*Stevens Indicator*, April, 1892) recommends that e not run over 1200 ft. per minute, to avoid great noise. The fg. Co., Cleveland, O., say that 2200 ft. per min. for iron gears and r wood and iron (mortise gears) are excessive, and should be possible. The Corliss engine at the Philadelphia Exhibition (1876) wheel 30 ft. in diameter running 35 rpm. geared into a pinion 12 ft. e speed of the pitch-line was 3300 ft. per min.

avy Machine-cut Spur-gear was made in 1891 by the fg. Co., Cleveland, O., for a diamond mine in South Africa, with s as follows: Number of teeth, 192; pitch diameter, 30′ 6.66″; face, , 6″; bore, 27″; diameter of hub, 9′ 2″; weight of hub, 15 tons; and ht of gear, 66¾ tons. The rim was made in 12 segments, the joints ments being fastened with two bolts each. The spokes were bolted ddle of the segments and to the hub with four bolts in each end.

onal Gearing.—In frictional gearing the wheels are toothless, wheel drives the other by means of the friction between the two which are pressed together. They may be used where the power smitted is not very great; when the speed is so high that toothed ould be noisy; when the shafts require to be frequently put into of gear or to have their relative direction of motion reversed; or desired to change the velocity-ratio while the machinery is in mo- the case of disk friction-wheels for changing the feed in machine

the normal pressure in pounds at the line of contact by which s are pressed together. T = tangential resistance of the driven *e line of contact*, f = the coefficient of friction, V = the velocity *h-surface in feet per second*, and H.P. = horse-power; the *qual to or less than* fP; H.P. $= TV \div 550$. The value of f i metal on metal may be taken at .15 to .20; for wood on metal, .25 to .30; for wood on compressed paper, .20. The tangential driving force T m[illegible] as high as 80 lbs. per inch width of face of the driving surface, but this [illegible] companied by great pressure and friction on the journal-bearings.

In frictional grooved gearing circumferential wedge-shaped groov[illegible] cut in the faces of two wheels in contact. If P = the force press[illegible] wheels together, and N = the normal pressure on all the grooves, [illegible] $(\sin a + f \cos a)$, in which $2a$ = the inclination of the sides of the gr[illegible] and the maximum tangential available force $T = fN$. The inclinati[illegible] sides of the grooves to a plane at right angles to the axis is usually 3[illegible]

Frictional Grooved Gearing.—A set of friction-gears for [illegible] mitting 150 H.P. is on a steam-dredge described in Proc. Inst. M. E. 1888. Two grooved pinions of 54 in. diam., with 9 grooves of 1¾ in. pit[illegible] angle of 40° cut on their face, are geared into two wheels of 127½ in[illegible] similarly grooved. The wheels can be thrown in and out of gear by operating eccentric bushes on the large wheel-shaft. The circumf[illegible] speed of the wheels is about 500 ft. per min. Allowing for engine-f[illegible] if half the power is transmitted through each set of gears the tan[illegible] force at the rims is about 3960 lbs., requiring, if the angle is 40° and [illegible] efficient of friction 0.18, a pressure of 7524 lbs. between the whe[illegible] pinion to prevent slipping.

The wear of the wheels proving excessive, the gears were replaced [illegible] gear wheels and brake-wheels with steel brake-bands, which arran[illegible] has proven more durable than the grooved wheels. Mr. Daniel A[illegible] states that if the frictional wheels had been run at a higher speed the[illegible] would have been better, and says they should run at least 30 ft. per [illegible]

HOISTING.

Approximate Weight and Strength of Cordage. [illegible] and Lockport Block Co.)—See also pages 339 to 345.

Size in Circumference.	Size in Diameter.	Weight of 100 ft. Manila, in lbs.	Strength of Manila Rope, in lbs.	Size in Circumference.	Size in Diameter.	Weight of 100 ft. Manila, in lbs.
inch.	inch.			inch.	inch.	
2	⅝	13	4,000	4¾	1 9/16	72
2¼	¾	16	5,000	5	1⅝	80
2½	13/16	20	6,250	5½	1¾	97
2¾	⅞	24	7,500	6	2	113
3	1	28	9,000	6½	2⅛	133
3¼	1 1/16	33	10,500	7	2¼	153
3½	1⅛	38	12,250	7½	2½	184
3¾	1¼	45	14,000	8	2⅝	211
4	1 5/16	51	16,000	8½	2⅞	236
4¼	1⅜	58	18,062	9	3	262
4½	1½	65	20,250			

Working Strength of Blocks. (B. & L. Block Co.)

Regular Mortise-blocks Single and Double, or Two Double Iron-strapped Blocks, will hoist about—		Wide Mortise and Extra Single and Double, or Two Iron-strapped Blocks, w[illegible] about—	
inch.	lbs.	inch.	lbs.
5	250	8	2,000
6	350	10	6,000
7	600	12	12,000
8	1,200	14	24,000
9	2,000	16	[illegible]
10	4,000	18	[illegible]
12	10,000	20	[illegible]
14	16,000		

Wh[illegible] double and triple block are used together, a certain [illegible] of weight can be safely hoisted, as larger hook[illegible]

Comparative Efficiency in Chain-blocks both in Hoisting and Lowering.

(Tests by Prof. R. H. Thurston, *Hoisting*, March, 1894.)

Number of Block.	Work of Hoisting. Load of 2000 lbs.				Work of Lowering. Load of 2000 lbs., lowered 7 ft. in each case.					
					Exclusive of Factor of Time.				Inclusive of Time.	
	Waste by Friction, per cent.	Actual Efficiency, per cent.	Relative Efficiency.	Velocity-ratio.	Pull on Hand Chain, lbs.	Length of Hand Chain, feet.	Work performed, ft.-lbs.	Relative Force expended by Operator.	Time in Min.	Relative Efficiency.
1	20.50	79.50	1.00	32.50	8.00	227.	1,816	1.00	0.75	1.000
2	68.00	32.00	.40	62.44	14.00	436.	6,104	3.33	1.20	.186
3	69.00	31.00	.39	30.00	92.30	196.	18,090	10.00	1.50	.050
4	71.20	28.80	.36	28.00	92.60	168.	15,556	8.60	2.50	.035
5	73.96	26.04	.33	48.00	73.30	17.5	1,282	0.71	2.80	.380
6	75.66	24.34	.31	53.00	56.60	270.	20,342	11.60	1.80	.036
7	77.00	23.00	.29	44.30	55.00	310.	17,050	9.40	2.75	.029
8	81.03	18.97	.24	61.00	48.50	426.	20,000	11.60	3.75	.018

No. 1 was Weston's triplex block; No. 3, Weston's differential; No. 4 'eston's imported. The others were from different makers, whose name e not given. All the blocks were of one-ton capacity.

Proportions of Hooks.—The following formulæ are given by enry R. Towne, in his Treatise on Cranes, as a result of an extensive cperimental and mathematical investi-ation. They apply to hooks of capaci-es from 250 lbs. to 20,000 lbs. Each size hook is made from some commercial ze of round iron. The basis in each se is, therefore, the size of iron of hich the hook is to be made, indicated A in the diagram. The dimension D arbitrarily assumed. The other di-ensions, as given by the formulæ, are ose which, while preserving a proper earing-face on the interior of the hook r the ropes or chains which may be assed through it, give the greatest re-istance to spreading and to ultimate upture, which the amount of material the original bar admits of. The sym-ol Δ is used to indicate the nominal ca-acity of the hook in tons of 2000 lbs. he formulæ which determine the lines f the other parts of the hooks of the everal sizes are as follows, the measure-nents being all expressed in inches:

FIG. 164.

$D = .5\ \Delta + 1.25$ $\quad G = .75D.$ $\quad H = 1.08A$ $\quad L = 1.05A$

$E = .64\ \Delta + 1.60$ $\quad O = .363\ \Delta + .66$ $\quad I = 1.33A$ $\quad M = .50A$

$F = .33\ \Delta + .85$ $\quad Q = .64\ \Delta + 1.60$ $\quad J = 1.20A$ $\quad N = .85B - .16$

$K = 1.13A$ $\quad U = .866A$

The dimensions A are necessarily based upon the ordinary merchant size f round iron. The sizes which it has been found best to select are th ollowing:

apacity of hook:											
⅛	¼	½	1	1½	2	3	4	5	6	8	10
nension A:											
	11/16	¾	1 1/16	1¼	1⅜	1¾	2	2¼	2½	2⅞	

h_1 = reduced length of rope in t attached to ascending cage;
h_2 = increased length of rope in t attached to descending cage;
w = weight of rope per foot in pounds. Then

$$A = \frac{\left[\left(\frac{Wv^2}{2gt}\right) + \left\{\left(\frac{vt}{L2}\right) - \frac{h_1w + h_2w}{2}\right\}\right]R}{PNSC.}$$

Applying the above formula when designing new engines, Mr found that 30 inches diameter of cylinders would produce equal res balanced, to those of the 36-inch cylinder in use, the latter bei anced.

Counterbalancing may be employed in the following methods:

(*a*) *Tapering Rope.*—At the initial stage the tapering rope ena wind from greater depths than is possible with ropes of uniform The thickness of such a rope at any point should only be such as bear the load on it at that point.

With tapering ropes we obtain a smaller difference between the final load, but the difference is still considerable, and for perfect tion of the load we must rely on some other resource. The theory ropes is to obtain a rope of uniform strength, thinner at the cage the weight is least, and thicker at the drum end where it is greate

(*b*) *The Counterpoise System* consists of a heavy chain workin down a staple pit, the motion being obtained by means of a spe drum placed on the same axis as the winding drum. It is so arra the chain hangs in full length down the staple pit at the commen the winding; in the centre of the run the whole of the chain re bottom of the pit, and, finally, at the end of the winding the co has been rewound upon the small drum, and is in the same cond was at the commencement.

(*c*) *Loaded-wagon System.*—A plan, formerly much employ have a loaded wagon running on a short incline in place of this he the rope actuating this wagon being connected in the same man above to a subsidiary drum. The incline was constructed steep a mencement, the inclination gradually decreasing to nothing. At ning of a wind the wagon was at the top of the incline, and durin of the run gradually passed down it till, at the meet of cages, n exerted on the engine—the wagon by this time being at the botto latter part of the wind the resistance was all against the engine its having to pull the wagon up the incline, and this resistance from nothing at the meet of cages to its greatest quantity at the of the lift.

(*d*) *The Endless-rope System* is preferable to all others, if the cient sump room and the shaft is free from tubes, cross timbers impediments. It consists in placing beneath the cages a tail ro in diameter to the winding rope, and, after conveying this down t attached beneath the other cage.

(*e*) *Flat Ropes Coiling on Reels.*—This means of winding allow tain equalization, for the radius of the coil of ascending rope co increase, while that of the descending one continues to diminis quently, as the resistance decreases in the ascending load th increases, and as the power increases in the other, the leverage d The variation in the leverage is a constant quantity, and is eq thickness of the rope where it is wound on the drum.

By the above means a remarkable uniformity in the load m tained, the only objection being the use of flat ropes, which weig and only last about two thirds the time of round ones.

(*f*) *Conical Drums.*—Results analogous to the preceding may b by using round ropes coiling on conical drums, which may either with the successive coils lying side by side, or they may be provi spiral groove. The objection to these forms is, that perfect equa not obtained with the conical drums unless the sides are very steep sequently there is great risk of the rope slipping; to obviate drums were proposed. They are, however, very expensive, and displacement of the winding rope from the centre line of the very great, owing to their necessary large width.

(*g*) *The Koepe System of Winding.*—An iron pulley with a groove takes the place of the ordinary drum. The windin fr over its head-gear pulley, round the drum, the other head-gear pulley, is connected with the second cage. The rope thus encircles about half the periphery of the drum in the anner as a driving-belt on an ordinary pulley. There is a balance neath the cages, passing round a pulley in the sump; the arrange- ay be likened to an endless rope, the two cages being simply points hment.

BELT-CONVEYORS.

n-elevators. — American Grain-elevators are described in a y E. Lee Heidenreich, read at the International Engineering Con- Chicago (Trans. A. S. C. E. 1893). See also Trans. A. S. M. E. vii, 660.
ls for carrying Grain.—Flexible-rubber bands are exten- sed for carrying grain in and around elevators and warehouses. An n the grain-storage warehouses of the Alexandria Dock, Liverpool ist. M. E., July, 1891), describes the performance of these bands, ting three miles in length. A band 16½ inches wide, 1270 feet long, 9 to 10 feet per second has a carrying capacity of 50 tons per hour. paper on Belts as Grain Conveyors, by T. W. Hugo, Trans. A. S. . 400.

ying-bands or Belts are used for the purpose both of sorting of removing impurities. These carrying-bands may be said to be to two descriptions, namely, the wire belt, which consists of an length of woven wire; and the steel-plate belt, which consists of hree endless chains, carrying steel plates varying in width from 6 14 inches. (Proc. Inst. M. E., July, 1890.)

CRANES.

ification of Cranes. (Henry R. Towne, Trans. A. S. M. E., iv. vised in *Hoisting*, published by The Yale & Towne Mfg. Co.)
st is a machine for raising and lowering weights. A Crane is a h the added capacity of moving the load in a horizontal or lateral .

are divided into two classes, as to their motions, viz., *Rotary* and *ear*, and into four groups, as to their source of motive power, viz.:
—When operated by manual power.
—When driven by power derived from line shafting.
, *Electric, Hydraulic, or Pneumatic.*—When driven by an engine or ttached to the crane, and operated by steam, electricity, water, or mitted to the crane from a fixed source of supply.
otive.—When the crane is provided with its own boiler or other r of power, and is self-propelling; usually being capable of both nd rectilinear motions.
and Rectilinear Cranes are thus subdivided:

Rotary Cranes.

ing-cranes.—Having rotation, but no trolley motion.
-cranes.—Having rotation, and a trolley travelling on the jib.
umn-cranes.—Identical with the jib-cranes, but rotating around a umn (which usually supports a floor above).
lar-cranes.—Having rotation only; the pillar or column being sup- ntirely from the foundation.
lar Jib-cranes.—Identical with the last, except in having a jib and notion.
rick-cranes.—Identical with jib-cranes, except that the head of the held in position by guy-rods, instead of by attachment to a roof or

lking-cranes.—Consisting of a pillar or jib-crane mounted on wheels nged to travel longitudinally upon one or more rails.
omotive-cranes.—Consisting of a pillar crane mounted on a truck, vided with a steam-engine capable of propelling and rotating the nd of hoisting and lowering the load.

Rectilinear Cranes.

dge-cranes.—Having a fixed bridge spanning an opening, and a *oving across the bridge.*
m-cranes.—Consisting of a truck, or short bridge, travelling *on overhead rails,* and without trolley motion.
elling-cranes.—Consisting of a bridge moving longitudi *acks, and a* trolley moving transversely on the bridge.

the crane. (Hoisting.)

A Large Travelling-crane, designed and built b Engineering Co., Alliance, O., for the 12-inch-gun shop at th Navy Yard, is described in *American Machinist*, June 12, 1 150 net tons; distance between centres of inside rails, 59 ft. 6 cross travel, 44 ft. 2 in.; effective lift, 40 ft.; four speeds for 4, and 8 ft. per min.; loads for these speeds, 150, 75, 37½, and 1 tively; traversing speeds of trolley on bridge, 25 and 50 f speeds of bridge on main track, 30 and 60 ft. per minute. Sq employed for driving.

A 150-ton Pillar-crane was erected in 1893 on Fi Glasgow. The jib is formed of two steel tubes, each 39 in. long. The radius of sweep for heavy lifts is 65 ft. The jib a counterbalanced by a balance-box weighted with 100 tons of punchings. In a test a 130-ton load was lifted at the rate of 4 and a complete revolution made with this load in 5 minutes July 20, 1893.

Compressed-air Travelling-cranes.—Compresse travelling-cranes have been built by the Lane & Bodley Co. They are of 20 tons nominal capacity, each about 50 ft. span a of travel, and are of the triple-motor type, a pair of simple re being used for each of the necessary operations, the pair of bridge and the pair for the trolley travel being each 5-inc stroke, while the pair for hoisting is 7-inch bore by 9-inch furnished by a compressor having steam and air cylinders e and 12-in. stroke, which with a boiler-pressure of about 80 pou pressure when required of somewhat over 100 pounds. The is allowed to run continuously without a governor, the speed by the resistance of the air in a receiver. From a pipe exte receiver along one of the supporting trusses communication maintained with an auxiliary receiver on each traveller by inch hose, the object of the auxiliary receiver being to prov air near the engines for immediate demands and independ connection, which may thus be of small dimension. Some of said to be possessed by this type of crane are: simplicity; abs ing parts, excepting those required for a particular motion w is in use; no danger from fire, leakage, electric shocks, or f repair; variable speeds and reversal without gearing; almos of noise; and moderate cost.

Quay-cranes.—An illustrated description of several v tionary and travelling cranes, with results of experiments paper on Quay-cranes in the Port of Hamburg by Chas. Nel

ail-rope System.
ndless-rope System.
able Tramway.

ving brief description of these systems is abridged from a n Wire-rope Haulage, by Wm. Hildenbrand, C.E., published by bling's Sons Co., Trenton, N. J.

Self-acting Inclined Plane.—The motive power for the nclined plane is gravity; consequently this mode of transports application only in places where the coal is conveyed from a lower point and where the plane has sufficient grade for the nding cars to raise the empty cars to an upper level.

d of the plane there is a drum, which is generally constructed ving a diameter of seven to ten feet. It is placed high enough n and cars to pass under it. Loaded cars coming from the pit ingly or in sets of two or three switched on the track of the heir speed in descending is regulated by a brake on the drum.

g rollers, to prevent the rope dragging on the ground, are wood, 5 to 6 inches in diameter and 18 to 24 inches long, with iron axles. The distance between the rollers varies from 15 to per planes requiring less rollers than those with easy grades. only the reduction of friction and what is best for the preserva- e, a general rule may be given to use rollers of the greatest meter, and to place them as close as economy will permit.

est angle of inclination at which a plane can be made self-acting en the motive and resisting forces balance each other. The es are the weights of the loaded car and of the descending rope. g forces consist of the weight of the empty car and ascending rolling and axle friction of the cars, and of the axle friction of ing rollers. The friction of the drum, stiffness of rope, and air may be neglected. A general rule cannot be given, because the length of the plane or in the weight of the cars changes the f the forces; also, because the coefficient of friction, depending tion of the road, construction of the cars, etc., is a very uncer-

ng a plane with a 5/8-inch steel rope and lowering from one to weighing empty 1400 lbs. and loaded 4000 lbs., the rise in 100 ry to make the plane self-acting will be from about 5 to 10 feet, s the number of cars increase, and increasing as the length of ses.

inclined plane should be slightly concave, steeper at the top bottom. The maximum deflection of the curve should be at an f 45 degrees, and diminish for smaller as well as for steeper

Simple Engine-plane.—The name "Engine-plane" is ane on which a load is raised or lowered by means of a single d stationary steam-engine. It is a cheap and simple method of oal underground, and therefore is applied wherever circum- nit it.

inary conditions such as prevail in the Pennsylvania mine in of twenty-five to thirty loaded cars will descend, with reason- , a straight plane 5000 feet long on a grade of 1¾ feet in 100, ld appear that 2¼ feet in 100 is necessary for the same number rs. For roads longer than 5000 feet, or when containing sharp rade should be correspondingly larger.

e Tail-rope System.—Of all methods for conveying coal by wire rope, the tail-rope system has found the most applica- be applied under almost any condition. The road may be urved, level or undulating, in one continuous line or with side n general principle a tail-rope plane is the same as an engine- d in both directions with two ropes. One rope, called the "main s for drawing the set of full cars outward; the other, called pe," is necessary to take back the empty set, which on a level g road cannot *return* by gravity. The two drums may be *be opposite ends* of the road, and driven by separate engines, *equently they are on* the same shaft at one end of the plane *se each rope would* require the length of the plane, but the *e tail rope* must be twice as long, being led from the *re at the other* end of the plane and back again to its sta point. When the main rope draws a set of full cars out, the tail-rope drum runs loose on the shaft, and the rope, being attached to the rear car, unwinds itself steadily. Going in, the reverse takes place. Each drum is provided with a brake to check the speed of the train on a down grade and prevent its overrunning the forward rope. As a rule, the tail rope is strained less than the main rope, but in cases of heavy grades dipping outward it is possible that the strain in the former may become as large, or even larger, than in the latter, and in the selection of the sizes reference should be had to this circumstance.

IV. The Endless-rope System.—The principal features of the system are as follows:

1. The rope, as the name indicates, is endless.
2. Motion is given to the rope by a single wheel or drum, and friction is obtained either by a grip-wheel or by passing the rope several times around the wheel.
3. The rope must be kept constantly tight, the tension to be produced by artificial means. It is done in placing either the return-wheel or an extra tension wheel on a carriage and connecting it with a weight hanging over a pulley, or attaching it to a fixed post by a screw which occasionally can be shortened.
4. The cars are attached to the rope by a grip or clutch, which can take hold at any place and let go again, starting and stopping the train at will without stopping the engine or the motion of the rope.
5. On a single-track road the rope works forward and backward, but on a double track it is possible to run it always in the same direction, the full cars going on one track and the empty cars on the other.

This method of conveying coal, as a rule, has not found as general an introduction as the tail-rope system, probably because its efficacy is not so apparent and the opposing difficulties require greater mechanical skill and more complicated appliances. Its advantages are, first, that it requires one third less rope than the tail-rope system. This advantage, however, is partially counterbalanced by the circumstance that the extra tension in the rope requires a heavier size to move the same load than when a main and tail rope are used. The second and principal advantage is that it is possible to start and stop trains at will without signalling to the engineer. On the other hand, it is more difficult to work curves with the endless system, and still more so to work different branches, and the constant stretch of the rope under tension or its elongation under changes of temperature frequently causes the rope to slip on the wheel, in spite of every attention, causing delay in the transportation and injury to the rope.

V. Wire-rope Tramways.—The methods of conveying products in a suspended rope tramway find especial application in places where a mine is located on one side of a river or deep ravine and the loading station on the other. A wire rope suspended between the two stations forms the track on which material in properly constructed "carriages" or "buggies" is transported. It saves the construction of a bridge or trestlework, and is practical for a distance of 2000 feet without an intermediate support.

There are two distinct classes of rope tramways:

1. The rope is stationary, forming the track on which a bucket holding the material moves forward and backward, pulled by a smaller endless wire rope.
2. The rope is movable, forming itself an endless line, which serves at the same time as supporting track and as pulling rope.

Of these two the first method has found more general application, and is especially adapted for long spans, steep inclinations, and heavy loads. The second method is used for long distances, divided into short spans, and is only applicable for light loads which are to be delivered at regular intervals.

For detailed descriptions of the several systems of wire-rope transportation, see circulars of John A. Roebling's Sons Co., The Trenton Iron Co., and other wire-rope manufacturers. See also paper on Two-rope Haulage Systems, by R. Van A. Norris, Trans. A. S. M. E., xii. 626.

In the Bleichert System of wire-rope tramways, in which the track rope is stationary, loads of 1000 pounds each and upward are carried. While the average spans on a level are from 150 to 200 feet, in crossing rivers, cañons, etc., spans up to 1500 feet are frequently adopted. In a tramway on this system at Granite, Montana, the total length of the line is [illegible] feet, with a fall of 1225 feet. The descending loads, amounting to a constant weight of about 11 tons, develop over 14 horse-power, which is sufficient to haul the empty [illegible] as well as about 50 tons of supplies per day up the [illegible] to run the ore crusher and elevator. It is capable of delivering 250 of material in 10 hours.

SUSPENSION CABLEWAYS OR CABLE HOISTS.

(Trenton Iron Co.)

quarrying, rock-cutting, stripping, piling, dam-building, and many r operations where it is necessary to hoist and convey large individual s economically, it frequently happens that the application of a system errricks is impracticable, by reason of the limited area of their efficiency the room which they occupy.

meet such conditions cable-hoists are adapted, as they can be efficiently ated in clear spans up to 1500 feet, and in lifting individual loads up to ns. Two types are made—one in which the hoisting and conveying are by separate running ropes, and the other applicable only to inclines, hich the carriage descends by gravity, and but one running rope is re-ed. The moving of the carriage in the former is effected by means of ndless rope, and these are commonly known as "endless-rope" cable-s to distinguish them from the latter, which are termed "inclined" -hoists.

e general arrangement of the endless-rope cable-hoists consists of a cable passing over towers, A frames or masts, as may be most conve-, and anchored firmly to the ground at each end, the requisite tension e cable being maintained by a turnbuckle at one anchorage.

on this cable travels the carriage, which is moved back and forth over ine by means of the endless rope. The hoisting is done by a separate both ropes being operated by an engine specially designed for the ose, which may be located at either end of the line, and is constructed ch a way that the hoisting-rope is coiled up or paid out automatically e carriage is moved in and out. Loads may be picked up or discharged y point along the line. Where sufficient inclination can be obtained in nain cable for the carriage to descend by gravity, and the loading and ading is done at fixed points, the endless rope can be dispensed with. carriage, which is similar in construction to the carriage used in the ess-rope cableways, is arrested in its descent by a stop-block, which be clamped to the main cable at any desired point, the speed of the ending carriage being under control of a brake on the engine-drum.

Stress in Hoisting-ropes on Inclined Planes.

(Trenton Iron Co.)

100 ft. horizontal.	Angle of inclination.	Stress in lbs. per ton of 2000 lbs.	Rise per 100 ft. horizontal.	Angle of inclination.	Stress in lbs. per ton of 2000 lbs.	Rise per 100 ft. horizontal.	Angle of inclination.	Stress in lbs. per ton of 2000 lbs.
t.			ft.			ft.		
[illegible]	2° 52′	140	55	28° 49′	1008	110	47° 44′	1516
[illegible]	5° 43′	240	60	30° 58′	1067	120	50° 12′	1573
[illegible]	8° 32′	336	65	33° 02′	1128	130	52° 26′	1620
[illegible]	11° 10′	432	70	35° 00′	1185	140	54° 28′	1663
[illegible]	14° 03′	527	75	36° 53′	1238	150	56° 19′	1699
[illegible]	16° 42′	613	80	38° 40′	1287	160	58° 00′	1730
[illegible]	19° 18′	700	85	40° 22′	1332	170	59° 33′	1758
[illegible]	21° 49′	782	90	42° 00′	1375	180	60° 57′	1782
[illegible]	24° 14′	860	95	43° 32′	1415	190	62° 15′	1804
[illegible]	26° 34′	933	100	45° 00′	1450	200	63° 27′	1822

e above table is based on an allowance of 40 lbs. per ton for rolling fric- but an additional allowance must be made for stress due to the weight ne rope proportional to the length of the plane. A factor of safety of 5 should be taken.

hoisting the slack-rope should be taken up gently before beginning the otherwise a severe extra strain will be brought on the rope.

e best *rope for inclined planes* is composed of six strands of seven wires *laid about a hempen centre.* The wires are much coarser than those *114-wire rope of the same* diameter, and for this reason the 42 *s better adapted to* withstand the rough usage and surfa *tered upon inclined planes.*

uble-suspension Cableway, carrying loads of 26 tons, ere

Williamsport, Pa., by the Trenton Iron Co., is described by E. G. Spilsbury in Trans. A. I. M. E. xx. 766. The span is 733 feet, crossing the Susquehanna River. Two steel cables, each 2 in. diam., are used. On these cables runs a carriage supported on four wheels and moved by an endless cable 1 inch in diam. The load consists of a cage carrying a railroad-car loaded with lumber, the latter weighing about 12 tons. The power is furnished by a 50-H.P. engine, and the trip across the river is made in about three minutes.

A hoisting cableway on the endless-rope system, erected by the Lidgerwood Mfg. Co., at the Austin Dam, Texas, had a single span 1350 ft. in length, with main cable 2½ in. diam., and hoisting-rope 1¾ in. diam. Loads of 7 to 8 tons were handled at a speed of 600 to 800 ft. per minute.

Tension required to Prevent Slipping of Wire on Drum. (Trenton Iron Co.)—The amount of artificial tension to be applied in an endless rope to prevent slipping on the driving-drum depends on the character of the drum, the condition of the rope and number of laps which it makes. If T and S represent respectively the tensions in the taut and slack lines of the rope; W, the necessary weight to be applied to the tail-sheave; R, the resistance of the cars and rope, allowing for friction; n, the number of half-laps of the rope on the driving-drum; and f, the coefficient of friction, the following relations must exist to prevent slipping:

$$T = Se^{fn\pi}, \quad W = T + S, \quad \text{and} \quad R = T - S;$$

from which we obtain $$W = \frac{e^{fn\pi} + 1}{e^{fn\pi} - 1} R,$$

in which $e = 2.71828$, the base of the Naperian system of logarithms.

The following are some of the values of f:

	Dry.	Wet.	Greasy.
Rope on a grooved iron drum....	.120	.085	.070
Rope on wood-filled sheaves......	.235	.170	.140
Rope on rubber and leather filling.........	.495	.400	.205

The values of the coefficient $\frac{e^{fn\pi} + 1}{e^{fn\pi} - 1}$, corresponding to the above values of f, for one up to six half-laps of the rope on the driving-drum or sheaves, are as follows:

f	n = Number of Half-laps on Driving-wheel.					
	1	2	3	4	5	6
.070	9.130	4.623	3.141	2.418	1.999	1.729
.085	7.536	3.833	2.629	2.047	1.714	1.505
.120	5.345	2.777	1.953	1.570	1.358	1.232
.140	4.623	2.418	1.729	1.416	1.249	1.154
.170	3.833	2.047	1.505	1.268	1.149	1.085
.205	3.212	1.762	1.338	1.165	1.083	1.043
.235	2.831	1.592	1.245	1.110	1.051	1.024
.400	1.795	1.176	1.047	1.013	1.004	1.001
.495	1.538	1.093	1.019	1.004	1.001	....

The importance of keeping the rope dry is evident from these figures.

When the rope is at rest the tension is distributed equally on the two lines of the rope, but when running there will be a difference in the tensions of the taut and slack lines equal to the resistance, and the values of T and S may be readily computed from the foregoing formulæ.

Taper Ropes of Uniform Tensile Strength.—Prof. A. S. Herschel in *The Engineer*, April, 1880, p. 267, gives an elaborate mathematical investigation of the problem of making a taper hoisting-rope of uniform tensile strength at every point in its length. Mr. Charles D. West, commenting on Prof. Herschel's paper, gives a similar solution, and derives therefrom the following formula, based on a breaking strain of 80,000 lbs. per sq. in. of the rope, core included, with a factor of safety of 10:

$$F = 3680[\log G - \log g]; \quad \log G = \frac{F}{3680} + \log g;$$

in which F = length in fathoms, and G and g the girth in inches at any two *sections* F fathoms apart.

AMPLE.—Let it be required to find the dimensions of a steel-wire rope to 6720 lbs.—cage, trams, and coal—from a depth of 400 fathoms.
ea of section at lower end = 6720 ÷ 8000 = .84 sq. in.; therefore girth = n. at bottom.

$$\text{Log } G = 400 \div 3680 + \log 3.25 = .10869 + .51188 = .62057;$$

efore $G = 4.174$, or, say, 4 3/16 in. girth at top.
e equations show that the true form of rope is not a regular taper or cated cone, but follows a logarithmic curve, the girth rapidly increas-owards the upper end.

ative Effect of Various-sized Sheaves or Drums on the Life of Wire Ropes.

(Thos. E. Hughes, *Coll'y Eng.*, April, 1893.)

CAST-STEEL ROPES FOR INCLINES.

Made of 6 strands, of 7 wires each, laid around a hemp core.

n. of e in hes.	Diameters of Sheaves or Drums in feet, showing percentages of life for various diameters.						
	100%.	90%.	80%.	75%.	60%.	50%.	25%.
[illegible]	16	14	12	11	9	7	4.75
[illegible]	14	12	10	8.5	7	6	4.5
[illegible]	12	10	8	7.25	6.5	5.5	4.25
[illegible]	10	8.5	7.75	7	6	5	4
	8.5	7.75	6.75	6	5	4.5	3.75
[illegible]	7.75	7	6.25	5.75	4.5	3.75	3.25
[illegible]	7	6.25	5.5	5	4.25	3.5	2.75
[illegible]	6	5.25	4.5	4	3.25	3	2.5
[illegible]	5	4.5	4	3.5	2.75	2.25	1.75

e use of iron ropes for inclines has been generally abandoned, steel s being more satisfactory and economical.

CAST-STEEL HOISTING-ROPES.

Made of 6 strands, of 19 wires each, laid around a hemp core.

n. of e in hes.	Diameters of Sheaves or Drums in feet, showing percentages of life for various diameters.						
	100%.	90%.	80%.	75%.	60%.	50%.	25%.
[illegible]	14	12	10	8.5	7	6	4.5
[illegible]	12	10	8	7	6	5.25	4.25
[illegible]	10	8.5	7.5	6.75	5.5	5	4
[illegible]	9	7.5	6.5	6	5	4.5	3.75
	8	7	6	5.5	4.5	4	3.50
[illegible]	7.5	6.75	5.75	5	4.25	3.5	3
[illegible]	5.5	4.5	4	3.75	3.25	3	2.25
[illegible]	4.5	4	3.75	3.25	3	2.5	2
[illegible]	4	3	3	2.75	2.25	2	1.5
[illegible]	3	....	....	2	....	1.5	....

WIRE-ROPE TRANSMISSION.

e following data and formulæ are taken from a paper by Wm. Hewitt, e Trenton Iron Co., 1890. (See also circulars of John A. Roebling's Co., Trenton, N. J.; "Transmission of Power by Wire Ropes," by A. ahl, Van Nostrand's Science Series No. 28; and Reuleaux's Constructor.)

e Section of Wire Rope best suited, under ordinary conditions, e transmission *of power is* composed of 6 strands of 7 wires each, laid *her about a hempen centre.* Ropes of 12 and 19 wires to the strand are *sed. They are more flexible,* and may be applied with advantage un-*nditions which do not allow* the use of large transmission wheels, bu *of high speed. They are* not as well adapted to stand surface wea *r, on account of* the smaller size of the wires.

The Driving-wheels (Fig. 165) are usually of cast iron, made as light as possible consistent with the requisite strength.

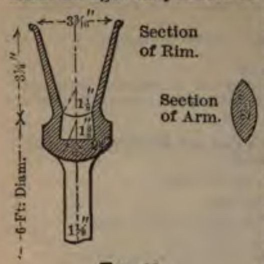


FIG. 165.

materials have been used for filling tom of the groove, such as tarred jute yarn, hard wood, India-rubber and The filling which gives the best sat however, consists of segments of le blocks of India-rubber, soaked in packed alternately in the groove, turned to a true surface.

In long spans, intermediate s wheels are frequently used, and it sufficient to support only the slack ing side of the rope; but whatever th that the power is transmitted, the dr of the rope will require a less numb ports than the slack side. The she porting the driving side, however, all cases be of equal diameter with th wheels. With the slack side small may be used, but their diameter should not be less than one half t driving-sheaves.

The system of carrying sheaves may generally be replaced to a by that of intermediate stations. The rope thus, instead of ru whole length of the transmission, runs only from one station to the it is advisable to make the stations equidistant, so that a rope m on hand, ready spliced, to put on the wheels of any span, shoul give out. This method is to be preferred where there is sometimes motion to the rope, as it prevents sudden movements of this kind ing transmitted over the entire line.

Gross horse-power transmitted $= N_0 = .0003702D^2v\left(k - \frac{ED}{18R}\right)$.

D = diameter of rope in inches (= 9 times diameter of single velocity of rope in feet per second; k = safe stress per square inc = for iron 25,700 lbs.; E = modulus of elasticity = 28,500,000 for radius of driving-wheels in inches. The term $\frac{ED}{18R}$ = the stress p inch due to bending of wires around sheaves.

Loss due to centrifugal force $= N_1 = .0000424D^2v^3$;

Loss due to journal friction of driving-wheels $= N_2 = .0000045($163

" " " " " " intermediate-wheels $= .0000045(W$

in which W = total weight of rope; w = weight of wheel and axl

Net horse-power transmitted,

$$N = N_0 - N_1 - N_2 = D^2v\left[.0003675\left(k - \frac{ED}{18R}\right) - .0000424v\right] -$$

For a maximum value of N the diameter of the wheels should mately from 185 to 192 times the diameter of the rope, and for ratio of diameters an approximate formula for the actual h transmitted is $N = 3.0148D^2V$, in which V = number of rev wheels per minute.

The proper deflections when the rope is at rest are obtained fr mula Deflection = .00005765 span², and are as follows:

Span in feet..	50	100	150	200	250	300	350	
Deflection....	1¾"	7"	1' 3⅝"	2' 3⅝"	3' 7¼"	5' 2¼"	7' ⅝"	9'

It has been found in practice that when the deflection of the is less than 3 inches the transmission cannot be effected with s and shafting or belting is to be preferred. This deflection corre span of about 54 feet. It is customary to make the under side c the driving side. The maximum limit of span is determined by mum deflection that may be given to the upper side of the rop motion. Assuming that the clearance between the upper and low the rope should not be less than two feet, and that the wheels ar feet in diameter, we have a maximum deflection of the upper s which corresponds to a span of about 370 feet.

Much greater spans than this are practicable, in cases whe of the [illegible] is such that the upper side of the rope wer, as in crossing gullies or valleys, and there is nothing to interfere with aining the proper deflections. Some very long transmissions of power e been effected in this way without an intervening support. There is one Lockport, N. Y., for instance, with a clear span of about 1700 feet.

n a later circular of the Trenton Iron Co. (1892) the above figures are newhat modified, giving lower values for the power transmitted by a given e, as follows:

The proper ratio between the diameters of rope and sheaves is that which l permit the maximum working tension to be obtained without overstrain- the wires in bending. For rope of 7-wire strands this ratio is about 150; for rope of 12-wire strands, 1 : 115; and for rope of 19-wire strands, 90; which gives the following minimum diameter of sheaves, in inches, responding to maximum efficiency.

am. rope, in inches.	¼	5/16	⅜	7/16	½	9/16	⅝	11/16	¾	⅞	1	1⅛
wire strands..	37	47	56	66	75	84	94	103	112	...	...	.. ..
" "	..	36	43	50	57	65	72	78	86	101	115	.. ..
" "	.	...	34	39	45	51	56	62	68	79	90	101

Assuming the sheaves are of equal diameter, and not smaller than con- tent with maximum efficiency as determined by the preceding table, the tual horse-power transmitted approximately equals 3.1 times the square the diameter of the rope in inches multiplied by the velocity in feet per cond.

From this rule we deduce the following:

Horse-power of Wire-rope Transmission.

locity, in feet per second.	20	30	40	50	60	70	80
iam. Rope, in inches.	Horse-power Transmitted.						
1/4	4	6	8	10	12	14	16
5/16	6	9	12	15	18	21	24
3/8	9	13	17	22	26	31	35
7/16	12	18	24	30	36	42	47
1/2	16	23	31	39	47	54	62
9/16	20	29	39	49	59	69	78
5/8	24	36	48	61	73	85	97
11/16	29	44	59	73	88	103	117
3/4	35	52	70	87	105	122	140
7/8	48	71	95	119	142	166	190
1	62	93	124	155	186	217	248

he proper deflection to give the rope in order to secure the necessary sion is

$$h = .0000695S^2.$$

the deflection with the rope at rest, and S = the span, both in feet.

urability of Wire Ropes.—At the Risdon Iron Works, San Fran- co, a steel wire rope 2¼ inches in circumference running over 10-foot aves at 5000 ft. per minute has transmitted 40 H.P. for six years without ewing the rope. At the wire-mills a steel wire rope 2¼ in. in circumfer- e running over 8-foot sheaves has been running steadily for a period of ee years at a velocity of 4500 ft. per minute, transmitting 80 H.P.

n Inclined Transmissions, when the angle of inclination is at, the proper deflections cannot be readily determined, and the rope be- nes more sensitive to the ordinary variations in the deflections, so that htening sheaves must be resorted to for producing the requisite tension, in the case of very short spans. When the horizontal distance between two wheels *is less than 60 ft.*, or when the angle of inclination exceeds *o 45 degrees, it will be found* desirable to use tightening sheaves. *ghtening pulleys should be placed* on the slack side of the rope.

he Wire-rope Catenary. (From an article on *smission, by M. Arthur* Achard, Proc. Inst. M. E., Jan *have to bear two distinct* molecular strains: First, t

well as upon its lineal weight, that the tension to which it
pends. By fixing the weight of the rope and its length, the
two spans assume in common, when at rest, is determined,
their common tension; which latter must be such as to pr
the two unequal tensions, T and t, necessary for the tra
power. The driving force $= T - t$.

Moreover, the tension in either span is not the same thro
length; it is a minimum at the lowest point of the curve and
ing towards the two extremities. The calculation of the te
est point is very complicated if based upon the true form
but by substituting a parabola for the catenary, which is al
all cases, the calculation becomes simple. If the two p
same level, the lowest point is midway between them, an
this point is $S_0 = \frac{pl^2}{8h}$, p being the lineal weight, or pounds
rope, l its horizontal projection, which is approximately equ
between the centres of the pulleys, and h the deflection in
catenary possesses the remarkable mechanical property th
between the tensions at any two points is equal to the weig
rope corresponding to the difference in level between the
tensions therefore at the two ends will be $S_1 = S_0 + ph$
substituting for S_1 in the above equation the required valu
solving it with relation to h, the deflections h_1 and h_2 o
trailing spans will be obtained. The deflection h_0, common
at rest, is given by the equation $h_0 = \sqrt{1/2h_1^2 + 1/2h_2^2}$. If
area of the iron portion of the rope, and S the unit strain wh
tension T produces on it, we have $wS = T = \frac{pl^2}{8h_1} + ph_1$. Ta
area w of the rope in square inches, and its weight p in pou
the ratio $w \div p$ differs little from a mean value of 0.24.
working tension usually assigned for iron-wire ropes is S
square inch. Hence $ws \div p = 0.24 \times 14{,}220 = 3410$; and we
imate equation $\frac{l^2}{8h_1} + h_1 = 3410$, which is useful as giving a
the length l and deflection h_1 for the driving-span of a rop
leather, $w \div p = 2.53$ approximately, and it is impossible t
value than about 355 lbs. per square inch; the relation o
$\frac{l^2}{8h_1} + h_1 = 900$, which with equal deflections would give mu

xtremities; the tension at the upper end of each exceeds that at the r by the quantity pH, H being the difference in level between the two mities, or, which is approximately the same, between the centres of wo pulleys. It is evidently the tension of the driving-span at its lower which must be regulated so as to obtain the proper driving tension T he transmission; so that there is a certain excess of tension at the r pulley. Large diameter of pulleys tends to preserve the ropes, makes ffect of stiffness insignificant, and diminishes the effect of friction on earings.

other formula for the tension at the ends of a catenary (assuming it to parabola) is $S_1 = \frac{W}{2h}\sqrt{(\frac{1}{2}l)^2 + (2h)^2}$, in which S = the tension in lbs ; weight of the rope in lbs.; l = span, and h = deflection, in feet.

iameter and Weight of Pulleys for Wire Rope, Ordi-y:

Diameter, ft.	18	14.9	12.4	7.0
Single groove, lbs...	6232	5180	2425	798
Double groove, lbs..	8267	6988	4078	1164

Table of Transmission of Power by Wire Ropes.

(J. A. Roebling's Sons Co., 1886.)

in feet.	Number of Revolutions.	Trade No. of Rope.	Diameter of Rope.	Horse-power.	Diameter of Wheel in feet.	Number of Revolutions.	Trade No. of Rope.	Diameter of Rope.	Horse-power.
	80	23	3/8	3	7	140	20	9/16	35
	100	23	3/8	3½	8	80	19	5/8	26
	120	23	3/8	4	8	100	19	5/8	32
	140	23	3/8	4½	8	120	19	5/8	39
	80	23	3/8	4	8	140	19	5/8	45
	100	23	3/8	5	9	80	20 19	9/16 5/8	47 48
	120	23	3/8	6	9	100	20 19	9/16 5/8	58 60
	140	23	3/8	7	9	120	20 19	9/16 5/8	69 73
	80	22	7/16	9	9	140	20 19	9/16 5/8	82 84
	100	22	7/16	11	10	80	19 18	5/8 11/16	64 68
	120	22	7/16	13	10	100	19 18	5/8 11/16	80 85
	140	22	7/16	15	10	120	19 18	5/8 11/16	96 102
	80	21	½	14	10	140	19 18	5/8 11/16	112 119
	100	21	½	17	12	80	18 17	11/16 ¾	93 99
	120	21	½	20	12	100	18 17	11/16 ¾	116 124
	140	21	½	23	12	120	18 17	11/16 ¾	140 149
	80	20	9/16	20	12	120	16	7/8	173
	100	20	9/16	25	14	80	8 7	1 1⅛	141 148
	120	20	9/16	30	14	100	8 7	1 1⅛	176 185

ng-distance Transmissions. (From Circular of the Trenton Co., 1892.)—In very long transmissions of power the conditions do not s admit of obtaining the proper tensions required in the ordinary sys-or "flying transmission of power," as it is termed. In other words, to the proper conditions, it would necessitate numerous and expensive mediate stations. In case, for instance, it is desired to utilize the power urbine to drive a factory, say a mile away, the best method is to em-a larger rope than would ordinarily be used, running it at a moderate speed. The rope may be in one continuous length, supported, at inte of about 100 ft., on sheaves of comparatively small diameter, since greater rigidity of these ropes preserves them from undue bending str Where sharp angles occur in the line, however, sheaves must be used size corresponding to the safe limit of tension due to bending. The ro run under a high working tension, far in excess of what in the ordi system would cause the rope to slip on the sheaves. The working te may be four or five times as great as the tension in the slack portion c rope, and in order to prevent slipping, the rope is wrapped several about grooved drums, or a series of sheaves at each end of the line provide for the slack due to the stretch of the rope, one of the shea placed on a slide worked by long-threaded bolts, or, better still, on a riage provided with counterweights, which runs back and forth on a t The latter preserves a uniform tension in the slack portion of the which is very important.

Wire-rope tramways are practically transmissions of power of this in which the load, however, instead of being concentrated at one term is distributed uniformly over the entire line. Cable railways are also t missions of this class. The amount of horse-power transmitted is giv the formula

$$N = [4.755D^2 - .000006\,(W + g + g_2)]v;$$

in which D = diameter of the rope in inches; v = velocity in ft. per se W = weight of the rope; g = weight of the terminal sheaves and axles g_2 = weight of the intermediate sheaves and axles.

ROPE-DRIVING.

The transmission of power by cotton or manila ropes promises to be a formidable competitor with gearing and leather belting for use whe amount of power is large, or the distance between the power and the is comparatively great. The following is condensed from a paper by C W. Hunt, Trans. A. S. M. E., vol. xii. p. 230:

But few accurate data are available, on account of the long peri quired in each experiment, a rope lasting from three to six years. In of the early applications so great a strain was put upon the rope th wear was rapid, and success only came when the work required of the was greatly reduced. The strain upon the rope has been decreased u is approximately known what it should be to secure reasonable dura Installations which have been successful, as well as those in which the of the rope was destructive, indicate that 200 lbs. on a rope one inch in eter is a safe and economical working strain. When the strain is mat increased, the wear is rapid.

In the following equations

C = circumference of rope in inches; g = gravity;
D = sag of the rope in inches; H = horse-power;
F = centrifugal force in pounds; L = distance between pulley
P = pounds per foot of rope; w = working strain in poun
R = force in pounds doing useful work;
S = strain in pounds on the rope at the pulley;
T = tension in pounds of driving side of the rope;
t = tension in pounds on slack side of the rope;
v = velocity of the rope in feet per second;
W = ultimate breaking strain in pounds.

$$W = 720C^2; \quad P = .032C^2; \quad w = 20C^2.$$

This makes the normal working strain equal to 1/36 of the br strength, and about 1/25 of the strength at the splice. The actual stra ordinarily much greater, owing to the vibrations in running, as well a imperfectly adjusted tension mechanism.

For this investigation we assume that the strain on the driving si rope is equal to 200 lbs. on a rope one inch in diameter, and an equ strain for other sizes, and that the rope is in motion at various veloci from 10 to 140 ft. per second.

The centrifugal force of the rope in running over the pulley wil

ount of force available for the transmission of power. The centrifu-
rce $F = Pv^2 \div g$.
speed of about 80 ft. per second, the centrifugal force increases faster he power from increased velocity of the rope, and at about 140 ft. per d equals the assumed allowable tension of the rope. Computing this at various speeds and then subtracting it from the assumed maximum n, we have the force available for the transmission of power. The of this force cannot be used, because a certain amount of tension on ack side of the rope is needed to give adhesion to the pulley. What n should be given to the rope for this purpose is uncertain, as there experiments which give accurate data. It is known from considerable ience that when the rope runs in a groove whose sides are inclined d each other at an angle of 45° there is sufficient adhesion when the of the tensions $T \div t = 2$.
the present purpose, T can be divided into three parts: 1. Tension useful work; 2. Tension from centrifugal force; 3. Tension to balance rain for adhesion.
tension t can be divided into two parts: 1. Tension for adhesion; sion from centrifugal force.
s evident, however, that the tension required to do a given work should e materially exceeded during the life of the rope.
re are two methods of putting ropes on the pulleys; one in which the are single and spliced on, being made very taut at first, and less so as pe lengthens, stretching until it slips, when it is respliced. The other od is to wind a single rope over the pulley as many turns as needed to n the necessary horse-power and put a tension pulley to give the neces- adhesion and also take up the wear. The tension t required to trans- he normal horse-power for the ordinary speeds and sizes of rope is com- l by formula (1), below. The total tension T on the driving side of the is assumed to be the same at all speeds. The centrifugal force, as well amount equal to the tension for adhesion on the slack side of the rope, be taken from the total tension T to ascertain the amount of force able for the transmission of power.
s assumed that the tension on the slack side necessary for giving ion is equal to one half the force doing useful work on the driving side rope; hence the force for useful work is $R = \frac{2(T - F)}{3}$; and the ten- n the slack side to give the required adhesion is $\frac{1}{3}(T - F)$. Hence

$$t = \frac{(T - F)}{3} + F. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

sum of the tensions T and t is not the same at different speeds, as the ion (1) indicates.
F varies as the square of the velocity, there is, with an increasing of the rope, a decreasing useful force, and an increasing total tension, he slack side.
h these assumptions of allowable strains the horse-power will be

$$H = \frac{2v(T - F)}{3 \times 550}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

nsmission ropes are usually from 1 to 1¾ inches in diameter. A com- on of the horse-power for four sizes at various speeds and under ry conditions, based on a maximum strain equivalent to 200 lbs. for a ne inch in diameter, is given in Fig. 166. The horse-power of other s readily obtained from these. The maximum power is transmitted, the assumed conditions, at a speed of about 80 feet per second.
wear of the rope is both internal and external; the internal is caused movement of the fibres on each other, under pressure in bending he sheaves, and the external is caused by the slipping and the wedg- the grooves of the pulley. Both of these causes of wear are, within its of ordinary practice, assumed to be directly proportional to the Hence, if we assume the coefficient of the wear to be k, the wear kv, in which the wear increases directly as the velocity, but the power that can be transmitted, as equation (2) shows, will not vary at me rate.
rope is supposed *to have the strain* T constant at all speeds on the *side, and in direct proportion* to the area of the cross-section; hence

the catenary of the driving side is not affected by the speed or by the diam eter of the rope.

The deflection of the rope between the pulleys on the slack side varie with each change of the load or change of the speed, as the tension equatio (1) indicates.

The deflection of the rope is computed for the assumed value of T and

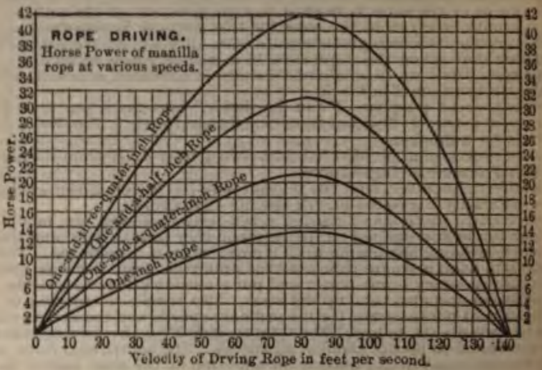


FIG. 166.

by the parabolic formula $S = \frac{PL^2}{8D} + PD$, S being the assumed strain T on the driving side, and t, calculated by equation (1), on the slack side. The tension t varies with the speed.

Horse-power of Transmission Rope at Various Speeds.

Computed from formula (2), given above.

Diam. of Ropes.	Speed of the Rope in feet per minute.											Smallest Diam. of Pulleys in Inches.
	1500	2000	2500	3000	3500	4000	4500	5000	6000	7000	8000	
1/2	1.45	1.9	2.3	2.7	3	3.2	3.4	3.4	3.1	2.2	0	20
5/8	2.3	3.2	3.6	4.2	4.6	5.0	5.3	5.3	4.9	3.4	0	24
3/4	3.3	4.3	5.2	5.8	6.7	7.2	7.7	7.7	7.1	4.9	0	30
7/8	4.5	5.9	7.0	8.2	9.1	9.8	10.8	10.8	9.3	6.9	0	36
1	5.8	7.7	9.2	10.7	11.9	12.8	13.6	13.7	12.5	8.8	0	42
1 1/4	9.2	12.1	14.3	16.8	18.6	20.0	21.2	21.4	19.5	13.8	0	54
1 1/2	13.1	17.4	20.7	23.1	26.8	28.8	30.6	30.8	28.2	19.8	0	60
1 3/4	18	23.7	28.2	32.8	36.4	39.2	41.5	41.8	37.4	27.6	0	72
2	23.2	30.8	36.8	42.8	47.6	51.2	54.4	54 8	50	35.2	0	84

The following notes are from the circular of the C. W. Hunt Co., New York:

For a temporary installation, when the rope is not to be long in use, it might be advisable to increase the work to double that given in the table.

For convenience in estimating the necessary clearance on the driving and on the slack sides, we insert a table showing the sag of the rope at different speeds when transmitting the horse-power given in the preceding table. When at rest the sag is not the same as when running, being greater on the driving and less on the slack sides of the rope. The sag of the driving side when transmitting the normal horse-power is the same no matter what size of rope is used or what the speed driven at, because the assumption is that the strain on the rope shall be the same at all speeds when transmitting the

rce available for the transmission of power. The centrifu-
$v^2 \div g$.
bout 80 ft. per second, the centrifugal force increases faster
rom increased velocity of the rope, and at about 140 ft. per
e assumed allowable tension of the rope. Computing this
speeds and then subtracting it from the assumed maximum
the force available for the transmission of power. The
ce cannot be used, because a certain amount of tension on
the rope is needed to give adhesion to the pulley. What
e given to the rope for this purpose is uncertain, as there
its which give accurate data. It is known from considerable
when the rope runs in a groove whose sides are inclined
er at an angle of 45° there is sufficient adhesion when the
ions $T \div t = 2$.
t purpose, T can be divided into three parts: 1. Tension
k; 2. Tension from centrifugal force; 3. Tension to balance
hesion.
can be divided into two parts: 1. Tension for adhesion;
centrifugal force.
owever, that the tension required to do a given work should
y exceeded during the life of the rope.
methods of putting ropes on the pulleys; one in which the
and spliced on, being made very taut at first, and less so as
ns, stretching until it slips, when it is respliced. The other
d a single rope over the pulley as many turns as needed to
sary horse-power and put a tension pulley to give the neces-
d also take up the wear. The tension t required to trans-
horse-power for the ordinary speeds and sizes of rope is com-
a (1), below. The total tension T on the driving side of the
to be the same at all speeds. The centrifugal force, as well
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that the tension on the slack side necessary for giving
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$$t = \frac{(T-F)}{3} + F. \qquad (1)$$

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$$H = \frac{2v(T-F)}{3 \times 550}. \qquad (2)$$

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horse-power for four sizes at various speeds and under
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If we assume the coefficient of the wear to be k, the wear
hich the wear increases directly as the velocity, but the
can be transmitted, as equation (2) shows, will not vary at

sed to have the strain T constant at all speeds on th
direct proportion to the area of the cross-section; henc

For large amounts of power it is common to use a number of ro[illegible] side by side in grooves, each spliced separately. For lighter dri[illegible] engineers use one rope wrapped as many times around the pulle[illegible] necessary to get the horse-power required, with a tension pulley to [illegible] the slack as the rope wears when first put in use. The weight put [illegible] tension pulley should be carefully adjusted, as the overstraining of [illegible] from this cause is one of the most common errors in rope drivi[illegible] therefore give a table showing the proper strain on the rope for the [illegible] sizes, from which the tension weight to transmit the horse-power [illegible] tables is easily deduced. This strain can be still further reduced [illegible] horse-power transmitted is usually less than the nominal work [illegible] rope was proportioned to do, or if the angle of groove is the [illegible] acute.

DIAMETER OF PULLEYS AND WEIGHT OF ROPE.

Diameter of Rope, in inches.	Smallest Diameter of Pulleys, in inches.	Length of Rope to allow for Splicing, in feet.	Approxim[illegible] Weight, [illegible] foot of rop[illegible]
½	20	6	[illegible]
⅝	24	6	[illegible]
¾	30	7	[illegible]
⅞	36	8	[illegible]
1	42	9	[illegible]
1¼	54	10	[illegible]
1½	60	12	[illegible]
1¾	72	13	[illegible]
2	84	14	[illegible]

With a given velocity of the driving-rope, the weight of rope requ[illegible] transmitting a given horse-power is the same, no matter what size [illegible] adopted. The smaller rope will require more parts, but the weight [illegible] the same.

Miscellaneous Notes on Rope-driving.—W. H. Booth [illegible] nicates to the *Amer. Machinist* the following data from English practi[illegible] cotton ropes. The calculated figures are based on a total allowable [illegible] on a 1¾-inch rope of 600 lbs., and an initial tension of 1/10 the total [illegible] stress, which corresponds fairly with practice.

Diameter of rope....................	1¼″	1⅜″	1½″	1⅝″	1¾″	[illegible]
Weight per foot, lbs................	.5	.6	.72	.844	[illegible]	[illegible]
Centrifugal tension = V^2 divided by	64	53	44	38	33	[illegible]
" for $V = 80$ ft. per sec., lbs.	100	121	145	170	[illegible]	[illegible]
Total tension allowable.............	300	360	430	500	600	[illegible]
Initial tension.....................	30	36	43	50	60	[illegible]
Net working tension at 80 ft. velocity	170	203	242	280	347	[illegible]
Horse-power per rope " "	24	28	34	41	49	[illegible]

The most usual practice in Lancashire is summed up roughly in the [illegible] lowing figures: 1¾-inch cotton ropes at 5000 ft. per minute velocity = [illegible] per rope. The most common sizes of rope now used are 1¾ and 1⅝ [illegible] maximum horse-power for a given rope is obtained at about [illegible] per second. Above that speed the power is reduced by centrifugal [illegible] At a speed of 2500 ft. per minute four ropes will do about the same [illegible] three at 5000 ft. per min.

Cotton ropes do not require much lubrication in the sense that [illegible] quired by ropes made of the rough fibre of manila hemp. Merely a [illegible] surface dressing is all that is required. For small ropes, common [illegible] ning machinery, from ½ to ¾ inch diameter, it is the custom to prev[illegible] fluffing of the ropes on the surface by a light application of a mixt[illegible] black-lead and molasses,—but only enough should be used to lay the fi[illegible] put upon one of the pulleys in a series of light dabs.

Reuleaux's Constructor gives as the "specific capacity" of hemp r[illegible] actual practice, that is, the horse-power transmitted per square [illegible] cross-section for each foot of linear velocity per minute, [illegible] cross-section being taken as that due to the full outside diamet[illegible] rope. For a 1¾-in. rope, with a cross-section of 2.405 sq. in., at [illegible] [illegible]00 ft. per min., this gives a horse-power of from 24 to [illegible] by Mr. Hunt's table and 49 by Mr. Booth's.

s formulæ for calculating sources of loss in hemp-rope to (1) journal friction, (2) stiffness of ropes, and (3) creep onstants in these formulæ are, however, uncertain from ental data. He calculates an average case giving loss of rnal friction = 4%, to stiffness 7.8%, and to creep 5%, or 16.8% his is not to be considered higher than the actual loss.

(in *Eng'g News*, Dec. 6, 1890) says: In England hemp and ve been largely superseded by ropes of cotton; and I am reason for this is that dry manila ropes wear out too fast, ropes give too low a coefficient of friction. The angle of e has been in use for 33 years, having been first introduced Belfast, Ireland; but if we are to use tallow-laid or other , we should certainly use a sharper angle in the groove, American system, which employs a continuous rope with

mula, Tension of driving side of rope ÷ tension of slack , implies a coefficient of friction of only .10. But I have cient of friction of .26, and have found one authority giving dvises for single-line transmission 30° angle of groove. English engineer, and Yale & Towne use a 30° groove in eels of travelling-cranes, and I hope to see the best Ameri- 30° or 35° as a standard groove angle. The work done in asy manila rope from a 30° groove is not worth considera- e hear a great deal about the loss of power on this account. favor of using the continuous-rope system, and also of pes than are recommended in Mr. Hunt's paper.

ect small transmission I have ever seen (about 20 H.P.) em- nila rope on wheels 30 in. in diameter, using a tension car- ian use large ropes I think it wiser to replace small ones doing a great gain may be made in efficiency, thus saving

ity of failures in the continuous-rope plan have occurred g and driven sheaves were of widely different diameters, as ving dynamos, or driving a line-shaft from an engine fly- arily installed the ropes will not pull alike, and by calcula- ment we may find one rope pulling twice or three times as ers on the sheave.

designed by the writer employs an engine-driving sheave s the diameter of the driven sheave. To equalize the pull ropes the grooves of the large driving-sheaves were made 60° and those of the small sheaves with an angle of 45°. roove angle has entirely remedied the unequal pulling com-

erved that in sheaves of the same diameter, by the use of a eight, the ropes may all pull alike; while, where the sheaves iameter, the pull is unequal. The only difference of condi- ases lies in the different arc of contact of the rope on the ch leads to a greater frictional hold of the rope on the large lize the frictional hold on the two sheaves we may sharpen small sheave or increase the angle of the large sheave.

g. Co. adopts a curved form of groove instead of one with lined to each other at 45°. The curves are concave to the rests on the sides of the groove in driving and driven pul- lleys the rope rests on the bottom of the groove, which is Walker Mfg. Co. also uses a "differential" drum for heavy which the grooves are contained each in a separate ring slide on the turned surface of the drum in case one rope another.

drive on the separate, or English, rope system is described *Power*, April, 1892. It is in use at the India Mill at Darwen, ll was originally driven by gears, but did not prove success- iving was resorted to. The 85,000 spindles and preparation 2000 horse-power tandem compound engine, with cylinders in diameter and 72-inch stroke, running at 54 revolutions fly-wheel is 30 feet in diameter, weighs 65 tons, and is grooves for 1¾-inch ropes. These ropes lead off to receiv- e several floors, so that each floor receives its power direct. The speed of the ropes is 5089 feet per minute, and five used, the number of ropes upon each being proportioned to the amount of power required upon the several floors. Lambeth ropes are used. (For much other information on this subject see " Driving," by J. J. Flather, John Wiley & Sons, 1895.)

FRICTION AND LUBRICATION.

Friction is defined by Rankine as that force which acts between bodies at their surface of contact so as to resist their sliding on each and which depends on the force with which the bodies are pressed tog

Coefficient of Friction.—The ratio of the force required to body along a horizontal plane surface to the weight of the body is coefficient of friction. It is equivalent to the tangent of the angle of which is the angle of inclination to the horizontal of an inclined which the body will just overcome its tendency to slide. The angle is denoted by θ, and the coefficient by f. $f = \tan \theta$.

Friction of Rest and of Motion.—The force required to body sliding is called the friction of rest, and the force required to its sliding after having started is called the friction of motion.

Rolling Friction is the force required to roll a cylindrical or cal body on a plane or on a curved surface. It depends on the nature surfaces and on the force with which they are pressed together, essentially different from ordinary, or sliding, friction.

Friction of Solids.—Rennie's experiments (1829) on friction of usually unlubricated and dry, led to the following conclusions:

1. The laws of sliding friction differ with the character of the rubbing together.
2. The friction of fibrous material is increased by increased ext surface and by time of contact, and is diminished by pressure and
3. With wood, metal, and stones, within the limit of abrasion, varies only with the pressure, and is independent of the extent of time of contact and velocity.
4. The limit of abrasion is determined by the hardness of the softer two rubbing parts.
5. Friction is greatest with soft and least with hard materials.
6. The friction of lubricated surfaces is determined by the nature lubricant rather than by that of the solids themselves.

Friction of Rest. (Rennie.)

Pressure, lbs. per square inch.	Values of f.			
	Wrought iron on Wrought Iron.	Wrought on Cast Iron.	Steel on Cast Iron.	Bra[illegible] Cas[illegible]
187	.25	.28	.30	[illegible]
224	.27	.29	.33	[illegible]
336	.31	.33	.35	[illegible]
448	.38	.37	.35	[illegible]
560	.41	.37	.36	[illegible]
672	Abraded	.38	.40	[illegible]
784	"	Abraded	Abraded	[illegible]

Law of Unlubricated Friction.—A. M. Wellington, *Eng'g* April 7, 1888, states that the most important and the best determined the laws of unlubricated friction may be thus expressed:

The coefficient of unlubricated friction decreases materially with is very much greater at minute velocities of 0 +, falls very rapid minute increases of such velocities, and continues to fall much less with higher velocities up to a certain varying point, following *laws which obtain with lubricated friction.*

Friction of Steel Tires Sliding on Steel Rails. (W*house* & Galton.)

Speed, miles per hour..........	10	15	25	[illegible]
Coefficient of friction..........	0.110	.087	[illegible]	[illegible]
Adhesion, lbs. per ton (2240 lbs.)	246	195	179	[illegible]

'riction is a consequence of the irregularities of form and of surface of bodies rolling one over the other. Its laws finitely established in consequence of the uncertainty which iment as to how much of the resistance is due to roughness of uch to original and permanent irregularity of form, and how tion under the load. (Thurston.)

ts of Rolling Friction.—If R = resistance applied at nce of the wheel, W = total weight, r = radius of the wheel, fficient, $R = fW \div r$. f is very variable. Coulomb gives .06 for metal, where W is in pounds and r in feet. Tredgold e of f for iron on iron .002.

on soft soil Morin found f = .065, and on hard smooth roads

of the Society of Arts (Clark, R. T. D.) reported a loaded ibit a resistance on various loads as below:

ent	Speed per hour.	Coefficient.	Resistance.
....	2.87 miles.	.007	17.41 per ton.
....	3.56 "	.0121	27.14 "
....	3.34 "	.0185	41.60 "
velled.....	3.45 "	.0199	44.48 "
ite, new..	3.51 "	.0451	101.09 "

es the value of f for ordinary railroads, .003, well-laid railroad st possible railroad track, .001.

eriments that have been made upon the coefficients of rolling from axle friction, are too incomplete to serve as a basis for . (Trautwine).

'luid Friction.—For all fluids, whether liquid or gaseous, is (1) independent of the pressure between the masses in rectly proportional to the area of rubbing-surface; (3) pro- e square of the relative velocity at moderate and high speeds, city nearly at low speeds; (4) independent of the nature of the solid against which the stream may flow, but dependent upon their degree of roughness; (5) proportional to the den- l, and related in some way to its viscosity. (Thurston.)

of Lubricated Surfaces approximates to that of solid fric- rnal is run dry, and to that of fluid friction as it is flooded

Repose and Coefficients of Friction of Build- Materials. (From Rankine's Applied Mechanics.)

	θ.	$f = \tan\theta$.	$\frac{1}{\tan\theta}$.
nd brickwork..	31° to 35°	.6 to .7	1.67 to 1.4
brickwork with r	36½°	.74	1.35
ie...............	22°	about .4	2.5
....	35° to 16⅔°	.7 to .3	1.43 to 3.3
ber.	26½° to 11⅓°	.5 to .2	2 to 5
als.	31° to 11⅓°	.6 to .2	1.67 to 5
als	14° to 8½°	.25 to .15	4 to 6.67
y clay.........	27°	.51	1.96
oist clay........	18¼°	.33	3.
....	14° to 45°	.25 to 1.0	4 to 1
dry sand, clay, rth.....	21° to 37°	.38 to .75	2.63 to 1.33
, damp clay....	45°	1.0	1
wet clay......	17°	.31	3.23
shingle and	39° to 48°	.81	1.23 to 0.9

f Motion.—*The following is a table of the angle of re- of friction $f = \tan\theta$, and its reciprocal, $1 \div f$, for th ism—condensed from the tables of General Morin , as given by Rankine:*

No.	Surfaces.	θ.	f.
1	Wood on wood, dry	14° to 26½°	.25 to .5
2	" " " soaped. .	11½° to 2°	.2 to .04
3	Metals on oak, dry	26½° to 31°	.5 to .6
4	" " " wet.	13½° to 14°	.24 to .26
5	" " " soapy. . .	11½°	.2
6	" " elm, dry	11½° to 14°	.2 to .25
7	Hemp on oak, dry.	28°	.53
8	" " " wet.	18½°	.33
9	Leather on oak.	15° to 19½°	.27 to .38
10	" " metals, dry. .	29½°	.56
11	" " " wet. .	20°	.36
12	" " " greasy	13°	.23
13	" " " oily. . .	8½°	.15
14	Metals on metals, dry. . .	8½° to 11°	.15 to .2
15	" " " wet. . .	16½°	.3
16	Smooth surfaces, occasionally greased.	4° to 4½°	.07 to .08
17	Smooth surfaces, continuously greased.	3°	.05
18	Smooth surfaces, best results.	1¾° to 2°	.03 to .036
19	Bronze on lignum vitæ, constantly wet.	3° ?	.05 ?

Coefficients of Friction of Journals. (Morin.)

Material.	Unguent.	Lubrication. Intermittent.
Cast iron on cast iron. . . .	Oil, lard tallow.	.07 to .08
	Unctuous and wet.	.14
Cast iron on bronze.	Oil, lard, tallow.	.07 to .08
	Unctuous and wet.	.16
Cast iron on lignum-vitæ. .	Oil, lard.	. . . .
Wrought iron on cast iron / " " " bronze. .	Oil, lard, tallow.	.07 to .08
Iron on lignum vitæ.	Oil, lard.	.11
	Unctuous.	.19
Bronze on bronze.	Olive-oil.	.10
	Lard.	.09

Prof. Thurston says concerning the above figures that much better [illegible] are probably obtained in good practice with ordinary machinery. [illegible] here given are so greatly modified by variations of speed, pressure, [illegible] perature, that they cannot be taken as correct for general purposes.

Average Coefficients of Friction. Journal of cast iron in [illegible] bearing; velocity 720 feet per minute; temperature 70° F.; inter[illegible] feed through an oil-hole. (Thurston on Friction and Lost Work.)

Oils.	Pressures, pounds per square inch: 8	16	32
Sperm, lard, neat's-foot, etc.	.159 to .250	.138 to .192	.086 to .141
Olive, cotton-seed, rape, etc.	.160 " .283	.107 " .245	.101 " .168
Cod and menhaden.	.248 " .278	.124 " .167	.097 " .102
Mineral lubricating-oils. . . .	.154 " .261	.145 " .222	[illegible] " [illegible]

With fine steel journals running in bronze bearings and continuous [illegible] cation, coefficients far below those above given are obtained. [illegible] [illegible] the coefficient with 50 lbs. per square inch pressure w[illegible] .0051; with 300 lbs., .0057.

ressures, as in spindles, the coefficients are much higher. ury found, at a temperature of 100° and a velocity of 600

, lbs. per sq. in......	1	2	3	4	5
t......	.38	.27	.22	.18	.17

fficients, however, and the great decrease in the coefficient sures are limited as a practical matter only to the smaller exist especially in spinning machinery, where the pressure film of oil so thick that the viscosity of the oil is an import- otal frictional resistance.

ts on Friction of a Journal Lubricated by an rted by the Committee on Friction, Proc. Inst. M. E., that the absolute friction, that is, the absolute tangential inch of bearing, required to resist the tendency of the brass the journal, is nearly a constant under all loads, within or- imits. Most certainly it does not increase in direct propor- as it should do according to the ordinary theory of solid ults of these experiments seem to show that the friction of ated journal follows the laws of liquid friction much more se of solid friction. They show that under these circum- ion is nearly independent of the pressure per square inch, ses with the velocity, though at a rate not nearly so rapid the velocity.

nts on friction at different temperatures indicate a great e friction as the temperature rises. Thus in the case of speed of 450 revolutions per minute, the coefficient of fric- ature of 120° is only one third of what it was at a tempera-

s of steel, 4 inches diameter and 6 inches long, and a gun- bracing somewhat less than half the circumference of the n its upeer side, on which the load was applied. When the ournal was immersed in oil, and the oil therefore carried by rotation of the journal, the greatest load carried with bs. per square inch, and with mineral oil 625 lbs.

with ordinary lubrication, the oil being fed in at the cen- the brass, and a distributing groove being cut in the brass is of the journal, the bearing would not run cool with only e inch, the oil being pressed out from the bearing-surface oil-hole, instead of being carried in by it. On introducing es through two parallel grooves, the lubrication appeared , but the bearing seized with 380 lbs. per square inch.

as introduced through two oil-holes, one near each end of ach connected with a curved groove, the brass refused to n cool, and seized with a load of only 200 lbs. per square

l under the journal feeding rape-oil, the bearing fairly car- . Tower's conclusion from these experiments is that the on the quantity and uniformity of distribution of the oil, hing between the oil-bath results and seizing, according to imperfection of the lubrication. The lubrication may be g a coefficient of 1/100; but it appeared as though it could d and the friction increased much beyond this point with- k of heating and seizing. The oil-bath probably represents lubrication possible, and the limit beyond which friction d by lubrication; and the experiments show that with speeds 0 feet per minute, by properly proportioning the bearing- d, it is possible to reduce the coefficient of friction to as low fficient of 1/1500 is easily attainable, and probably is fre- , in ordinary engine-bearings in which the direction of the lternating and the oil given an opportunity to get between le the duration of the force in one direction is not sufficient the *oil film to be squeezed out.*

n the behavior of the apparatus gave reason to believe that *ication the speed of* minimum friction was from 100 to 1*nd that this* speed of minimum friction tends to be *of load*, and also with less perfect lubrication. *friction is meant* that speed in approaching whi *inishes*, and above which the friction increases

Coefficients of Friction of Journal with Oil-bat[h] strract of results of Tower's experiments on friction (Proc. Inst. M. [E.] 1883). Journal, 4 in. diam., 6 in. long; temperature, 90° F.

Lubricant in Bath.	Nominal Load, in pounds per squa[re inch]				
	625	520	415	310	205
	Coefficients of Friction.				
Lard-oil:					
157 ft. per min.	....	.0009	.0012	.0014	.0020
471 " "	....	.0017	.0021	.0029	.0042
Mineral grease:					
157 ft. per min.	.001	.0014	.0016	.0022	.0034
471 " "	.002	.0022	.0027	.004	.0066
Sperm-oil:					
157 ft. per min.	....	seiz'd	.0015	.0011	.0016
471 " "	....	......	.0021	.0019	.0027
Rape-oil:	(573 lb.)				
157 ft. per min.	.001	.001	.0009	.0008	.0014
471 " "	....	.0015	.0016	.0016	.0024
Mineral-oil:					
157 ft. per min.	.0013	.0012	.0012	.0014	.0021
471 " "	....	.0018	.002	.0024	.0035
Rape-oil fed by syphon lubricator:					
157 ft. per min.	....	......	......	.0056	.0098
314 " "	....	......	......	.0068	.0077
Rape-oil, pad under journal:					
157 ft. per min.	....	......	......	.0099	.0105
314 " "	....	......	......	.0099	.0078

Comparative friction of different lubricants under same circum[stances], temperature 90°, oil-bath:

Sperm-oil	100 per cent.	Lard	135
Rape-oil	106 "	Olive-oil	135
Mineral oil	129 "	Mineral grease	217

Coefficients of Friction of Motion and of Re[st of a] Journal.—A cast-iron journal in steel boxes, tested by Prof. Th[urston] a speed of rubbing of 150 feet per minute, with lard and with [sperm] gave the following:

Pressures per sq. in., lbs.	50	100	250	500	75[0]
Coeff., with sperm	.013	.008	.005	.004	.00[illegible]
" " lard	.02	.0137	.0085	.0053	.00[illegible]

The coefficients at starting were:

With sperm	.07	.135	.14	.15	.1[illegible]
With lard	.07	.11	.11	.10	.1[illegible]

The coefficient at a speed of 150 feet per minute decreases with [increase] of pressure until 500 lbs. per sq. in. is reached; above this it incre[ases]. coefficient at rest or at starting increases with the pressure throu[ghout the] range of the tests.

Value of Anti-friction Metals. (Denton.)—The vari[ous] metals available for lining brasses do not afford coefficients [of friction] lower than can be obtained with bare brass, but they are les[s liable to] "overheating," because of the superiority of such material over [brass in] ability to permit of abrasion or crushing, without excessive in[crease of] friction.

Thurston (Friction and Lost Work) says that gun-bronze, Bab[bitt, and] other soft white alloys have substantially the same friction; in ot[her words,] the friction is determined by the nature of the unguent and not [by that of] the rubbing-surfaces, when the latter are in good order. The w[hite metals] run at higher temperatures than the bronze. This, however, do[es not nec]essarily indicate a serious defect, but simply deficient condu[ctivity. The] value of the white alloys for bearings lies mainly in their re[adiness to resume a] [smo]oth surface after any local or general injury by alte[ration of] form.

on for Bearings. (Joshua Rose.)--Cast iron appears to be an to the general rule, that the harder the metal the greater the to wear, because cast iron is softer in its texture and easier to eel tools than steel or wrought iron, but in some situations it is urable than hardened steel; thus when surrounded by steam it etter than will any other metal. Thus, for instance, experience strated that piston-rings of cast iron will wear smoother, better, y as long as those of steel, and longer than those of either on or brass, whether the cylinder in which it works be composed eel, wrought iron, or cast iron; the latter being the more note-ice two surfaces of the same metal do not, as a rule, wear or ogether. So also slide-valves of brass are not found to wear so moothly as those of cast iron, let the metal of which the seating d be whatever it may; while, on the other hand, a cast iron slide-vear longer of itself and cause less wear to its seat, if the latter on, than if of steel, wrought iron, or brass.

n of Metals under Steam-pressure.—The friction of iron under steam-pressure is double that of iron upon iron. ock, Trans. A. S. M. E., i. 151.)

s "Laws of Friction."—1. The friction between two bodies roportioned to the pressure; i.e., the coefficient is constant for s.

efficient and amount of friction, pressure being the same, is in-of the areas in contact.

efficient of friction is independent of velocity, although static ction of rest) is greater than the friction of motion.

es, April 7, 1888, comments on these "laws" as follows: From ut 1876 there was no attempt worth speaking of to enlarge our of the laws of friction, which during all that period was assumed ete, although it was really worse than nothing, since it was for rt wholly false. In the year first mentioned Morin began a se-riments which extended over two or three years, and which the enunciation of these three "fundamental laws of friction," hich is even approximately true.

years these laws were accepted as axiomatic, and were quoted as it question in every scientific work published during that whole w that they are so thoroughly discredited it has been attempted way their defects on the ground that they cover only a very lim-of pressures, areas, velocities, etc., and that Morin himself only them as true within the range of his conditions. It is now clearly that there are no limits or conditions within which any one of pproximates to exactitude, and that there are many conditions they lead to the wildest kind of error, while many of the con-as inaccurate as the laws. For example, in Morin's "Table of of Moving Friction of Smooth Plane Surfaces, perfectly lubri-ch may be found in hundreds of text-books now in use, the coeffi-ught iron on brass is given as .075 to .103, which would make the ion of railway trains 15 to 20 lbs. per ton instead of the 3 to 6 lbs. ually is.

Iorin, in a letter to the Secretary of the Institution of Mechanical lated March 15, 1879, writes as follows concerning his experiments nade more than forty years before: "The results furnished by my s as to the relations between pressure, surface, and speed on the nd sliding friction on the other, have always been regarded by as mathematical laws, but as close approximations to the truth, imits of the data of the experiments themselves. The same holds, on, for many other laws of practical mechanics, such as those of tance, fluid resistance, etc."

. Denton (*Stevens Indicator*, July, 1890) says: It has been gen-ned that friction between lubricated surfaces follows the simple amount of the friction is some fixed fraction of the pressure be-urfaces, such fraction being independent of the intensity of the r square inch and *the velocity* of rubbing, between certain limits *and that the fixed fraction* referred to is represented by the co-*friction given by th*e experiments of Morin or obtained from ex-*ata which represent* conditions of practical lubrication, such as *Webber's Manual of Power.*

eriments of Thurston, Woodbury, Tower, etc., however, it *he friction between* lubricated metallic surfaces, such as ma chine bearings, is not directly proportional to the pressure, is not indep- dent of the speed, and that the coefficients of Morin and Webber are ab tenfold too great for modern journals.

Prof. Denton offers an explanation of this apparent contradiction of a thorities by showing, with laboratory testing-machine data, that M laws hold for bearings lubricated by a restricted feed of lubricant, suc is afforded by the oil-cups common to machinery; whereas the modern e periments have been made with a surplus feed or superabundance of lub cant, such as is provided only in railroad-car journals, and a few spec cases of practice.

That the low coefficients of friction obtained under the latter condit are realized in the case of car journals, is proved by the fact that the te perature of car-boxes remains at 100° at high velocities; and experiment sh that this temperature is consistent only with a coefficient of friction o fraction of one per cent. Deductions from experiments on train resist also indicate the same low degree of friction. But these low co-efficient not account for the internal friction of steam-engines as well as do the efficients of Morin and Webber.

In *American Machinist*, Oct. 23, 1890, Prof. Denton says: Morin's meas ment of friction of lubricated journals did not extend to light press They apply only to the conditions of general shafting and engine work.

He clearly understood that there was a frictional resistance, due sol the viscosity of the oil, and that therefore, for very light pressures, the which he enunciated did not prevail.

He applied his dynamometers to ordinary shaft-journals without sp preparation of the rubbing-surfaces, and without resorting to artif methods of supplying the oil.

Later experimenters have with few exceptions devoted themselves ex sively to the measurement of resistance practically due to viscosity a They have eliminated the resistance to which Morin confined his meas ments, namely, the friction due to such contact of the rubbing-surfac prevail with a very thin film of lubricant between comparatively rough faces.

Prof. Denton also says (Trans. A. S. M. E., x. 518): "I do not believe t is a particle of proof in any investigation of friction ever made, that Mo laws do not hold for ordinary practical oil-cups or restricted rates of fe

Laws of Friction of well-lubricated Journals.— Goodman (Trans. Inst. C. E. 1886, *Eng'g News*, Apr. 7 and 14, 1886), rev ing the results obtained from the testing-machines of Thurston, Tower Stroudley, arrives at the following laws:

LAWS OF FRICTION: WELL-LUBRICATED SURFACES.
(Oil-bath.)

1. The coefficient of friction with the surfaces efficiently lubricated is 1/6 to 1/10 that for dry or scantily lubricated surfaces.

2. The coefficient of friction for moderate pressures and speeds var proximately inversely as the normal pressure; the frictional resistan ries as the area in contact, the normal pressure remaining constant.

3. At very low journal speeds t e coefficient of friction is abnor high; but as the speed of sliding increases from about 10 to 100 ft. per the friction diminishes, and again rises when that speed is exceeded, v approximately as the square root of the speed.

4. The coefficient of friction varies approximately inversely as the te ature, within certain limits, namely, just before abrasion takes place.

The evidence upon which these laws are based is taken from various ern experiments. That relating to Law 1 is derived from the "First R on Friction Experiments," by Mr. Beauchamp Tower.

Method of Lubrication.	Coefficient of Friction.	Compar[ative] Frictio[n]
Oil-bath	.00139	1.[illegible]
Siphon lubricator	.0098	[illegible]
Pad under journal	.0090	[illegible]

With a load of 293 lbs. per sq. in., and a journal speed of [illegible] found the coefficient of friction to be .0016 with an

imes as much, with a pad. The very low coefficients ob-
Tower will be accounted for by Law 2, as he found that the
tance per square inch under varying loads is nearly constant,

r sq. in.....	529	468	415	363	310	258	205	153	100
st. per sq. in.	.416	.514	.498	.472	.464	.438	.43	.458	.45

l resistance per square inch is the product of the coefficient the load per square inch on horizontal sections of the brass. product be a constant, the one factor must vary inversely as high load will give a low coefficient, and *vice versa*.
lubrication, the coefficient is more constant under varying tional resistance then varies directly as the load, as shown by Table VIII of his report (Proc. Inst. M. E. 1883).
t to Law 3, A. M. Wellington (Trans. A. S. C. E. 1884), in ex-ournals revolving at very low velocities, found that the friction great, and nearly constant under varying conditions of the d, and temperature. But as the speed increased the friction regularly, and again returned to the original amount when as reduced to the same rate. This is shown in the following

r minute:							
3.33	4.86	8.82	21.42	35.37	53.01	89.28	106.02
friction:							
.070	.069	.055	.047	.040	.035	.030	.026

ound by Prof. Kimball that when the journal velocity was in-
to 110 ft. per minute, the friction was reduced 70%; in another
n was reduced 67% when the velocity was increased from 1 to
ute; but after that point was reached the coefficient varied
with the square root of the velocity.
g results were obtained by Mr. Tower:

te...	209	262	314	366	419	471	Nominal Load per sq. in.
on..	.0010	.0012	.0013	.0014	.0015	.0017	520 lbs.
	.0013	.0014	.0015	.0017	.0018	.002	468 "
	.0014	.0015	.0017	.0019	.0021	.0024	415 "

n of friction with temperature is approximately in the inverse
Take, for example, Mr. Tower's results, at 262 ft. per minute:

	110°	100°	90°	80°	70°	60°
	.0044	.0051	.006	.0073	.0092	.0119
	.00451	.00518	.00608	.00733	.00964	.01252

s not hold good for pad or siphon lubrication, as then the co-
tion diminishes more rapidly for given increments of tem-
n a gradually decreasing scale, until the normal temperature
hed; this normal temperature increases directly as the load
is is shown in the following table taken from Mr. Stroudley's
ith a pad of rape oil:

...	105°	110°	115°	120°	125°	130°	135°	140°	145°
....	.022	.0180	.0160	.0140	.0125	.0115	.0110	.0106	.0102
eff..	......	.0040	.0020	0020	.0015	.0010	.0005	.0004	.0002

-Westinghouse experiments it was found that with velocities
per min., and with low pressures, the frictional resistance
as the normal pressure; but when a velocity of 100 ft. per
ded, the coefficient of friction greatly diminished; from the
nts Prof. Kennedy found that the coefficient of fric
as sensibly less than for low.
Pressures on Bearing-surfaces. (Proc. I
Committee on Friction experimented with a st

increased, and may be stated approximately as 1/20 at 15
diminishing to 1/30 at 75 lbs. per sq. in.

The high coefficients of friction are explained by the difficu
ing a collar-bearing. It is similar to the slide-block of an en
carry only about one tenth the load per sq. in. that can be
crank-pins.

In experiments on cylindrical journals it has been show
cylindrical journal was lubricated from the side on which th
100 lbs. per sq. in. was the limit of pressure that it would car
came to be lubricated on the lower side and was allowed to
with it, 600 lbs. per sq. in. was reached with impunity; and i
sq. in., which was reckoned upon the full diameter of the b
be reckoned on the sixth part of the circle that was taking
portion of the load, it followed that the pressure upon that p
amounted to about 1200 lbs. per sq. in.

In connection with these experiments Mr. Wicksteed sta
ing-machines the pressure on the collars is frequently as hig
sq. in., but the speed of rubbing in this case is lower than it
the experiments of the Research Committee. In machine
slowly and intermittently, as in testing-machines, very mu
sures are admissible.

Mr. Adamson mentions the case of a heavy upright shaft
small footstep-bearing, where a weight of at least 20 tons w
shaft of 5 in. diameter, or, say, 20 sq. in. area, giving a press
sq. in. The speed was 190 to 200 revs. per min. It was necess
oil under the bearing by means of a pump. For heavy ho
such as a fly-wheel shaft, carrying 100 tons on two journals,
getting oil into the bearings was to flatten the journal
throughout its whole length to the extent of about an eight
width for each inch in diameter up to 8 in. diameter; above
less flat in proportion to the diameter. At first sight it app
to get a continuous flat place coming round in every revolut
loaded shaft; yet it carried the oil effectually into the bea
much better in consequence than a truly cylindrical journa
side.

In thrust-bearings on torpedo-boats Mr. Thornycroft allow
never more than 50 lbs. per sq. in.

Prof. Thurston (Friction and Lost Work, p. 240) says
pressure per square inch is reached on the slow-working an
pivots of swing bridges.

Mr. Tower says (Proc. Inst. M. E., Jan. 1884): In eccentric

'roc. Inst. C. E. 1886) found that the total frictional re-
ly reduced by diminishing the width of the brass.
s most efficient in reducing the friction when the brass
of from 120° to 60°. The film is probably at its best be-
)° and 110°.
brass of a railway axle-bearing where an oil-groove is cut
d an oil-hole is drilled through the top of the brass into it,
bly on the off side, which is probably due to the oil escap-
aches the crown of the brass, and so leaving the off side
the wear consequently ensues.
the brass wears always on the forward side. The same ob-
made in marine-engine journals, which always wear in
e way to what they might be expected. Mr. Stroudley
ity is due to a film of lubricant being drawn in from the un-
rnal to the aft part of the brass, which effectually lubri-
wear on that side; and that when the lubricant reaches
the brass it is so attenuated down to a wedge shape that
lubrication, and greater wear consequently follows.
on (*Am. Mach.*, Oct. 30, 1890) says: Regarding the pres-
subjected in railroad car-service, it is probably more severe
lass of practice. Car brasses, when used bare, are so im-
he journal, that during the early stages of their use the
y be but about one square inch. In this case the pressure
upwards of 6000 lbs. But at the slowest speeds of freight
f a brass is so rapid that, within about thirty minutes the
ased to about three inches, and is thereby able to relieve
atter can successfully prevent overheating of the journal,
takes place with *any oil*, and measures of relief must be
nate the question of differences of lubricating power
t lubricants available. A brass which has been run about
00 lbs. load may have extended the area of bearing-surface
re inches. The pressure is then about 1700 lbs. per square
ssumed that this is an average minimum area for car-ser-
nt and unmanageable overheating has occurred during the
short time. This area will very slowly increase with any

r, Feb. 1893) says: One of the most vital points of an en-
ervice is that of main bearings. They should have a sur-
ot exceeding 350 feet per minute, with a mean bearing-
e inch of projected area of journal of not more than 80
erably within the safe limit of cool performance and easy
bearings are designed in this way, it would admit the use
e main wearing-surface, which in a large type of engines
rk we think advisable.
in a Bearing.—Mr. Beauchamp Tower (Proc. Inst.
ade experiments with a brass bearing 4 inches diameter
determine the pressure of the oil between the brass and
bearing was half immersed in oil, and had a total load of
The journal rotated 150 revolutions per minute. The
was determined by drilling small holes in the bearing at
d connecting them by tubes to a Bourdon gauge. It was
sure varied from 310 to 625 lbs. per square inch, the great-
a little to the "off" side of the centre line of the top of
direction of motion of the journal. The sum of the up-
d by these pressures for the whole lubricated area was
e total pressure on the bearing. The speed was reduced
lutions, but the oil-pressure remained the same, showing
s as completely oil-borne at the lower speed as at the
ving was the observed friction at the lower speed:

, lbs. per square inch...	443	333	211	89
f friction	.00132	.00168	.00247	.0044

*per square inch is the total load divided by the product of
ngth of the journal.* At the same low speed of 20 revo-
t was increased to 676 lbs. per square inch without any
eizing.
r-journal Brasses. (J. E. Dento
rass dressed with an emery-wheel, lo
earing-surface on the journal, as sh

of a portion of the surface, of only 1 square inch. With this pressure o lbs. per square inch, the coefficient of friction may be 6%, and the bras be overheated, scarred and cut but, on the contrary, it may wear down to a smooth bearing, giving a highly polished area of contact of 3 s inches, or more, inside of two hours of running, gradually decreasin pressure per square inch of contact, and a coefficient of friction of les 0.5%. A reciprocating motion in the direction of the axis is of impo in reducing the friction. With such polished surfaces any oil will lubr and the coefficient of friction then depends on the viscosity of the oil. a pressure of 1000 lbs. per square inch, revolutions from 170 to 320 per m and temperatures of 75° to 113° F. with both sperm and parraffine oils efficient of as low as 0.11% has been obtained, the oil being fed contin by a pad.

Experiments on Overheating of Bearings.—Hot B (Denton.)—Tests with car brasses loaded from 1100 to 4500 lbs. per s inch gave 7 cases of overheating out of 32 trials. The tests show how a matter of chance is the overheating, as a brass which ran hot at 5 load on one day would run cool on a later date at the same or higher sure. The explanation of this apparently arbitrary difference of beha that the accidental variations of the smoothness of the surfaces, alm finitesimal in their magnitude, cause variations of friction which are tending to produce overheating, and it is solely a matter of chance these tendencies preponderate over the lubricating influence of t There is no appreciable advantage shown by sperm-oil, when there is dency to overheat—that is, paraffine can lubricate under the highest sures which occur, as well as sperm, when the surfaces are within the tions affording the minimum coefficients of friction.

Sperm and other oils of high heat-resisting qualities, like vegetable petroleum cylinder stocks, only differ from the more volatile lubr like paraffine, in their ability to reduce the chances of the continua dental infinitesimal abrasion producing overheating.

The effect of emery or other gritty substance in reducing overheatin bearing is thus explained:

The effect of the emery upon the surfaces of the bearings is to cov latter with a series of parallel grooves, and apparently after such g are made the presence of the emery does not practically increase the f over the amount of the latter when pure oil only is between the su The infinite number of grooves constitute a very perfect means of in a uniform oil supply at every point of the bearings. As long as groo the journal match with those in the brasses the friction appears to a to only about 10% to 15% of the pressure. But if a smooth journal is between a set of brasses which are grooved, and pressure be applie journal crushes the grooves and becomes brazed or coated with bras then the coefficient of friction becomes upward of 40%. If then em applied, the friction is made very much less by its presence, becau grooves are made to match each other, and a uniform oil supply prev every point of the bearings, whereas before the application of the many spots of the latter receive no oil between them.

Moment of Friction and Work of Friction of Slid surfaces, etc.

	Moment of Friction, inch-lbs.	Energy lost by F in ft.-lbs. per m
Flat surfaces....................	....	fWS
Shafts and journals..............	$\frac{1}{2}fWd$	$.2618fWdn$
Flat pivots......................	$\frac{2}{3}fWr$	$.1745fWrn$
Collar-bearing...................	$\frac{2}{3}fW\frac{r_2^3 - r_1^3}{r_2^2 - r_1^2}$	$.1745fWn\frac{r_2^3 -}{r_2^2 -}$
Conical pivot....................	$\frac{2}{3}fWr$ cosec α	$.1745fWrn$ cose
Conical journal................	$\frac{2}{3}fWr$ sec α	$.1745fWrn$ sec
Truncated-cone pivot.............	$\frac{2}{3}fW\frac{r_2^3 - r_1^3}{r_2 \sin \alpha}$	$.1745fW\frac{r_2^3 -}{r_2 \sin}$
Hemispherical pivot..............	fWr	$.5236fWr$
[illegible] Schiele's "anti-[illegible]vot...............	fWr	$.5236fWr$

he above f = coefficient of friction;
W = weight on journal or pivot in pounds;
r = radius, d = diameter, in inches;
S = space in feet through which sliding takes place;
r_2 = outer radius, r_1 = inner radius;
n = number of revolutions per minute;
a = the half-angle of the cone, i.e., the angle of the slope with the axis.

o obtain the horse-power, divide the quantities in the last column by 00. Horse-power absorbed by friction of a shaft $= \frac{fWdn}{126050}$.

e formula for energy lost by shafts and journals is approximately true loosely fitted bearings. Prof. Thurston shows that the correct formula es according to the character of fit of the bearing; thus for loosely d journals, if U = the energy lost,

$$U = \frac{2f\pi r}{\sqrt{1+f^2}} Wn \text{ inch-pounds} = \frac{.2618fWdn}{\sqrt{1+f^2}} \text{ foot-lbs.}$$

perfectly fitted journals $U = 2.54f\pi rWn$ inch-lbs. $= .3325fWdn$, ft.-lbs. r a bearing in which the journal is so grasped as to give a uniform sure throughout, $U = f\pi^2 rWn$ inch-lbs. $= .4112fWdn$, ft.-lbs. sistance of railway trains and wagons due to friction of trains:

$$\text{Pull on draw-bar} = \frac{f \times 2240}{R} \text{ pounds per gross ton,}$$

hich R is the ratio of the radius of the wheel to the radius of journal. cylindrical journal, perfectly fitted into a bearing, and carrying a total , distributes the pressure due to this load unequally on the bearing, the imum pressure being at the extremity of the vertical radius, while at extremities of the horizontal diameter the pressure is zero. At any t of the bearing-surface at the extremity of a radius which makes an e θ with the vertical radius the normal pressure is proportional to cos θ. = normal pressure on a unit of surface, w = total load on a unit of th of the journal, and r = radius of journal,

$$w \cos\theta = 1.57rp, \quad p = \frac{w \cos\theta}{1.57r}.$$

PIVOT-BEARINGS.

he Schiele Curve.—W. H. Harrison, in a letter to the *Am. Machin-* 1891, says the Schiele curve is not as good a form for a bearing as the nent of a sphere. He says: A mill-stone weighing a ton frequently s its whole weight upon the flat end of a hard-steel pivot 1⅛" diameter, ne square inch area of bearing; but to carry a weight of 3000 lbs. he ses an end bearing about 4 inches diameter, made in the form of a seg- t of a sphere about ⅛ inch in height. The die or fixed bearing should lished to fit the pivot. This form gives a chance for the bearing to st itself, which it does not have when made flat, or when made with the ele curve. If a side bearing is necessary it can be arranged farther up shaft. The pivot and die should be of steel, hardened; cross-gutters ld be in the die to allow oil to flow, and a central oil-hole should be e in the shaft.

e advantage claimed for the Schiele bearing is that the pressure is uni- nly distributed over its surface, and that it therefore wears uniformly. red Lewis (*Am. Mach.*, April 19, 1894) says that its merits as a thrust- ing have been vastly overestimated; that the term "anti-friction" lied to it is a misnomer, since its friction is greater than that of a flat or collar of the same diameter. He advises that flat thrust-bearings ld always be annular in form, having an inside diameter one half of external diameter

riction of a Flat Pivot-bearing.—The Research Committee riction (*Proc. Inst. M. E.* 1891) experimented on a step-bearing, fla t, 3 in. diam., the oil being forced into the bearing through a hole tre and distributed through two radial grooves, insuring thoro tion. The step was of steel and the bearing of manganese-bron

At revolutions per min..........	50	128	194	280
The coefficient of friction varied between	.0181 and .0221	.0053 .0113	.0051 .0102	.0041 .0178

With a white-metal bearing at 128 revolutions the coefficient of fr[illegible] was a little larger than with the manganese-bronze. At the higher s[illegible] the coefficient of friction was less, owing to the more perfect lubricati[illegible] shown by the more rapid circulation of the oil. At 128 revolutio[illegible] bronze bearing heated and seized on one occasion with a load of 260 [illegible] and on another occasion with 300 pounds per square inch. The white bearing under similar conditions heated and seized with a load [illegible] pounds per square inch. The steel footstep on manganese-bronze wa[illegible] wards tried, lubricating with three and with four radial grooves; [illegible] friction was from one and a half times to twice as great as with only [illegible] grooves. (See also Allowable Pressures, page 936.)

Mercury-bath Pivot.—A nearly frictionless step-bearing [illegible] obtained by floating the bearing with its superincumbent weight up[illegible] cury. Such an apparatus is used in the lighthouses of La Heve, Ha[illegible] is thus described in *Eng'g*, July 14, 1893, p. 41:

The optical apparatus, weighing about 1 ton, rests on a circular [illegible] table, which is supported by a vertical shaft of wrought iron [illegible] diameter.

This is kept in position at the top by a bronze ring and outer iron [illegible] and at the bottom in the same way, while it rotates on a removal[illegible] pivot resting in a steel socket, which is fitted to the base of the supp[illegible] the vertical shaft there is rigidly fixed a floating cast-iron ring 17.1 [illegible] eter and 11.8 in. in depth, which is plunged into and rotates in a [illegible] bath contained in a fixed outer drum or tank, the clearance betw[illegible] vertical surfaces of the drum and ring being only 0.2 in., so as to re[illegible] much as possible the volume of mercury (about 220 lbs.), while the [illegible] tal clearance at the bottom is 0.4 in.

BALL-BEARINGS, FRICTION ROLLERS, ET[illegible]

A. H. Tyler (*Eng'g*, Oct. 20, 1893, p. 483), after experiments a[illegible] parison with experiments of others arrives at the following conclusi[illegible]

That each ball must have two points of contact only.

The balls and race must be of glass hardness, and of absolute trut[illegible]

The balls should be of the largest possible diameter which the [illegible] disposal will admit of.

Any one ball should be capable of carrying the total load upon the [illegible]

Two rows of balls are always sufficient.

A ball-bearing requires no oil, and has no tendency to heat unle[illegible] loaded.

Until the crushing strength of the balls is being neared, the fricti[illegible] sistance is proportional to the load.

The frictional resistance is inversely proportional to the diamete[illegible] balls, but in what exact proportion Mr. Tyler is unable to say. Pr[illegible] varies with the square.

The resistance is independent of the number of balls and of the sp[illegible]

No rubbing action will take place between the balls, and devices [illegible] against it are unnecessary, and usually injurious.

The above will show that the ball-bearing is most suitable for hig[illegible] and light loads. On the spindles of wood-carving machines some [illegible] much as 30,000 revolutions per minute. They run perfectly cool, an[illegible] have any oil upon them. For heavy loads the balls should not be [illegible] two thirds the diameter of the shaft, and are better if made equal t[illegible]

Ball-bearings have not been found satisfactory for thrust-bl[illegible] the reason apparently that the tables crowd together. Better resu[illegible] been obtained from coned rollers. A combined system of rollers a[illegible] is described in *Eng'g*, Oct. 6, 1893, p. 429.

Friction-rollers.—If a journal instead of revolving on [illegible] bearings be supported on friction-rollers the force required to make [illegible] nal revolve will be reduced in nearly the same proportion that the d[illegible] of the axles of the rollers is less than the diameter of the rollers the[illegible] In experiments by A. M. Wellington with a journal 3½ in. diam., [illegible] on rollers 8 in. diam., whose axles were 1¾ in. diam., the friction [illegible] [illegible] rest was ¼ the friction of an ordinary 3½-in. bearing, [illegible] [illegible] of 10 miles per hour it was ½ that of the ordinary bearin[illegible] [illegible] of the axle to diam. of roller was 1¾:8, or as 1 to [illegible]

earings for Very High Rotative Speeds. (Proc. Inst 1888, p. 482.)—In the Parsons steam-turbine, which has a spee as 18,000 rev. per min., as it is impossible to secure absolute ac alance, the bearings are of special construction so as to allow of a small amount of lateral freedom. For this purpose the bearing de 1 by two sets of steel washers 1/16 inch thick and of different s, the larger fitting close in the casing and about 1/32 inch clear ing, and the smaller fitting close on the bearing and about 1/3 r of the casing. These are arranged alternately, and are pr ther by a spiral spring. Consequently any lateral movement o ing causes them to slide mutually against one another, and by ion to check or damp any vibrations that may be set up in the spi tendency of the spindle is then to rotate about its axis of mass, or l axis as it is called; and the bearings are thereby relieved from ex pressure, and the machine from undue vibration. The finding of re of gyration, or rather allowing the turbine itself to find its re of gyration, is a well-known device in other branches of mechan n the instance of the centrifugal hydro-extractor, where a mass v h out of balance is allowed to find its own centre of gyration; the fas n the more steadily did it revolve and the less was the vibration. r illustration is to be found in the spindles of spinning machine ch run at about 10,000 or 11,000 revolutions per minute: they are ma ardened and tempered steel, and although of very small dimensions, t side diameter of the largest portion or driving whorl being perhaps n e than 1¼ in., it is found impracticable to run them at that speed t might be called a hard-and-fast bearing. They are therefore run wit e elastic substance surrounding the bearing, such as steel springs, hem ork. Any elastic substance is sufficient to absorb the vibration, an mit of absolutely steady running.

FRICTION OF STEAM-ENGINES.

istribution of the Friction of Engines.—Prof. Thurston in "Friction and Lost Work," gives the following:

	1.	2.	3.
Main bearings	47.0	35.4	35.0
Piston and rod	32.9	25.0	21.0
Crank-pin	6.8	5.1	13.0
Cross-head and wrist-pin	5.4	4.1	
Valve and rod	2.5	26.4	22.0
Eccentric strap	5.3	4.0	
Link and eccentric	....	....	9.01
Total	100.0	100.0	100.0

o. 1, Straight-line, 6″ × 12″, balanced valve; No. 2, Straight-line, 6″ × 12″, alanced valve; No. 3, 7″ × 10″, Lansing traction locomotive valve-gear. rof. Thurston's tests on a number of different styles of engines indicate t the friction of any engine is practically constant under all loads. ans. A. S. M. E., viii. 86; ix. 74.)

a Straight-line engine, 8″ × 14″, I.H.P. from 7.41 to 57.54, the friction H. aried irregularly between 1.97 and 4.02, the variation being independent he load. With 50 H.P. on the brake the I.H.P. was only 52.6, the friction g only 2.6 H.P., or about 5%.

a compound condensing-engine, tested from 0 to 102.6 brake H.P., gave P. from 14.92 to 117.8 H.P., the friction H.P. varying only from 14.92 to . At the maximum load the friction was 15.2 H.P., or 12.9%.

e friction increases with increase of the boiler-pressure from 30 to 70 nd then becomes constant. The friction generally increases with in- of speed, but there are exceptions to this rule.

. Denton (*Stevens Indicator*, July, 1890), comparing the calculated n of a number of engines with the friction as determined by measure- finds that in one case, a 75-ton ammonia ice-machine, the friction of npressor, 17½ H.P., is accounted for by a coefficient of friction of 7½% he external bearings, allowing 6% of the entire friction of the machine friction of pistons, stuffing-boxes, and valves. In the case of the ket pumping-engine, estimating the friction of the external bearings oefficient of friction of 6% and that of the pistons, valves, and stuff- s as *in the case of the* ice-machine, we have the total friction *ed as follows:*

	Horse-power.	Per cent of Whole.
Crank-pins and effect of piston-thrust on main shaft..	0.71	11.4
Weight of fly-wheel and main shaft	1.95	[illegible]
Steam-valves	0.23	[illegible]
Eccentric	0.07	[illegible]
Pistons	0.43	[illegible]
Stuffing-boxes, six altogether	0.72	[illegible]
Air-pump	2.10	[illegible]
Total friction of engine with load	6.21	100
Total friction per cent of indicated power	4.27	

The friction of this engine, though very low in proportion to the ind power, is satisfactorily accounted for by Morin's law used with a coef of friction of 5%. In both cases the main items of friction are those the weight of the fly-wheel and main shaft and to the piston-thr crank-pins and main-shaft bearings. In the ice-machine the latter are the larger owing to the extra crank-pin to work the pumps in the Pawtucket engine the former preponderates, as the crank-thr partly absorbed by the pump-pistons, and only the surplus effect the crank-shaft.

Prof. Denton describes in Trans. A. S. M. E., x. 392, an appar which he measured the friction of a piston packing-ring. When th of the piston were thoroughly devoid of lubricant, the coefficient of was found to be about 7½%; with an oil-feed of one drop in two min coefficient was about 5%; with one drop per minute it was about 3% rates of feed gave unsatisfactory lubrication, the piston groaning ends of the stroke when run slowly, and the flow of oil left upon the was found by analysis to contain about 50% of iron. A feed of two d minute reduced the coefficient of friction to about 1%, and gave pr perfect lubrication, the oil retaining its natural color and purity.

LUBRICATION.

Measurement of the Durability of Lubricants. (J. ton, Trans. A. S. M. E., xi. 1013.)—Practical differences of durability cants depend not on any differences of inherent ability to resist bein out" by rubbing, but upon the rate at which they flow through a from the bearing-surfaces. The conditions which control this flo delicate in their influence that all attempts thus far made to measu bility of lubricants may be said to have failed to make distinctions cating value having any practical significance. In some kinds of se limit to the consumption of oil depends upon the extent to which dust refuse becomes mixed with it, as in railroad-car lubrication and in of agricultural machinery. The economy of one oil over another, the quality used is concerned—that is, so far as durability is conce simply proportional to the rate at which it can insinuate itself into out of minute orifices or cracks. Oils will differ in their ability to first, in proportion to their viscosity, and, second, in proportion to pillary properties which they may possess by virtue of the particul dients used in their composition. Where the thickness of film betw bing-surfaces must be so great that large amounts of oil pass bearings in a given time, and the surroundings are such as to perm be fed at high temperatures or applied by a method not requiring a fluidity, it is probable that the least amount of oil will be used when cosity is as great as in the petroleum cylinder stocks. When, howe oil must flow freely at ordinary temperatures and the feed o restricted, as in the case of crank-pin bearings, it is not practicable such heavy oils in a satisfactory manner. Oils of less viscosity fluidity approximating to lard-oil must then be used.

Relative Value of Lubricants. (J. E. Denton, *Am. Mach.*, 1890.)—The three elements which determine the value of a lubricant *cost due to* consumption of lubricants, the cost spent for coal to o *the frictional* resistance caused by use of the lubricant, and the cos *metallic* wear on the journal and the brasses. In cotton-mills *the power* is alone to be considered; in rolling-mills and mari *cost of the* quantity of lubricant used is the only important *railroads not* only do both these elements enter the problem t the cost of the wearing away of the metallic parts enters in ad- furthermore, the latter is the greatest element of cost in the case.

ualifications of a Good Lubricant, as laid down by ey, in Proc Inst. C. E., vol. xiv., p. 372, are: 1. Sufficient body surfaces free from contact under maximum pressure. 2. The ossible fluidity consistent with the foregoing condition. 3. The sible coefficient of friction, which in bath lubrication would be for on approximately. 4. The greatest capacity for storing and way heat. 5. A high temperature of decomposition. 6. Power idation or the action of the atmosphere. 7. Freedom from cor- on on the metals upon which used.

Lubricants for Different Purposes. (Thurston.)

eratures, as in rock-drills y compressed air:	Light mineral lubricating-oils.
pressures, slow speed..	Graphite, soapstone, and other solid lubricants.
ssures, with slow speed...	The above, and lard, tallow, and other greases.
ssures and high speed....	Sperm-oil, castor-oil, and heavy mineral oils.
sures and high speed.....	Sperm, refined petroleum, olive, rape, cotton-seed.
nachinery	Lard-oil, tallow-oil, heavy mineral oils, and the heavier vegetable oils.
inders......................	Heavy mineral oils, lard, tallow.
nd other delicate mecha-	Clarified sperm, neat's-foot, porpoise, olive, and light mineral lubricating oils.

ture with mineral oils, sperm is best; lard is much used; olive and d are good.

nt of Oil needed to Run an Engine.—The Vacuum Oil 2, in response to an inquiry as to cost of oil to run a 1000-H.P. gine, wrote: The cost of running two engines of equal size of the te is not always the same. Therefore while we could furnish owing what it is costing some of our customers having Corliss 1000 H.P., we could only give a general idea, which in itself considerably out of the way as to the probable cost of cylinder- e-oils per year for a particular engine. Such an engine ought to y on less than 8 drops of 600 W oil per minute. If 3000 drops are the quart, and 8 drops used per minute, it would take about ne half barrels (52.5 gallons) of 600 W cylinder-oil, at 65 cents per about $85 for cylinder-oil per year, running 6 days a week and 10 y. Engine-oil would be even more difficult to guess at what the l be, because it would depend upon the number of cups required ine, which varies somewhat according to the style of the engine. loubtless be safe, however, to calculate at the outside that not twice as much engine-oil would be required as of cylinder-oil.

uum Oil Co. in 1892 published the following results of practice W" cylinder-oil:

npound engine,	20 and 33 × 48; 83 revs. per min.; 1 drop of oil per min. to 1 drop in two minutes.
ple exp. "	20, 33, and 46 × 48; 1 drop every 2 minutes.
en "	20 and 36 × 36; 143 revs. per min.; 2 drops of oil per min., reduced afterwards to 1 drop per min.
"	15 × 25 × 16; 240 revs. per min.; 1 drop every 4 minutes.

of tests on ocean-steamers communicated to the author by Prof. 1892 gave: for 1200-H.P. marine engine, 5 to 6 English gallons (6 to als.) of engine-oil per 24 hours for external lubrication; and for a marine engine, triple expansion, running 75 revs. per min., 6 to 7 *ls. per 24 hours. The cylinder-oil consumption is exceedingly from 1 to 4 gals. per day* on different engines, including cylinder- *wab the piston-rods.*

y of Oil used on a Locomotive Crank-pi

s. A. S. M. E., xi. 1020, says: A very economical case *ion is when a* locomotive main crank-pin consume cubic inches of oil in a thousand miles of service. This is equivalent to a consumption of one milligram to seventy square inches of surface rubbed over.

The Examination of Lubricating-oils. (Prof. Thos. B. Stillman, *Stevens Indicator*, July, 1890.)—The generally accepted conditions of a good lubricant are as follows:

1. "Body" enough to prevent the surfaces, to which it is applied, from coming in contact with each other. (Viscosity.)
2. Freedom from corrosive acid, either of mineral or animal origin.
3. As fluid as possible consistent with "body."
4. A minimum coefficient of friction.
5. High "flash" and burning points.
6. Freedom from all materials liable to produce oxidation or "gumming."

The examinations to be made to verify the above are both chemical and mechanical, and are usually arranged in the following order:

1. Identification of the oil, whether a simple mineral oil, or animal oil, or a mixture. 2. Density. 3. Viscosity. 4. Flash-point. 5. Burning-point. 6. Acidity. 7. Coefficient of friction. 8. Cold test.

Detailed directions for making all of the above tests are given in Prof. Stillman's article.

Weights of Oil per Gallon.—The following are approximately the weights per gallon of different kinds of oil (Penn. R. R. Specifications):

Lard-oil, tallow-oil, neat's-foot oil, bone-oil, colza-oil, mustard-seed oil, rape-seed oil, paraffine-oil, 500° fire-test oil, engine-oil, and cylinder lubricant, 7½ pounds per gallon.

Well-oil and passenger-car oil, 7.4 pounds per gallon; navy sperm-oil, 7.2 pounds per gallon; signal-oil, 7.1 pounds per gallon; 300° burning-oil, 6.[illegible] pounds per gallon; and 150° burning-oil, 6.6 pounds per gallon.

Penna. R. R. Specifications for Petroleum Products, 1889.—Five different grades of petroleum products will be used.

The materials desired under this specification are the products of the distillation and refining of petroleum unmixed with any other substances.

150° *Fire-test Oil.*—This grade of oil will not be accepted if sample (1) is not "water-white" in color; (2) flashes below 130° Fahrenheit; (3) burns below 151° Fahrenheit; (4) is cloudy or shipment has cloudy barrels when received, from the presence of glue or suspended matter; (5) becomes opaque or shows cloud when the sample has been 10 minutes at a temperature of 0° Fahrenheit.

The flashing and burning points are determined by heating the oil in an open vessel, not less than 12° per minute, and applying the test flame every 7°, beginning at 123° Fahrenheit. The cold test may be conveniently made by having an ounce of the oil, in a four-ounce sample bottle, with a thermometer suspended in the oil, and exposing this to a freezing mixture of ice and salt. It is advisable to stir with the thermometer while the oil is cooling. The oil must remain transparent in the freezing mixture ten minutes after it has cooled to zero.

300° *Fire-test Oil.*—This grade of oil will not be accepted if sample (1) is not "water white" in color; (2) flashes below 249° Fahrenheit; (3) burns below 298° Fahrenheit; (4) is cloudy or shipment has cloudy barrels when received, from the presence of glue or suspended matter; (5) becomes opaque or shows cloud when the sample has been 10 minutes at a temperature of 32° Fahrenheit.

The flashing and burning points are determined the same as for 150° fire-test oil, except that the oil is heated 15° per minute, test-flame being applied first at 242° Fahrenheit. The cold test is made the same as above, except that ice and water are used.

Paraffine-oil.—This grade of oil will not be accepted if the sample (1) is other than pale-lemon color; (2) flashes below 249° Fahrenheit; (3) shows viscosity less than 40 seconds or more than 65 seconds when tested as described under "Well Oil" at 100° Fahrenheit throughout the year; (4) has gravity at 60° Fahrenheit, below 24° Baumé, or above 29° Baumé; (5) from October 1st to May 1st has a cold test above 10° Fahrenheit.

The flashing-point is determined same as for 300° fire-test oil. The cold test is determined as follows: A couple of ounces of oil is put in a four-ounce sample bottle, and a thermometer placed in it. The oil is then frozen, a freezing mixture of ice and salt being used if necessary. When the oil has become hard, the bottle is removed from the freezing mixture and the frozen oil allowed to soften, being stirred and thoroughly mixed at the same time by means of the thermometer, until the mass will run from one end of

to the other. The reading of the thermometer when this is the
arded as the cold test of the oil.
.—This grade of oil will not be accepted if the sample (1) flashes, 1st to October 1st, below 249° Fahrenheit, or from October 1st to low 200° Fahrenheit; (2) has a gravity, at 60° Fahrenheit, below , or above 30°; (3) from October 1st to May 1st has a cold test Fahrenheit; (4) shows any precipitation in 10 minutes when 5 imetres are mixed with 95 cubic centimetres of 88° gasoline; (5) cosity less than 55 seconds, or more than 100 seconds, when tested d below. From October 1st to May 1st the test must be made renheit, and from May 1st to October 1st at 110° Fahrenheit.
mer oil the flashing-point is determined the same as for paraffine-r winter oil the same, except that the test-flame is applied first renheit. The cold test is made the same as for paraffine-oil.
cipitation test is to exclude tarry and suspended matter. It is de by putting 5 cubic centimetres of the oil in a 100-cubic-cen-aduate, then filling to the mark with gasoline, and thoroughly

sity test is made as follows: A 100 cubic-centimetre pipette of the orm is regraduated to hold just 100 cubic centimetres to the bottom The size of the aperture at the bottom is then made such that entimetres of water at 100° Fahrenheit will run out the pipette e bottom of the bulb in 34 seconds. Pipettes with bulbs varying ches to 1¼ inches in diameter outside, and about 4¼ inches long t exactly the same results, provided the aperture at the bottom er size. The pipette being obtained, the oil sample is heated to d temperature, care being taken to have it uniformly heated, and wn up into the pipette to the proper mark. The time occupied in running out, down to the bottom of the bulb, gives the test

-test Oil.—This grade of oil will not be accepted if sample (1) ow 415° Fahrenheit; (2) shows precipitation with gasoline when escribed for well-oil.
ing-point is determined the same as for well-oil, except that the s applied first at 438° Fahrenheit.

SOLID LUBRICANTS.

ite in a condition of powder and used as a *solid lubricant*, so istinguish it from a liquid lubricant, has been found to do well atter has failed.
n 1829, says: "Graphite lessened friction in all cases where it General Morin, at a later date, concluded from experiments uld be used with advantage under heavy pressures; and Prof. ound it well adapted for use under both light and heavy pressures d with certain oils. It is especially valuable to prevent abrasion under heavy loads and at low velocities.
one, also called talc and steatite, in the form of powder and oil or fat, is sometimes used as a lubricant. Graphite or soap-d with soap, is used on surfaces of wood working against either d.
raphite.—A new self-lubricating bearing known as fibre-described by John H. Cooper in Trans. A. S. M. E., xiii, 374, as on of P. H. Holmes, of Gardiner, Me. This bearing material is of selected natural graphite, which has been finely divided and foreign and gritty matter, to which is added wood-fibre or other ced in water in various proportions, according to the purpose to and then solidified by pressure in specially prepared moulds; val from which the bearings are first thoroughly dried, then satu-a drying oil, and finally subjected to a current of hot, dry air for e of oxidizing the oil, and hardening the mass. When finished, e "machined" to size or shape with the same facility and means n metals.
ne is a solid compound, usually containing graphite, made in the *all cylinders which are fitted* permanently into holes drilled in *of the bearing*. The bearing thus fitted runs without any other

hour, of cupolas from 24 inches to 84 inches in diameter.

Capacity of Cupola.—The accompanying table will be of mining the capacity of cupola needed for the production of of iron in a specified time.

First, ascertain the amount of iron which is likely to be cast, and the length of time which can be devoted profitabl and supposing that two hours is all that can be spared for t that ten tons is the amount which must be melted, find in th ing Capacity per hour in Pounds, the nearest figure to fiv which is found to be 10,760 pounds per hour, opposite to whi Diameter of Cupolas, Inside Lining, will be found 48 inches size of cupola required to furnish ten tons of molten iron in

Or suppose that the heats were likely to average 6 tons, w increase up to ten, then it might not be thought wise to inc pense consequent on working a 48-inch cupola, in which ca the directions given, it will be found that a 40-inch cupola w purpose for 6 tons, but would require an additional hour's t whenever the 10-ton heat came along.

The quotations in the table are not supposed to be all th in the hour by some of the very best cupolas, but are sim which a common cupola under ordinary circumstances may melt in the time specified.

Height of Cupola.—By height of cupola is meant the di base to the bottom side of the charging hole.

Depth of Bottom of Cupola.—Depth of bottom is the di sand-bed, after it has been formed at the bottom of the c under side of the tuyeres.

All the amounts for fuel are based upon a bottom of 10 i any departure from this depth must be met by a correspo the quantity of fuel used on the bed; more in proportion increased, and less when it is made shallower.

Amount of Fuel Required on the Bed.—The column "Am quired on Bed, in Pounds" is based on the supposition tha straight one all through, and that the bottom is 10 inches de tom be more, as in those of the Colliau type, then additio needed.

The amounts being given in pounds, answer for both co should coal be used, it would reach about 15 inches above t same weight of coke would bring it up to about 22 inches ab which is a reliable amount to stock with.

First Charge of Iron.—The amounts given in this column

D... in...	Heig... bot... of ... door	Dept... sand... under Tuy...	Am't requ... bed.	First ... of Ir...	Succe... char... Fue...	Succe... char... Iron	Diam... Blast... when... ceed'... in le...	Numb... Tuye... inche... requ... each	to the 6-inch round ones.		Blast-... requ...	Sizes o... Blow... quire...	Revol... per m...	H.P. ... gine to Root B...	Sizes of Sturte... Blowe...	H.P. o... gine to Sturte... Blowe...	Meltin... pacit... hour
Inches.	Feet.	Inches.	Pounds.	Pounds.	Pounds.	Pounds.	Inches.	No.	No.	Dimensions.	Oz.	No.		H.P.	No.	H.P.	Pounds.
24	10	10	300	900	50	500	10	1.5	2	10″ × 2″	6	½	250	¾	2	1	1,500
26	10	10	390	1,170	92	828	10	2.	2	14½″ × 2″	6	½	335	1	2	1	2,000
28	10	10	480	1,440	134	1,206	10	2.6	3	12½″ × 2″	6	1	210	1¼	2	1	2,500
30	12	10	570	1,710	176	1,584	12	2.6	3	12½″ × 2″	7	1	250	1½	3	2	3,000
32	12	10	660	1,980	218	1,962	12	3.3	4	12″ × 2″	7	1	292	1¾	3	2	3,500
34	12	10	750	2,250	260	2,340	12	3.3	4	12″ × 2″	7	1	333	2	3	2	4,000
36	12	10	840	2,520	302	2,718	14	3.7	4	13½″ × 2″	8	2	241	2½	4	3	4,820
38	12	10	930	2,790	344	3,096	14	5.	6	12″ × 2″	8	2	282	3½	4	3	5,640
40	13	10	1,020	3,060	386	3,474	14	5.	6	12″ × 2″	8	2	325	4	4	3	6,460
42	13	10	1,110	3,330	428	3,852	16	5.8	6	11½″ × 2½″	10	3	235	6	5	5½	7,550
44	13	10	1,200	3,600	470	4,230	16	5.8	6	11½″ × 2½″	10	3	270	7½	5	5½	8,640
46	13	10	1,290	3,870	512	4,608	16	6.8	6	13½″ × 2½″	10	3	304	8½	5	5½	9,730
48	13	10	1,380	4,140	554	4,986	18	6.8	6	13½″ × 2½″	12	4	212	10	6	9¾	10,760
50	14	10	1,470	4,410	596	5,364	18	8.	8	11½″ × 2½″	12	4	236	12	6	9¾	11,790
52	14	10	1,560	4,680	638	5,742	18	10.7	8	15½″ × 2½″	12	4	256	13	6	9¾	12,820
54	14	10	1,650	4,950	680	6,120	20	10.7	8	15½″ × 2½″	14	4	277	14	7	16	13,850
56	14	10	1,740	5,220	722	6,498	20	12.2	8	17½″ × 2½″	14	5	170	16	7	16	14,880
58	14	10	1,830	5,490	764	6,876	20	12.2	8	17½″ × 2½″	14	5	180	17½	7	16	15,910
60	15	10	1,920	5,760	806	7,254	22	13.7	8	16½″ × 3″	14	5	192	18½	8	22	16,940
62	15	10	2,010	6,030	848	7,632	22	13.7	8	16½″ × 3″	14	5	208	20	8	22	18,340
64	15	10	2,100	6,300	890	8,110	22	15.4	8	18½″ × 3″	14	5	225	22	8	22	19,770
66	15	10	2,190	6,570	932	8,388	22	15.4	8	18½″ × 3″	14	5	240	23	8	22	21,200
68	15	10	2,280	6,840	974	8,766	22	17.1	10	16½″ × 3″	14	6	142	25	8	22	22,630
70	16	10	2,370	7,110	1,016	9,144	22	19.	10	18½″ × 3″	14	6	150	27	8	22	24,060
72	16	10	2,460	7,380	1,058	9,522	24	19.	10	18½″ × 3″	16	6	163	33	9	35	26,070
74	16	10	2,550	7,650	1,100	9,900	24	23.9	12	16½″ × 3½″	16	6	175	35	9	35	27,980
76	16	10	2,640	7,920	1,142	10,278	24	23.9	12	16½″ × 3½″	16	6	187	37	9	35	29,890
78	16	10	2,730	8,190	1,184	10,656	24	26.	12	18½″ × 3½″	16	6	200	40	9	35	31,800
80	16	10	2,820	8,460	1,226	11,134	24	26.	12	18½″ × 3½″	16	7	140	42	9	35	33,710
82	16	10	2,910	8,730	1,268	11,412	24	28.	14	17″ × 3½″	16	7	148	45	9	35	35,620
84	16	10	3,000	9,000	1,310	11,790	26	31.	16	16″ × 3½″	16	7	160	47	10	48	37,530

tuyeres 16 inches by 13½ inches.

If it is found that the given number of flat tuyeres exc that of the diminished part of the cupola, they can be the decreased length to be added to the depth, or the end; by so doing, we arrive at a modified form of the B

Another important point in this connection is to ar such a manner as will concentrate the fire at the m smallest possible compass, so that the metal in fusion to traverse while exposed to the oxidizing influence of t

To accomplish this, recourse has been had to the p rows of tuyeres in some instances—the "Stewart ra three rows, and the "Colliau cupola furnace" having t

Blast-pressure.—Experiments show that about 30,000 consumed in melting a ton of iron, which would weigh or more than both iron and fuel. When the proper q plied, the combustion of the fuel is perfect, and car result. When the supply of air is insufficient, the comb and carbonic-oxide gas is the result. The amount of h two cases is as 15 to 4½ showing a loss of over two third perfect combustion.

It is not always true that we obtain the most rapid r forcing into the cupola the largest quantity of air. S to elevate the temperature of the air supplied to the po into combustion. If more air than this is supplied, it r reduces the temperature, and retards combustion, and may be extinguished with too much blast.

Slag in Cupolas.—A certain amount of slag is nece molten iron which has fallen to the bottom from the ac it was not there, the iron would suffer from decarboniz

When slag from any cause forms in too great abunda away by inserting a hole a little below the tuyeres, t find its way as the iron rises in the bottom.

In the event of clean iron and fuel, slag seldom form extent in small heats; this renders any preparation fo necessary, but when the cupola is to be taxed to its t then incumbent on the melter to flux the charges all th rying it away in the manner directed.

ld 8 to 10 pounds of metal; any well-constructed cupola will

(*Am. Mach.*, Mar. 5, 1891) gives the following as the practice ron-works, Carteret, N. J.: "We melt daily from twenty to n, with an average of 11.2 pounds of iron to one of fuel. In seven to nine pounds is good melting, but in a cupola that inches, anything less than nine pounds shows a defect in tuyeres or strength of blast, or in charging up."

r's Text-book," by Thos. D. West, gives forty-six reports in f cupola practice in thirty States, reaching from Maine to

arges in Stove-foundries. (*Iron Age*, April 14, 1892.) are charged exactly the same. The amount of fuel on the the charges differs, while varying amounts of iron are used Below will be found charging-lists from some of the promdries in the country:

	lbs.		lbs.
coke..	1,500	Four next charges of coke, each	150
e of iron	5,000	Six next charges of coke, each	120
arges of iron..	1,000	Nineteen next charges of coke, each........................	100
second charges ach	200		

elt of 18 tons there would be 5120 lbs. of coke used, giving a Increase the amount of iron melted to 24 tons, and a ratio of n to 1 of coal is obtained.

	lbs.		lbs.
coke	1,600	Second and third charges of fuel.........................	130
e of iron	1,800	All other charges of fuel, each	100
e of fuel.......	150		
harges of iron, ...	1,000		

melt 5060 lbs. of coke would be necessary, giving a ratio of o 1 pound of coke.

	lbs.		lbs.
coke	1,600	All other charges of iron.......	2,000
e of iron.......	4,000	All other charges of coke......	150
econd charges	200		

8 tons 4100 lbs. of coke would be used, or a ratio of 8.5 to 1.

	lbs.		lbs.
coke	1,800	All charges of coke, each......	200
e of iron	5,600	All other charges of iron	2,900

18 tons, 3900 lbs. of fuel would be used, giving a ratio of 9.4 to 1 of coke. Very high, indeed, for stove-plate.

	lbs.		lbs.
coal	1,900	All other charges of iron, each	2,000
e of iron.......	5,000	All other charges of coal, each	175
e of coal.......	200		

18 tons 4700 lbs. of coal would be used, giving a ratio of 7.7 lb. of coal.

ficient to demonstrate the varying practices existing among foundries. In all these places the iron was proper for stove- and apparently there was little or no difference in the kind sand at the different foundries.

Increased Driving. (Erie City Iron-works, 1891.)— 60-in. cupola, 100 tons clean castings a week, melting 8 tons er pound of fuel, 7½ lbs.; per cent weight of good castings to 4. *Jan.-May, 1891:* Increased rate of melting to 11½ tons per *b. fuel, 9½; per cent* weight of good castings, 75; one week, *ur, 10.3 lbs. iron per lb.* fuel; per cent weight of good cast- *crease was made by* putting in an additional row of t- *r blast, 14 ounces.* Coke was used as fuel. (W. O. V *xii. 1045.)*

Buffalo Steel Pressure-blowers. Speeds and Capacit[illegible] as applied to Cupolas.

Sq. in. Blast.	No. of Blower.	Diameter inside of Cupola, in.	Pressure in oz.	Speed—No. of Revolutions per minute.	Melting Capacity in pounds, per hour.	Cubic Feet of Air required per minute.	Horse-power required.	Pressure in oz.	Speed—No. of Revolutions per minute.	Melting Capacity in Pounds, per hour.	Cubic Feet of Air required per minute.
4	4	20	8	4793	1545	412	1.0	9	5095	1667	47[illegible]
6	5	25	8	3911	2321	619	1.2	10	4509	[illegible]	[illegible]
8	6	30	8	3456	3093	825	2.05	10	3974	3671	[illegible]
11	7	35	8	3092	4218	1125	3.1	10	3476	4777	[illegible]
14	8	40	8	2702	5425	1444	3.9	10	3034	6085	[illegible]
18	9	45	10	2617	7818	2085	7.1	12	2916	8598	[illegible]
26	10	55	10	2139	11295	3012	10.2	12	2353	12357	[illegible]
46	11	73	12	1639	21978	5861	23.9	14	1777	23838	[illegible]
68	12	88	12	1639	32395	8636	35.2	14	1777	35190	[illegible]

In the table are given two different speeds and pressures for each [illegible] blower, and the quantity of iron that may be melted, per hour, with [illegible] In all cases it is recommended to use the lowest pressure of blast that [illegible] the work. Run up to the speed given for that pressure, and regulate [illegible] tity of air by the blast-gate. The tuyere area should be at least one [illegible] of the area of cupola in square inches, with not less than four tuyere[illegible] equal distances around cupola, so as to equalize the blast throughout. [illegible] riations in temperature affect the working of cupolas materially, [illegible] weather requiring increase in volume of air.

(For tables of the Sturtevant blower see pages 519 and 520.)

Loss in Melting Iron in Cupolas.—G. O. Vair, *Am. M*[illegible] March 5, 1891, gives a record of a 45-in. Colliau cupola as follows:

Ratio of fuel to iron, 1 to 7.42.

Good castings	21,314 lbs.
New scrap	3,005 "
Millings	200 "
Loss of metal	1,481 "
Amount melted	26,000 lbs.

Loss of metal, 5.69%. Ratio of loss, 1 to 17.55.

Use of Softeners in Foundry Practice. (W. Graham, *Iro*[illegible] June 27, 1889.)—In the foundry the problem is to have the right propo[illegible] of combined and graphitic carbon in the resulting casting; this is do[illegible] getting the proper proportion of silicon. The variations in the propo[illegible] of silicon afford a reliable and inexpensive means of producing a cas[illegible] of any required mechanical character which is possible with the ma[illegible] employed. In this way, by mixing suitable irons in the right propor[illegible] a required grade of casting can be made more cheaply than by using [illegible] in which the necessary proportions are already found.

If a strong machine casting were required, it would be necessary to [illegible] the phosphorus, sulphur, and manganese within certain limits. Pro[illegible] Turner found that cast iron which possessed the maximum of the d[illegible] qualities contained, graphite, 2.59%; silicon, 1.42%; phosphorus, 0.3[illegible] phur, 0.06%; manganese, 0.58%.

A strong casting could not be made if there was much increase [illegible] amount of phosphorus, sulphur, or manganese. Irons of the above p[illegible] ages of phosphorus, sulphur, and manganese would be most suitable f[illegible] purpose, but they could be of different grades, having different perce[illegible] of silicon, combined and graphitic carbon. Thus hard irons, mottl[illegible] white irons, and even steel scrap, all containing low percentages of [illegible] and high percentages of combined carbon, could be employed [illegible] having a large amount of silicon were mixed with them in sufficien[illegible] This [illegible] the silicon to the proper proportion and would [illegible] com[illegible] to be forced into the graphitic state, and t[illegible]

l be soft. High-silicon irons used in this way are called "soft-
ing are typical analyses of softeners:

Ferro-silicon.				Softeners, American.			Scotch Irons, No. 1.	
Foreign.		American.		Well-ston.	Globe	Belle-fonte.	Eg-linton	Colt-ness.
10.55	9.80	12.08	10.34	6.67	5.89	3 to 6	2.15	2.59
1.84	0.69	0.06	0.07	....	0.30	0.25	0.21	......
0.52	1.12	1.52	1.92	2.57	2.85	3.	3.76	......
3.86	1.95	0.76	0.52	. . .	1.00	0.53	2.80	1.70
0.04	0.21	0.48	0.45	0.50	1.10	0.35	0.62	0.85
0.03	0.04	Trace	Trace	Trace	0.02	0.03	0.03	0.01

(For other analyses, see pages 371 to 373.)

ons contain a low percentage of total carbon and a high per-
ombined carbon. Carbon is the most important constituent of
d there should be about 3.4% total carbon present. By adding
which contains only 2% of carbon the amount of carbon in the
xture is lessened.
found that more silicon is lost during the remelting of pig of
con than in remelting pig iron of lower percentages of silicon.
ts out the possible disadvantage of using ferro-silicons contain-
a percentage of combined carbon as 0.70% to overcome the bad
mbined carbon in other irons.
h irons generally contain much more phosphorus than is desired
e employed in making the strongest castings. It is a mistake to
rong low-phosphorus irons an iron that would increase the
hosphorus for the sake of adding softening qualities, when soft-
produced by mixing irons of the same low phosphorus.
er discussion of the influence of silicon see page 365.)
age of Castings.—The allowance necessary for shrinkage
ifferent kinds of metal, and the different conditions under which
st. For castings where the thickness runs about one inch, cast
ary conditions, the following allowance can be made:

iron, 1/8	inch per foot.	For zinc,	5/16 inch per foot.
s, 3/16	" " "	" tin,	1/12 " " "
. 1/4	" " "	" aluminum,	3/16 " " "
iron, 1/8	" " "	" Britannia,	1/32 " " "

astings, under the same conditions, will shrink less, and thinner
than this standard. The quality of the material and the manner
g and cooling will also make a difference.
s experiments by W. J. Keep (see Trans. A. S. M. E., vol. xvi.)
t the shrinkage of cast iron of a given section decreases as the
of silicon increases, while for a given percentage of silicon the
lecreases as the section is increased. Mr. Keep gives the follow-
howing the approximate relation of shrinkage to size and per-
silicon:

Sectional Area of Casting.					
½″ □	1″ □	1″ × 2″	2″ □	3″ □	4″ □
Shrinkage in Decimals of an inch per foot of Length.					
.183	.158	.146	.130	.113	.102
.171	.145	.133	.117	.098	.087
.159	.133	.121	.104	.085	.074
.147	.121	.108	.092	.073	.060
.135	.108	.095	.077	.059	.045
.123	.095	.082	.065	.046	.032

iron is 20 ft. per minute, whether for the lathe, planing, shaping machine. (Proc. Inst. M. E., April, 1883, p. 248.)

Table of Cutting-speeds.

Diameter, inches.	Feet per minute.							
	5	10	15	20	25	30	35	40
	Revolutions per minute.							
¼	76.4	152.8	229.2	305.6	382.0	458.4	534.8	611.2
⅜	50.9	101.9	152.8	203.7	254.6	305.6	356.5	407.4
½	38.2	76.4	114.6	152.8	191.0	229.2	267.4	305.6
⅝	30.6	61.1	91.7	122.2	152.8	183.4	213.9	244.5
¾	25.5	50.9	76.4	101.8	127.3	152.8	178.2	203.7
⅞	21.8	43.7	65.5	87.3	109.1	130.9	152.8	174.6
1	19.1	38.2	57.3	76.4	95.5	114.6	133.7	152.8
1⅛	17.0	34.0	50.9	67.9	84.9	101.8	118.8	135.8
1¼	15.3	30.6	45.8	61.1	76.4	91.7	106.9	122.2
1⅜	13.9	27.8	41.7	55.6	69.5	83.3	97.2	111.1
1½	12.7	25.5	38.2	50.9	63.6	76.4	89.1	101.8
1¾	10.9	21.8	32.7	43.7	54.6	65.5	76.4	87.3
2	9.6	19.1	28.7	38.2	47.8	57.3	66.9	76.4
2¼	8.5	17.0	25.5	34.0	42.5	50.9	59.4	67.9
2½	7.6	15.3	22.9	30.6	38.2	45.8	53.5	61.1
2¾	6.9	13.9	20.8	27.8	34.7	41.7	48.6	55.6
3	6.4	12.7	19.1	25.5	31.8	38.2	44.6	50.9
3½	5.5	10.9	16.4	21.8	27.3	32.7	38.2	43.7
4	4.8	9.6	14.3	19.1	23.9	28.7	33.4	38.2
4½	4.2	8.5	12.7	17.0	21.2	25.5	29.7	34.0
5	3.8	7.6	11.5	15.3	19.1	22.9	26.7	30.6
5½	3.5	6.9	10.4	13.9	17.4	20.8	24.3	27.8
6	3.2	6.4	9.5	12.7	15.9	19.1	22.3	25.5
7	2.7	5.5	8.2	10.9	13.6	16.4	19.1	21.8
8	2.4	4.8	7.2	9.6	11.9	14.3	16.7	19.1
9	2.1	4.2	6.4	8.5	10.6	12.7	14.8	17.0
10	1.9	3.8	5.7	7.6	9.6	11.5	13.3	15.3
11	1.7	3.5	5.2	6.9	8.7	10.4	12.2	13.9
12	1.6	3.2	4.8	6.4	8.0	9.5	11.1	12.7
13	1.5	2.9	4.4	5.9	7.3	8.8	10.3	11.8
14	1.4	2.7	4.1	5.5	6.8	8.2	9.5	10.9
15	1.3	2.5	3.8	5.1	6.4	7.6	8.9	10.2
16	1.2	2.4	3.6	4.8	6.0	7.2	8.4	9.5
18	1.1	2.1	3.2	4.2	5.3	6.4	7.4	8.5
20	1.0	1.9	2.9	3.8	4.8	5.7	6.7	7.6
22	.9	1.7	2.6	3.5	4.3	5.2	6.1	6.9
24	.8	1.6	2.4	3.2	4.0	4.8	5.6	6.4
26	.7	1.5	2.2	2.9	3.7	4.4	5.1	5.9
28	.7	1.4	2.0	2.7	3.4	4.1	4.8	5.5
30	.6	1.3	1.9	2.5	3.2	3.8	4.5	5.1
36	.5	1.1	1.6	2.1	2.7	3.2	3.7	4.2
42	.5	.9	1.4	1.8	2.3	2.7	3.2	3.6
48	.4	.8	1.2	1.6	2.0	2.4	2.8	3.2
54	.4	.7	1.1	1.4	1.8	2.1	2.5	2.8
60	.3	.6	1.0	1.3	1.6	1.9	2.2	2.5

Speed of Cutting with Turret Lathes.—Jones & L chine Co. give the following cutting-speeds for use with their lathe:

		Ft. p
Threading	Tool steel and taper on tubing.	
	Machinery.	
	Very soft steel..................	
Turning machinery steel	Cut which reduces the stock to ½ of its original dian	
	Cut which reduces the stock to ¾ of its original diam	
	Cut which reduces the stock to ⅞ of its original diar	
	Cut which reduces the stock to 15/16 of its original di	
Turning very soft machinery steel, light cut and cool work..		

Metal-cutting Tools.—"*Hutte*," the German Engi-
ook, gives the following cutting-angles for using least power:

	Top Rake.	Angle of Cutting-edge.
on........................	3°	51°
....	4°	51°
....	4°	66°

n Machinist comments on these figures as follows: We are ive the best nor even the generally used angles for tools, vary so much to suit different circumstances, such as degree f the metal being cut, quality of steel of which the tool is f cut, kind of finish desired, etc. The angles that cut with nditure of power are easily determined by a few experiments, ngles must be determined by good judgment, guided by expe- rly all cases, however, we think the best practical angles are ose given.

ions and descriptions of various forms of cutting-tools, see the Tools in App. Cyc. App. Mech., vol. ii., and in Modern

els.—Angle of cutting-faces (Joshua Rose): For cast steel, es; for gun-metal or brass, about 50 degrees; for copper and out 30 to 35 degrees.

Gearing Lathes for Screw-cutting. (Garvin Ma- ad from the lathe index the number of threads per inch cut , and multiply it by any number that will give for a product ndex; put this gear upon the stud, then multiply the number inch to be cut by the same number, and put the resulting gear

o cut 11½ threads per inch. We find on the index that 48 into ds per inch, then $6 \times 4 = 24$, gear on stud, and $11\frac{1}{2} \times 4 = 46$, Any multiplier may be used so long as the products include ng with the lathe. For instance, instead of 4 as a multiplier Thus, $6 \times 6 = 36$, gear upon stud, and $11\frac{1}{2} \times 6 = 69$, gear

Calculating Simple and Compound Gearing re is no Index. (*Am Mach.*)—If the lathe is simple- e stud runs at the same speed as the spindle, select some gear and multiply its number of teeth by the number of threads lead-screw, and divide this result by the number of threads cut. This will give the number of teeth in the gear for the esult is a fractional number, or a number which is not among and, then try some other gear for the screw. Or, select the ud first, then multiply its number of teeth by the number of ch to be cut, and divide by the number of threads per inch on . This will give the number of teeth for the gear on the lathe is compound, select at random all the driving-gears, umbers of their teeth together, and this product by the num- to be cut. Then select at random all the driven gears except the numbers of their teeth together, and this product by the eads per inch in the lead-screw. Now divide the first result by obtain the number of teeth in the remaining driven gear. Or, om all the driven gears. Multiply the numbers of their teeth this product by the number of threads per inch in the lead- select at random all the driving-gears except one. Multiply f their teeth together, and this result by the number of threads screw to be cut. Divide the first result by the last, to obtain teeth in the remaining driver. When the gears on the com- are fast together, and cannot be changed, then the driven one ice as many teeth as the other, or driver, in which case in the onsider the lead-screw to have twice as many threads per inch has, and then ignore the compounding entirely. Some lathes cted that the stud on which the first driver is placed revolves ast as the spindle. This can be ignored in the calculations by *umber of threads* of the lead-screw. If both the last condi- *nt ignore them in the calculations* by multiplying the number *inch in the lead-screw* by four. If the thread to be cut i- *r if the pitch of the* lead-screw is fractional, or if bo- *reduce the fractions* to a common denominator, *f these fractions* as if they equalled the pitch of th to be cut, and of the lead-screw, respectively. Then use that part o given above which applies to the lathe in question. For instance it is desired to cut a thread of 25/32-inch pitch, and the lead-sc threads per inch Then the pitch of the lead-screw will be ¼ inch equal to 8/32 inch. We now have two fraction, 25/32 and 8/32, an screws will be in the proportion of 25 to 8, and the gears can be f the above rule, assuming the number of threads to be cut to be 8 and those on the lead-screw to be 25 per inch. But this latter nu be further modified by conditions named above, such as a reduced the stud, or fixed compound gears. In the instance given, if the l had been 2½ threads per inch, then its pitch being 4/10 inch, we fractions 4/10 and 25/32, which, reduced to a common denomi 64/160 and 125/160, and the gears will be the same as if the lead-scre threads per inch, and the screw to be cut 64 threads per inch.

On this subject consult also "Formulas in Gearing," published & Sharpe Mfg. Co., and Jamieson's Applied Mechanics.

Change-gears for Screw-cutting Lathes.—There is uniformity among lathe-builders as to the change-gears provided cutting. W. R. Macdonald, in *Am. Mach.*, April 7, 1892, proposes t ing series, by which 33 whole threads (not fractional) may be cut b of only nine gears:

Screw.	Spindle.									Whole	
	20	30	40	50	60	70	110	120	130		
20	....	8	6	4 4/5	4	3 3/7	2 2/11	2	1 11/13	2	11
30	18	....	9	7 1/5	6	5 1/7	3 3/11	3	2 10/13	3	12
40	24	16	12	9 3/5	8	6 6/7	4 4/11	4	3 9/13	4	13
50	30	20	15	......	10	8 4/7	5 5/11	5	4 8/13	5	14
60	36	24	18	14 2/5	....	10 2/7	6 6/11	6	5 7/13	6	15
70	42	28	21	16 4/5	14	......	7 7/11	7	6 6/13	7	16
110	66	44	33	26 2/5	22	18 6/7	....	11	10 2/13	8	18
120	72	48	36	28 4/5	24	20 4/7	13 1/11	....	11 1/13	9	20
130	78	52	39	31 1/5	26	22 3/7	14 2/11	13	......	10	21

Ten gears are sufficient to cut all the usual threads, with the exc perhaps 11½, the standard pipe-thread; in ordinary practice any f thread between 11 and 12 will be near enough for the customary sh thread: if not, the addition of a single gear will give it.

In this table the pitch of the lead-screw is 12, and it may be objec too fine for the purpose. This may be rectified by making the rea or any other desirable pitch, and establishing the proper ratio bet lathe spindle and the gear-stud.

Metric Screw-threads may be cut on lathes with inch-div ing-screws, by the use of change wheels with 50 and 127 teeth centimetres = 50 inches (127 × 0.3937 = 49.9999 in.).

Rule for Setting the Taper in a Lathe. (*Am. M* rule can be given which will produce exact results, owing to the the centres enter the work an indefinite distance. If it were not fo cumstance the following would be an exact rule, and it is an appr as it is. To find the distance to set the centre over: Divide the diff the diameters of the large and small end of the taper by 2, and mul quotient by the ratio which the total length of the shaft bears to t of the tapered portion. Example: Suppose a shaft three feet long a taper turned on the end one foot long, the large end of the taper inches and the small end one inch diameter. $\frac{2-1}{2} \times \frac{3}{1} = 1\frac{1}{2}$ inche

Electric Drilling-machines - Speed of Drilling H Steel Plates. (Proc. Inst. M. E., Aug. 1887, p. 329.)—In drilling the shell of the S.S. "Albania," after a very small amount of pra men working the machines drilled the ⅞-inch holes in the shell w rapidity, doing the work at the rate of one hole every 68 seconds, the time occupied in altering the position of the machines by usi ential pulley-blocks, which were not conveniently arranged th ... Repeated trials of these drilling-machines sh ...ing electrical energy in both holding-on magn to about ¾ H.P., they have drilled holes of 1 inch diameter ¾ inch thickness of solid wrought iron, or through 1¾ inch of mild o plates of 13/16 inch each, taking exactly 1¾ minutes for each

of Twist-drills.—The cutting-speeds and rates of feed recom- the Morse Twist-drill and Machine Company are given in the table.

ons per minute for drills 1/16 in. to 2 in. diam., as usually applied:

Speed for Steel.	Speed for Iron.	Speed for Brass.	Diameter of Drills.	Speed for Steel.	Speed for Iron.	Speed for Brass.
			inch.			
940	1280	1560	1 1/16	54	75	95
460	660	785	1⅛	52	70	90
316	420	540	1 3/16	49	66	85
230	320	400	1¼	46	62	80
190	260	320	1 5/16	44	60	75
150	220	260	1⅜	42	58	72
130	185	230	1 7/16	40	56	69
115	160	200	1½	39	54	66
100	140	180	1 9/16	37	51	63
95	130	160	1⅝	36	49	60
85	115	145	1 11/16	34	47	58
75	105	130	1¾	33	45	56
70	100	120	1 13/16	32	43	54
65	90	115	1⅞	31	41	52
62	85	110	1 15/16	30	40	51
58	80	100	2	29	39	49

one inch in soft cast iron will usually require: For ¼-in. drill, 125 ns; for ½-in drill, 120 revolutions; for ¾-in. drill, 100 revolutions; rill, 95 revolutions.

es of feed for twist drills are thus given by the same company:

of drill...........	1/16	¼	⅜	½	¾	1	1½
inch depth of hole.	125	125	120 to 140		1 inch feed per min.		

MILLING-CUTTERS.

Addy, (Proc. Inst. M. E., Oct. 1890, p. 537), gives the following: **yses of Steel.**—The following are analyses of milling-cutter ade from best quality crucible cast steel and from self-hardening e" steel:

	Crucible Cast Steel, per cent.	Ivanhoe Steel, per cent.
on	1.2	1.67
n..........................	0.112	0.252
phorus	0.018	0.051
anese......................	0.36	2.557
ur.........................	0.02	0.01
sten	......	4.65
by difference	98.29	90.81
	100.000	100.000

st analysis is of a cutter 14 in. diam., 1 in. wide, which gave very vice at a cutting-speed of 60 ft. per min. Large milling-cutters are es built up, the cutting-edges only being of tool steel. A cutter 22 in. 5½ in. wide has been made in this way, the teeth being clamped two cast-iron flanges. Mr. Addy recommends for this form of with a cutting-angle of 70°, the face of the tooth being set 10° back al line on the cutter, the clearance-angle being thus 10°. At the *Iron-works, Leeds,* the face of the tooth is set 10° back of the radial *tting wrought iron* and 20° for steel.

of Teeth.—*For obtaining a suitable pitch of teeth for milling- arious diameters there exists no standard rule, the pitch* being *ided in an arbitrary* manner, according to individu

For estimating the pitch of teeth in a cutter of any diameter from [illegible] in., Mr. Addy has worked out the following rule, which he has found [illegible] ble of giving good results in practice:

$$\text{Pitch in inches} = \sqrt{(\text{diam. in inches} \times 8)} \times 0.0625 = .177 \sqrt{\text{diam.}}$$

J. M. Gray gives a rule for pitch as follows: The number of teeth [illegible] milling-cutter ought to be 100 times the pitch in inches; that is, if [illegible] were 27 teeth, the pitch ought to be 0.27 in. The rules are practically [illegible] same, for if d = diam., n = No. of teeth, p = pitch, c = circumference [illegible] pn; $d = \frac{pn}{\pi} = \frac{100p^2}{\pi} = 31.83p^2$; $p = \sqrt{.0314d} = .177\sqrt{d}$; No. of teeth [illegible] $3.14d \div p$.

Number of Teeth in Mills or Cutters. (Joshua Rose.)—The [illegible] of cutters must obviously be spaced wide enough apart to admit of the [illegible] wheel grinding one tooth without touching the next one, and the front [illegible] of the teeth are always made in the plane of a line radiating from the [illegible] the cutter. In cutters up to 3 in. in diam. it is good practice to give [illegible] teeth per in. of diam., while in cutters above that diameter the [illegible] may be coarser, as follows:

Diameter of cutter,	6 in.; number of teeth in cutter,	40
" " "	7 " " " " " "	45
" " "	8 " " " " " "	50

Speed of Cutters.—The cutting speed for milling was originally [illegible] very low; but experience has shown that with the improvements [illegible] use it may with advantage be considerably increased, especially with [illegible] of large diameter. The following are recommended as safe speeds for [illegible] ters of 6 in. and upwards, provided there is not any great depth of [illegible] to cut away:

	Steel.	Wrought iron.	Cast iron.	Brass
Feet per minute..	36	48	60	12[illegible]
Feed, inch per min....	½	1	1⅝	[illegible]

Should it be desired to remove any large quantity of material, the [illegible] cutting-speeds are still recommended, but with a finer feed. A simple [illegible] for cutting-speed is: Number of revolutions per minute which the [illegible] spindle should make when working on cast iron = 240, divided by the [illegible] eter of the cutter in inches.

Speed of Milling-cutters. (Proc. Inst. M. E., April, 1883, p. [illegible] The cutting-speed which can be employed in milling is much greater [illegible] that which can be used in any of the ordinary operations of turning [illegible] lathe, or of planing, shaping, or slotting. A milling-cutter with a [illegible] supply of oil, or soap and water, can be run at from 80 to 100 ft. per [illegible] when cutting wrought iron. The same metal can only be turned in a [illegible] with a tool-holder having a good cutter, at the rate of 30 ft. per min., [illegible] about one third the speed of milling. A milling-cutter will cut cast [illegible] the rate of 25 to 30 ft. per min.

The following extracts are taken from an article on speed and feed [illegible] milling-cutters in *Eng'g*, Oct. 22, 1891: Milling-cutters are successfully [illegible] ployed on cast iron at a speed of 250 ft. per min.; on wrought iron at [illegible] 80 ft. to 100 ft. per min. The latter materials need a copious supply of [illegible] lubricant, such as oil or soapy water. These rates of speed are not [illegible] proached by other tools. The usual cutting-speeds on the lathe, [illegible] shaping, and slotting machines rarely exceed about one third of those [illegible] above, and frequently average about a fifth, the time lost in back strokes being reckoned.

The feed in the direction of cutting is said by one writer to vary, [illegible] nary work, from 40 to 70 revs. of a 4-in. cutter per in. of feed. It must [illegible] to an extent depend on the character of the work done, but the above [illegible] shavings of extreme thinness. For example, the circumference of a [illegible] cutter being, say, 12½ in., and having, say, 60 teeth, the advance [illegible] sponding to the passage of one cutting-tooth over the surface, in the [illegible] of the above-named feed-motions, is $1/40 \times 1/60 = 1/2400$ in.; the [illegible] gives an advance for each tooth of only $1/70 \times 1/60 = 1/4200$ in. [illegible] feeds as these are used only for light finishing cuts, and the same [illegible] ity recommends, also for finishing, a cutter about 9 in. in circum[illegible] nearly 3 in. in diameter, which should be run at about 60 revs. [illegible] cut tough [illegible]ht steel, 120 for ordinary cast iron, about [illegible]

40 to 160 for the various qualtities of gun-metal and brass.
naller or larger the rates of revolution are increased or
cord with the following table, which gives these rates of
nd shows the lineal speed of the cutting-edge:

	Steel.	Wrought Iron.	Cast Iron.	Gun-metal.	Brass.
...	45	60	90	105	120

re intended for very light finishing cuts, and they must be
t one half for heavy cutting.
results have been found to be the highest that could be at-
ry workshop routine, having due consideration to economy
en to change and grind the cutters when they become dull:
36 ft. to 40 ft. per min.; depth of cut, 1 in.; feed, 5/8 in. per
steel—About 30 ft. per min.; depth of cut, 1/4 in.; feed, 3/4
ugh gun-metal—80 ft. per min.; depth of cut, 1/8 in.; feed, 3/4
st-iron gear-wheels—26 1/8 ft. per min.; depth of cut, 1/8 in.;
nin. Hard, close-grained cast iron—30 ft. per min.; depth
feed, 5/16 in. per min. Gun-metal joints, 53 ft. per min.;
in.; feed, 5/8 in. per min. Steel-bars—21 ft. per min.; depth
feed, 3/4 in. per min.
ling-cutter, 4 in. in diam. and 12 in. wide, tested under two
eed in the same machine, gave the following results: The
stances was worked up to its maximum speed before it gave
being to ascertain definitely the relative amount of work
speed and a light feed, as compared with a low speed and a
machine was used single-geared and double-geared, and in
idth of cut was 10 1/8 in.
ft. per min.; 5/16 in. depth of cut; feed, 1.3 in. per min. =
nin. Double-gear, 19 ft. per min.; 3/8 in. depth of cut; feed,
= 2.40 cu. in. per min.
Results with Milling-machines. — Horace L.
ch., Dec. 28, 1893) gives the following results in flat-surface
d in a Pratt & Whitney milling-machine: The mills for the
diam., 12 teeth, 40 to 50 revs. and 4 7/8" feed per min. One
run over this piece at a feed of 9" per min., but the mills
at the end that this rate was greater than they could endure.
hese mills the figures are as follows, with 4 7/8" feed: Surface
arly; feed per tooth, 0.00812": cuts per inch, 123. And with
: Surface speed, 64 ft. per min.; feed per tooth, 0.015"; cuts

7/8" per min. the mills stood up well in this job of cast-iron
with a 9" feed they required grinding after surfacing one
words, it did not damage the mill-teeth to do this job with
of surface finished, but they would not endure 66 2/3 cuts per
st-iron milling the surface speed of the mills does not seem
of mill destruction: it is the increase of feed per tooth that
sed production of finished surface. This is precisely the re-
ion of single-pointed lathe and planer tools in general: with
is a surface-speed limit which cannot be economically ex-
uts, and so long as this surface-speed limit is not reached,
h or feed can be made anything up to the limit of the driv-
e lathe or planer, or to the safe strain on the work itself,
any cases be easily broken by a too great feed.
etal extreme figures were obtained in one experiment made
ays 5/16" wide by 1/8" deep in a bank of 8 shafts 1 1/4" diam.
att & Whitney No. 3 column milling-machine. The 8 mills
lly operated with 45 ft. surface speed and 19 1/8 in. per min.
s were 5" diam., with 28 teeth, giving the following figures,
e speed, 45 ft. per min.; feed per tooth, 0.02024"; cuts per
Fed with the revolution of mill. Flooded with oil, that is,
of oil running constantly over each mill. Face of tooth
ulting keyway was described as having a heavy wave or
he bottom, and it was said to have shown no signs of being
he cutters or on the machine. As a result of the experiment
r economical steady work to run at 17 revs., with a fee
ed cut, work fed with mill revolution, giving the foll
peed, 22 1/4 ft. per min.; feed per tooth, 0.0084"; cu

the teeth of the cutter strike on the top surface or "s
work in process of being milled, and "against the feed"
teeth begin to cut in the clean, newly cut surface of th
wards toward the scale—showed a decided advantage i
the cutter against the feed. The result is directly opposi
in tests of a Pratt & Whitney machine, by experts of the

In the tests with the Brown & Sharpe machine the cu
inches face by 4½ and 3 inches diameter respectively, 15
42 revolutions per minute in each case, or nearly 50 feet p
speed for the 4½-inch and 33 feet per minute for the 3-inc
lution marks were 6 to the inch, giving a feed of 7 inches
cut per tooth of .011″. When the machine was forced
driving the depth of cut was 11/32 inch when the cutte
way, or against the feed, and only ¼ inch when it ran i
or with the feed. The endurance of the milling-cutters
when they were run in the "old" way.

Spiral Milling-cutters.—There is no rule for fi
the spiral; from 10° to 15° is usually considered sufficien
the end thrust on the spindle will be increased to an exte
some machines.

Milling-cutters with Inserted Teeth.—Wh
use milling-cutters of a greater diameter than about 8 in.
insert the teeth in a disk or head, so as to avoid the
solid cutters and the difficulty of hardening them, not
the risk of breakage in hardening them, but also on acco
in obtaining a uniform degree of hardness or temper.

Milling-machine versus Planer.—For co
work done by each see paper by J. J. Grant, Trans. A. S.
says: The advantages of the milling machine over the
among which are the following: Exact duplication of wo
duction — the cutting being continuous; cost of prod
machines can be operated by one workman, and he not
and cost of tools for producing a given amount of work.

POWER REQUIRED FOR MACHIN

Resistance Overcome in Cutting Metal. (
viii. 308.)—Some experiments made at the works of Wi
showed that the resistance in cutting steel in a lathe

re to remove it. The weight of metal removed per hour would be $\times .375 \times .26 \times 60 = 1082.8$ lbs. Our earlier form of 36" planer has ed with one tool on ¾" cut on work 200 lbs. of metal per hour, and " machine has more than five times its capacity. The total pulling of the planer is 45,000 lbs.

rse-power Required to Run Lathes. (J. J. Flather, *Am.* , April 23, 1891.)—The power required to do useful work varies with pth and breadth of chip, with the shape of tool, and with the nature ensity of metal operated upon; and the power required to run a ma- empty is often a variable quantity.

instance, when the machine is new, and the working parts have not ne worn or fitted to each other as they will be after running a few hs. the power required will be greater than will be the case after the ng parts have become better fitted.

other cause of variation of the power absorbed is the driving-belt; a belt will increase the friction, hence to obtain the greatest efficiency machine we should use wide belts, and run them just tight enough to ent slip. The belts should also be soft and pliable, otherwise power is umed in bending them to the curvature of the pulleys.

hird cause is the variation of journal-friction, due to slacking up or ening the cap-screws, and also the end-thrust bearing screw.

rtig's investigations show that it requires less total power to turn off a weight of metal in a given time than it does to plane off the same unt; and also that the power is less for large than for small diameters. following table gives the actual horse-power required to drive a lathe y at varying numbers of revolutions of main spindle.

HORSE-POWER FOR SMALL LATHES.

hout Back Gears.		With Back Gears.		Remarks.
s. of ndle min.	H.P. required to drive empty.	Revs. of Spindle per min.	H.P. required to drive empty.	
2.72 9.08 5.00	.145 .197 .310	14.6 24.33 38.42	.126 .141 .274	20" Fitchburg lathe.
7.4 5.0 8	.159 .259 .339	4.84 12.8 19 2	.132 .187 .230	Smallla the (13½"), Chemnitz. Germany. New machine.
4.6 2 3	.206 .339 .455	6.61 14.8 22.1	.157 .206 .249	17½" lathe do. New machine.
.8 .6 .2	.086 .210 .326	2.31 6.72 10.8	.035 .063 .087	26" lathe do.

$P._0$ = horse-power necessary to drive lathe empty, and N = number lutions per minute, then the equation for average small lathes is $= 0.095 + 0.0012N$.

he power necessary to drive the lathes empty when the back gears an average equation for lathes under 20" swing is

$$H.P._0 = 0.10 + 0.006N.$$

arger lathes vary so much in construction and detail that no general n be obtained which will give, even approximately, the power re- to run them, and although the average formula shows that at least rse-power is needed to *start the* small lathes, there are many Amer- thes under *20" swing working* on a consumption of less than .05 *ower.*

The amount of power required to remove metal in a machine is able within more accurate limits.

Referring to Dr. Hartig's researches, H.P.$_1$ = CW, where C is and W the weight of chips removed per hour.

Average values of C are .030 for cast-iron, .032 for wrought-ir steel.

The size of lathe, and, therefore, the diameter of work, has no effect on the cutting power. If the lathe be heavy, the cut can be and consequently the weight of chips increased, but the value of to be about the same for a given metal through several varyin lathes.

HORSE-POWER REQUIRED TO REMOVE CAST IRON IN A 20-INCH

(J. J. Hobart.)

Descriptive No.	Number of Trials.	Tool used.	Average Cutting-speed in feet per minute.	Depth of Cut in inches.	Average Breadth of Cut in inches.	Average H.P. required to remove Metal.	Average pounds Metal turned off
1	22	Side tool............	37.90	.125	.015	.342	[illegible]
2	15	Diamond........ ..	30.50	.125	.015	.218	[illegible]
3	17	Round nose.........	42.61	.125	.015	.252	[illegible]
4	2	Left-hand round nose...............	26.29	.125	.015	.237	[illegible]
5	4	Square-faced tool ½″ broad.........	25.82	.015	.125	.255	[illegible]
6	1	"	25.27	.048	.048	.200	[illegible]
7	1	"	25.64	.125	.015	.246	[illegible]

The above table shows that an average of .26 horse-power is r turn off 10 pounds of cast-iron per hour, from which we obtain th value of the constant C = .024.

Most of the cuts were taken so that the metal would be reduc diameter; with a broad surface cut and a coarse feed, as in No. 5, required per pound of chips removed in a given time was a maxi least power per unit of weight removed being required when the square, as in No. 6.

HORSE-POWER REQUIRED TO REMOVE METAL IN A 29-INCH LA

(R. H. Smith.)

Number of Experiments.	Metal.	Cutting-speed, ft. per min.	Depth of Cut, in.	Average Breadth of Cut, in.	Average H.P. required to remove Metal.	Average pounds Metal removed
4	Cast iron	12.7	.05	.046	.105	[illegible]
4	Cast iron	11.1	.135	.046	.217	[illegible]
2	Cast iron	12.85	.04	.038	.098	[illegible]
4	Wrought iron	9.6	.03	.046	.059	[illegible]
4	Wrought iron	9.1	.06	.046	.138	[illegible]
4	Wrought iron	7.9	.14	.046	.196	[illegible]
2	Wrought iron	9.35	.045	.038	.095	[illegible]
4	Steel	6.00	.02	.046	.063	[illegible]
4	Steel	5.8	.04	.046	.083	[illegible]
4	Steel	5.1	.06	.046	.108	[illegible]

ies of C, .017 and .019, obtained for cast iron are probably ms: the iron was soft and of fine quality, known as pulley less power to cut; and, as Prof. Smith remarks, a lower so takes less horse-power.

metals and forms of tools vary, otherwise the amount of per hour per horse-power would be practically constant, the peeds decreasing but slightly the visible work done.

count these variations, the weight of metal removed per by a certain constant, is equal to the power necessary to do

according to the above tests, is as follows:

	Cast Iron.	Wrought Iron.	Steel.
....	.030	.032	.047
....	.023	.028	.042
....	.024		
....	.026	.030	.044

cessary to run the lathe empty will vary from about .05 to .3 uld be ascertained and added to the useful horse-power, to power expended.

d by Machine-tools. (R. E. Dinsmore, from the *Elec-*

g 2 3/16″ × 180 ft. at 160 revs., carrying 26 pulleys to 36″, and running 20 idle machine belts........	1.32 H.P.
upright back-geared drill-press with table, 28″ g ⅜″ hole in cast iron, with a feed of 1 in. per	0.78 H.P.
drill grinder No. 2, carrying 2″ × 6″ wheels at 3200	0.29 H.P.
30″ × 36″, table 6 ft., planing cast iron, cut ¼″ 6 sq. in. per minute, at 9 reversals...	1.06 H.P.
hine 22″ stroke, cutting steel die, 6″ stroke, ⅛″ at rate of 1.7 square inch per minute....	0.37 H.P.
17″ swing, turning steel shaft 2⅜″ diam., cut 3/16 7.92 inch per minute........	0.43 H.P.
21″ swing, boring cast-iron hole 5″ diam., cut 3/16 0.3″ per minute........................	0.23 H.P.
o. 2, monogram blower at 1800 revs. per minute, ...	0.8 H.P.
r 28″ × 28″ × 14 ft. bed, stroke 8″, cutting steel, er minute.	3.2 H.P.

the next page compiled from various sources, principally esearches, by Prof. J. J. Flather (*Am. Mach.*, April 12, 1894), a guide in estimating the power required to run a given must be understood that these values, although determined ric measurements for the individual machines designated, rily representative, as the power required to drive a machine lent largely on its particular design and construction. The le work to be done may also affect the power required to a machine to be used exclusively for brass work may be 0% to 15% higher than if it were to be used for iron work of l the power required will be proportionately greater.

is to be transmitted to the machines by means of shafting fts, an additional amount, varying from 30% to 50% of the total l by the machines, will be necessary to overcome the friction

ver required to drive Shafting.—Samuel Webber, of Power" gives among numerous tables of power required machinery, a table of results of tests of shafting. A line of 42 ft. long, weighing 4098 lbs., with pulleys weighing 5331 lbs., *9 lbs., supported on* 47 bearings, 216 revolutions per min *P. to drive it.* This gives a coefficient of friction of *ts the coefficient* ranged from 3.34% to 11.4%, aver

Horse-power Required to Drive Machinery.

Name of Machine.	Observed Horse-p	
	Total Work.	Running
Small screw-cutting lathe 13½″ swing, B. G.	0.41	[illegible]
Screw-cutting lathe 17½″, B. G.	0.867	[illegible]
Screw-cutting lathe 20″ (Fitchburg), B. G.	0.47	[illegible]
Screw-cutting lathe 26″, B. G	0.462	[illegible]
Lathe, 80″ face plate, will swing 108″, T. G.	0.53	[illegible]
Large facing lathe, will swing 68″, T. G.	0.91	[illegible]
Wheel lathe 60″ swing		[illegible]
Small shaper (stroke 4″, traverse 11″)	0.16	[illegible]
Small shaper, Richards (9½″ × 22″)	0.24	[illegible]
Shaper (15″ stroke Gould & Eberhardt)	0.63	[illegible]
Large shaper, Richards (29″ × 91″)	1.14	[illegible]
Crank planer (capacity 23″ × 27″ × 28½″ stroke)	0.24	[illegible]
Planer (capacity 36″ × 36″ × 11 feet)	0.84	[illegible]
Large planer (capacity 76″ × 76″ × 57 feet	1.47	[illegible]
Small drill press	0.62	[illegible]
Upright slot drilling mach. (will drill 2½″ diam.)	0.41	[illegible]
Medium drill press	1.33	[illegible]
Large drill press	1.24	[illegible]
Radial drill 6 feet swing	0.53	[illegible]
Radial drill 8½ feet swing	0.67	[illegible]
Radial drill press	1.08	[illegible]
Slotter (8″ stroke)	0.28	[illegible]
Slotter (9½″ stroke)	0.44	[illegible]
Slotter (15″ stroke)	0.95	[illegible]
Universal milling mach (Brown & Sharpe No. 1)	0.28	[illegible]
Milling machine (13″ cutter-head, 12 cutters)	0.66	[illegible]
Small head traversing milling machine (cutter-head 11″ diameter, 16 cutters)	0.18	[illegible]
Gear cutter will cut 20″ diameter	0.28	[illegible]
Horizontal boring machine for iron, 23½″ swing	0.93	[illegible]
Hydraulic shearing machine	1.52	[illegible]
Large plate shears—knives 28″ long, 3″ stroke	7.12	[illegible]
Large punch press, over-reach 28″, 3″ stroke, 1½″ stock can be punched	4.41	[illegible]
Small punch and shear comb'd, 7½″ knives, 1½″ str.	0.79	[illegible]
Circular saw for hot iron (30½″ diameter of saw)	4.12	[illegible]
Plate-bending rolls, diam. of rolls 13″, length 9½ ft.	2.70	[illegible]
Wood planer 13½″ (rotary knives, 2 hor'l 2 vert.	4.24	[illegible]
Wood planer 24″ (rotary knives)	3.03	[illegible]
Wood planer 17½″ (rotary knives)	4.63	[illegible]
Wood planer 28″ (rotary knives)	5.00	[illegible]
Wood planer 28″ (Daniel's pattern)	3.30	[illegible]
Wood planer and matcher (capacity 14½ × 4¾″)	6.91	[illegible]
Circular saw for wood (23″ diameter of saw)	3.23	[illegible]
Circular saw for wood (35″ diameter of saw)	5.64	[illegible]
Band saw for wood (34″ band wheel)	0.96	[illegible]
Wood-mortising and boring machine	0.49	[illegible]
Hor'l wood-boring and mortising machine, drill 4″ diam., mortise 8½ deep × 11½″ long	3.68	[illegible]
Tenon and mortising machine	2.11	[illegible]
Tenon and mortising machine	2.73	[illegible]
Tenon and mortising machine	2.25	[illegible]
Edge-molder and shaper. (Vertical spindle)	2.06	[illegible]
Wood-molding mach. (cap. 7½ × 2½). Hor. spindle	2.45	[illegible]
Grindstone for tools, 31″ diam., 6″ face. Velocity 680 ft. per minute	1.32	[illegible]
Grindstone for stock, 42″ × 12″. Vel. 1680 ft. per min.	3.11	[illegible]
Emery wheel 11½″ diameter × ¼″. Saw grinder	0.34	[illegible]

* With back gears. † Without back gears. ‡ For surface [illegible]
B. G., back-geared. T. G., triple-geared.

er consumed in Machine-shops.—How much
l to drive ordinary machine-tools? and how many men can
horse-power? are questions which it is impossible to answer
. The power varies greatly according to the conditions in
following table given by J. J. Flather in his work on Dyna-
n idea of the variation in several large works. The percen-
power required to drive the shafting varies from 15 to 80,
of men employed per total H.P. varies from 0.62 to 6.04.

se-power; Friction; Men Employed.

irm.	Kind of Work.	Horse-power. Total.	Required to drive Shafting.	Required to drive Machinery.	Per cent to drive Shafting.	Number of Men.	No. of Men per Total H.P.	No. of Men per Effective H.P.
... .	E. & W. W.	58				132	2.27	
....	W. W.	100	15	85	15	300	3.00	3.53
ts.........	E., M. M.	400	95	305	23	1600	4.00	5.24
Brass W'ks	M. E., etc.	25	8	17	32	150	6.00	8.82
....	E.	95				230	2.42	
Works..	L.	2500	2000	500	80	4100	1.64	8.20
. (one de-	*							
....	H. M.	102	41	61	40	300	2.94	4.87
ool Co	M. T.	180	75	105	41	432	2.40	4.11
Co.......	"	120				725	6.04	
Co.......	"	230				900	3.91	
o..........	C. & L.	137	67	68	49	700	5.11	10.25
ne Co.....	P. & D.	35	11	24	31	90	2.57	3.75
s..........	P. & S.	12				30	2.50	
Co	H. F.	150	75	75	50	130	.86	1.73
....	S. M.	1300				3500	2.69	
....	"	350				1500	4.28	
Screw Co	M. S.	40				80	2.00	
" "	"	400	100	300	25	250	0.62	0.83
o..........	F.	350				400	1.14	
....	....	346.4			38.6%	818.3	2.96	5.13

E., engine; W.W., wood-working machinery; M. M., min-
M. E., marine engines; L., locomotives; H. M., heavy ma-
nachine tools; C. & L., cranes and locks; P. & D., presses
pulleys and shafting; H. F., heavy forgings; S. M., sewing-
machine-screws; F., files.

states (Trans. A. S. M. E., vi. 462) that in print-mills which
friction of the shafting and engine was in 7 cases below
es between 20% and 30%, in 11 cases from 30% to 35% and in 2
the average being 25.9%. Mr. Barrus in eight cotton-mills
to be between 18% and 25.7%, the average being 22%. Mr.
that for shops using heavy machinery the percentage of
o drive the shafting will average from 40% to 50% of the total
. This presupposes that under the head of shafting are
rs, fans, and blowers.

ABRASIVE PROCESSES.

g is performed by means of stones, sand,
undum, crocus, rouge, chilled globules of
le iron alone. (See paper by John Rich
ety of the Pacific Coast. Am. Mach.,
uly 25 and Aug. 15, 1891.)

cold saw to cutting iron or steel in the form of t
in which the piece to be cut is made to revolve
than the saw. By this means only a small surfa
presented at a time to the circumference of the s
same size as the cold saw above described, and
about 25,000 feet per minute. The heat generated
against the small surface of the bar rotated aga
particles of iron or steel in the bar are actually
welds as it falls into a solid mass. This disk will cu
iron, or steel. It will cut a bar of steel 1⅜ inch d
cluding the time of setting it in the machine, the
200 turns per minute.

Cutting Stone with Wire.—A plan of c
wire cord has been tried in Europe. While reta
agent, M. Paulin Gay, of Marseilles, has succeeded
ical means, and as continuously as formerly the
with both of which appliances his system—tha
cord "—has considerable analogy. An engine pu
wire cord (varying from five to seven thirty-secon
according to the work), composed of three mild-s
tain pitch, that is found to give the best results
from 15 to 17 feet per second.

The Sand-blast.—In the sand-blast, inven
Philadelphia, and first exhibited at the American
in 1871, common sand, powdered quartz, emery, c
rial is blown by a jet of air or steam on glass, met
brittle substance, by which means the latter is
To protect those portions of the surface which
abraded it is only necessary to cover them with
such as lead, rubber, leather, paper, wax, or rubb
in App. Cyc. Mech.; also U. S. report of Vienna E

A "jet of sand" impelled by steam of modera
blast of an ordinary fan, depolishes glass in a few
rapidly; and metals are given the so-called "fr
rapidity. With a jet issuing from under 300 p
cut through a piece of corundrum 1½ inches thic

The sand-blast has been applied to the clean

r. The same weight of small forgings and stampings can be scaled in 20 to 30 minutes.—*Iron Age*, March 8, 1894.

EMERY-WHEELS AND GRINDSTONES.

e Selection of Emery-wheels.—A pamphlet entitled "Emery-ls, their Selection and Use," published by the Brown & Sharpe Mfg. after calling attention to the fact that too much should not be expected e wheel, and commenting upon the importance of selecting the proper l for the work to be done, says:

eels are numbered from coarse to fine; that is, a wheel made of No. ery is coarser than one made of No. 100. Within certain limits, and things being equal, a coarse wheel is less liable to change the tem-ure of the work and less liable to glaze than a fine wheel. As a rule, arder the stock the coarser the wheel required to produce a given . For example, coarser wheels are required to produce a given sur-upon hardened steel than upon soft steel, while finer wheels are re-d to produce this surface upon brass or copper than upon either ened or soft steel.

eels are graded from soft to hard, and the grade is denoted by the rs of the alphabet, A denoting the softest grade. A wheel is soft or chiefly on account of the amount and character of the material com-l in its manufacture with emery or corundum. But other character-being equal, a wheel that is composed of fine emery is more compact harder than one made of coarser emery. For instance, a wheel of No. mery, grade B, will be harder than one of No. 60 emery, same grade.

e softness of a wheel is generally its most important characteristic. A wheel is less apt to cause a change of temperature in the work, or to me glazed, than a harder one. It is best for grinding hardened steel, iron, brass, copper, and rubber, while a harder or more compact wheel tter for grinding soft steel and wrought iron. As a rule, other things g equal, the harder the stock the softer the wheel required to produce ven finish.

enerally speaking, a wheel should be softer as the surface in contact h the work is increased. For example, a wheel 1/16-inch face should be der than one ½-inch face. If a wheel is hard and heats or chatters, it often be made somewhat more effective by turning off a part of its ting surface; but it should be clearly understood that while this will netimes prevent a hard wheel from heating or chattering the work, such wheel will not prove as economical as one of the full width and proper de, for it should be borne in mind that the grade should always bear the per relation to the width. (See the pamphlet referred to for other in-mation. See also lecture by T. Dunkin Paret, Pres't of The Tanite Co., Emery-wheels, Jour. Frank. Inst., March, 1890.)

Speed of Emery-wheels.—The following speeds are recommended different makers:

Revolutions per minute.				Diameter of Wheel, inches.	Revolutions per minute.			
Waltham E. W. Co.	The Tanite Co.	Grant Corundum Wheel Co.	Norton E. W. Co		Waltham E. W. Co.	The Tanite Co.	Grant Corundum Wheel Co.	Norton E. W. Co.
19,000	....	....	....	10	1,950	2,160	2,200	2,200
12,500	14,400	....	12,000	12	1,600	1,800	1,800	1,850
9,500	10,800	....	10,000	14	1,400	1,570	1,600	1,600
7,600	8,640	....	8,500	16	1,200	1,350	1,400	1,400
6,400	7,200	7,400	7,400	18	1,050	1,222	1,250	1,250
4,800	5,400	5,400	5,450	20	950	1,080	1,100	1,100
3,800	4,320	4,400	4,400	22	875	1,000	1,000	1,000
3,200	3,600	3,600	3,600	24	800	917	925	925
2,700	3,080	3,200	3,150	26	750	....	600	825
2,400	2,700	2,700	2,750	30	675	733	500	735
2,150	2,400	2,400	2,450	36	550	611	400	55[illegible]

"*e advise the regular speed of 5500 feet per minute.*" (Detroit Em *Co.*)

perience has demonstrated that there is no advantage in ru solid emery-wheels at a higher rate than 5500 feet per minute speed." (Springfield E. W. Mfg. Co.)

"Although there is no exactly defined limit at which a wheel to render it effective, experience has demonstrated that, taking safety, durability, and liability to heat, 5500 feet per minute at the gives the best results. All first-class wheels have the number of necessary to give this rate marked on their labels, and a column in the price-list gives a corresponding rate. Above this speed are unsafe. If run much below it they wear away rapidly in pro what they accomplish." (Northampton E. W. Co.)

Grades of Emery.—The numbers representing the grades run from 8 to 120, and the degree of smoothness of surface they be compared to that left by files as follows:

8 and 10	represent the cut of	a wood rasp.
16 " 20	" " " "	a coarse rough file.
24 " 30	" " " "	an ordinary rough file.
36 " 40	" " " "	a bastard file.
46 " 60	" " " "	a second-cut file.
70 " 80	" " " "	a smooth "
90 " 100	" " " "	a superfine "
120 F and FF	" " " "	a dead-smooth file.

Speed of Polishing-wheels.

Wood covered with leather, about	7000 ft. per
" " " a hair brush, about	2500 revs.
" " 1½″ to 8″ diam., hair 1″ to 1¼″ long, ab.	4500 "
Walrus-hide wheels, about.....	8000 ft. per
Rag-wheels, 4 to 8 in. diameter, about..................	7000 " "

Safe Speeds for Grindstones and Emery-wheel Hiscox (*Iron Age*, April 7, 1892), by an application of the formula ugal force in fly-wheels (see Fly-wheels), obtains the figures for grindstones and emery-wheels which are given in the tables be formulæ are:

Stress per sq. in. of section of a grindstone $= (.7071 D \times N)^2$

" " " " " " " an emery-wheel $= (.7071 D \times N)^2$

D = diameter in feet, N = revolutions per minute.

He takes the weight of sandstone at .078 lb. per cubic inch, and emery-wheel at 0.1 lb. per cubic inch; Ohio stone weighs about Huron stone about .089 lb. per cubic inch. The Ohio stone will be at the periphery of 2500 to 3000 ft. per min., which latter should exceeded. The Huron stone can be trusted up to 4000 ft., when clamped between flanges and not excessively wedged in setting from the speed of grindstones as a cause of bursting, probably the of accidents have really been caused by wedging them on the shaft wedging to true them. The holes being square, the excessive wedges to true the stones starts cracks in the corners that even out until the centrifugal strain becomes greater than the tenac remaining solid stone. Hence the necessity of great caution in wedges, as well as the holding of large quick-running stones betw flanges and leather washers.

Strains in Grindstones.

LIMIT OF VELOCITY AND APPROXIMATE ACTUAL STRAIN PER SQUAR SECTIONAL AREA FOR GRINDSTONES OF MEDIUM TENSILE STR

Diameter.	Revolutions per minute.					
	100	150	200	250	300	350
feet.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.
2	1.58	3.57	6.35	9.93	14.30	18.36
2½	2.47	5.57	9.88	15.49	22.29	28.64
3	3.57	8.04	14.28	22.34	32.16	....
3½	4.86	10.93	19.44	30.38	....	....
4	6.35	14.30	27.28	....	....	....
4½	8.04	18.08	32.16	Approximate break		
5	9.93	22.34	....	times the strain		
6	14.30	32.17	....	the bottom figure		
7	19.44	....	....			

figures at the bottom of columns designate the limit of velocity (in tions per minute), at the head of the columns for stones of the diam- the first column opposite the designating figure.

neral rule of safety for any size grindstone that has a compact and grain is to limit the peripheral velocity to 47 feet per second.

e is a large variation in the listed speeds of emery-wheels by different s—4000 as a minimum and 5600 maximum feet per minute, while claim a maximum speed of 10,000 feet per minute as the safe speed r best emery-wheels. Rim wheels and iron centre wheels are special- at require the maker's guarantee and assignment of speed.

Strains in Emery-wheels.

L STRAIN PER SQUARE INCH OF SECTION IN EMERY-WHEELS AT THE VELOCITIES AT HEAD OF COLUMNS FOR SIZES IN FIRST COLUMN.

Revolutions per minute.										
500	800	1000	1200	1400	1600	1800	2000	2200	2400	2600
......	......	......	......	......	......	......	22.67	27.43	32.64	38.31
......	......	......	......	......	......	......	51.13	61.86	73.62	86.40
......	......	22.67	32.65	44.45	58.05	73.47	90.71	109.76	130.62	153.30
......	......	35.47	51.08	69.51	90.81	114.94	141.90	171.71	......	......
18.40	32.72	51.12	73.62	100.21	130.88	165.65	......	......	......	......
24.80	43.90	68.70	99.21	134.65	175.60	......	......			
32.57	57.65	90.24	130.31	177.80	......	......	......			
41.41	73.62	115.03	165.65	......	......	......	......			
50.98	90.23	141.22	......	......	......	......	......			
61.81	109.41	171.23	......	......	......	......	......			
73.62	130.88	......	......	......	......	......	......			
86.36	152.85	......	......	......	......	......	......			
115.04	......	......	......	......	......	......	......			
165.64	......	......	......	......	......	......	......			

Diam	Revs. per min.	
in.	2800	3000
4	44.43	51.12
6	100.21	115.03
8	177.80	......

hua Rose (Modern Machine-shop Practice) says: The average speed of stones in workshops may be given as follows:

Circumferential Speed of Stone.

For grinding machinists' tools, about ...... 900 feet per minute.
" " carpenters' " " .... 600 " " "

e speeds of stones for file-grinding, and other similar rapid grinding is given in the "Grinders' List."

. ft.......	8	7½	7	6½	6	5½	5	4½	4	3½	3
per min.	135	144	154	166	180	196	216	240	270	308	360

e following table, from the *Mechanical World*, is for the diameter of s and the number of revolutions they should run per minute (not to be eded), with the diameter of change of shift-pulleys required, varying shift or change 2½ inches, 2¼ inches, or 2 inches in diameter for each ction of 6 inches in the diameter of the stone.

iameter f Stone.		Revolutions per minute.	Shift of Pulleys, in inches.		
			2½	2¼	2
t.	in.				
8	0	135	40	36	32
7	6	144	37½	33¾	30
7	0	154	35	31½	28
6	6	166	32½	29¼	26
6	0	180	30	27	24
5	6	196	27½	24¾	22
5	0	216	25	22½	20
4	6	240	22½	20¼	18
4	0	270	20	18	16
[illegible]	6	308	17½	15¾	14
	0	360	15	13½	12
		2	3	4	

Columns 3, 4, and 5 are given to show that if we start an 8-foot stone with, say, a countershaft pulley driving a 40-inch pulley on the grindstone spindle, and the stone makes the right number (135) of revolutions per minute, the reduction in the diameter of the pulley on the grinding-stone spindle, when the stone has been reduced 6 inches in diameter, will require to be also reduced 2½ inches in diameter, or to shift from 40 inches to 37½ inches, and so on similarly for columns 4 and 5. Any other suitable dimensions of pulleys may be used for the stone when eight feet in diameter, but the number of inches in each shift named, in order to be correct, will have to be proportional to the numbers of revolutions the stone should run, as given in column 2 of the table.

Varieties of Grindstones.

(Joshua Rose.)

FOR GRINDING MACHINISTS' TOOLS.

Name of Stone.	Kind of Grit.	Texture of Stone.	Color of Stone.
Nova Scotia,	All kinds, from finest to coarsest	All kinds, from hardest to softest	Blue or yellowish gray
Bay Chaleur (New Brunswick),	Medium to finest	Soft and sharp	Uniformly light blue
Liverpool or Melling.	Medium to fine	Soft, with sharp grit	Reddish

FOR WOOD-WORKING TOOLS.

Name of Stone.	Kind of Grit.	Texture of Stone.	Color of Stone.
Wickersley.........	Medium to fine	Very soft	Grayish yellow
Liverpool or Melling.	Medium to fine	Soft, with sharp grit	Reddish
Bay Chaleur (New Brunswick),	Medium to finest	Soft and sharp	Uniform light blue
Huron, Michigan ...	Fine	Soft and sharp	Uniform light blue

FOR GRINDING BROAD SURFACES, AS SAWS OR IRON PLATES.

Name of Stone.	Kind of Grit.	Texture of Stone.	Color of Stone.
Newcastle...........	Coarse to med'm	The hard ones	Yellow
Independence.........	Coarse	Hard to medium	Grayish white
Massillon	Coarse	Hard to medium	Yellowish white

TAP DRILLS.

Taps for Machine-screws. (The Pratt & Whitney Co.)

Approx. Diameter, fractions of an inch.	Wire Gauge.	No. of Threads to inch.	Approx. Diameter, fractions of an inch.	Wire Gauge.	No. of Threads to inch.
	No. 1	60, 72		No. 13	20, 24
	2	48, 56, 64	¼	14	16, 18, 20, 22,
	3	40, 48, 56		15	18, 20, 24
7/64	4	32, 36, 40	17/64	16	16, 18, 20, 22
	5	30, 32, 36, 40	9/32	18	16, 18, 20
9/64	6	30, 32, 36, 40		19	16, 18, 20
	7	24, 30, 32	5/16	20	16, 18, 20
5/32	8	24, 30, 32, 36, 40		22	16, 18
	9	24, 28, 30, 32	⅜	24	14, 16, 18
3/16	10	20, 22, 24, 30, 32		26	16
	11	22, 24		28	16
7/32	12	20, 22, 24		30	16

The Morse Twist Drill and Machine Co. gives the following table of the different sizes of drills that should be used when a full thread is tapped in a hole. The sizes given are practically correct.

(The Morse Twist Drill and Machine Co.)

Diam. of Tap.	No. Threads to inch.			Drill for V Thread.			Drill for U. S. S. Thread.			Diam. of Tap.	No. Threads to inch.		Drill for V Thread.		Drill for U. S. S. Thread.	
¼	16	18	20	5/32	5/32	11/64	.....	.....	3/16	1 ⅛	7	8	29/32	15/16	15/16	....
9/32	16	18	20	3/16	13/64	13/64	.....	.....	....	1 5/32	7	8	15/16	31/32	......	....
5/16	16	18	..	7/32	15/64	.....	.....	¼	....	1 3/16	7	8	31/32	1	......	....
11/32	16	18	..	¼	17/64	.....	.....	.....	....	1 7/32	7	8	1	1 1/32	......	....
⅜	14	16	18	¼	9/32	9/32	.....	9/32	....	1 ¼	7	..	1 1/32	......	1 1/16	....
13/32	14	16	18	19/64	21/64	21/64	.....	.....	....	1 9/32	7	..	1 1/16	......	......	....
7/16	14	16	..	21/64	11/32	.....	11/32	.....	....	1 5/16	7	..	1 3/32	......	......	....
15/32	14	16	..	23/64	⅜	.....	.....	.....	....	1 11/32	7	..	1 ⅛	......	......	....
½	12	13	14	⅜	25/64	25/64	.....	13/32	....	1 ⅜	6	..	1 ⅛	......	1 5/32	....
17/32	12	13	14	13/32	27/64	27/64	.....	.....	....	1 13/32	6	..	1 5/32	......	......	....
9/16	12	14	..	7/16	29/64	.....	7/16	.....	....	1 7/16	6	..	1 5/32	......	......	....
19/32	12	14	..	15/32	31/64	.....	.....	.....	....	1 15/32	6	..	1 3/16	......	......	....
⅝	10	11	12	15/32	½	½	.....	½	....	1 ½	6	..	1 15/64	......	1 9/32	....
21/32	10	11	12	½	17/32	17/32	.....	.....	....	1 17/32	6	..	1 9/32	......	......	....
11/16	11	12	..	9/16	9/16	.....	.....	.....	....	1 9/16	6	..	1 9/32	......	......	....
23/32	11	12	..	19/32	19/32	.....	.....	.....	....	1 19/32	6	..	1 5/16	......	......	....
¾	10	11	12	19/32	⅝	⅝	⅝	.....	....	1 ⅝	5	5½	1 9/32	1 5/16	......	1⅜
25/32	10	11	12	⅝	21/32	21/32	.....	.....	....	1 21/32	5	5½	1 5/16	1 11/32	......	....
13/16	10	..	..	21/32	.....	.....	.....	.....	....	1 11/16	5	5½	1 11/32	1 ⅜	......	....
27/32	10	..	..	11/16	.....	.....	.....	.....	....	1 23/32	5	5½	1 ⅜	1 13/32	......	....
⅞	9	10	..	45/64	23/32	.....	23/32	.....	....	1 ¾	5	..	1 13/32	......	1½	....
29/32	9	10	..	47/64	¾	.....	.....	.....	....	1 25/32	5	..	1 7/16	......	......	....
15/16	9	..	..	49/64	.....	.....	.....	.....	....	1 13/16	5	..	1 15/32	......	......	....
31/32	9	..	..	51/64	.....	.....	.....	.....	....	1 27/32	5	..	1 ½	..	......	....
1	8	..	..	13/16	.....	.....	27/32	.....	....	1 ⅞	4½	5	1 17/32	1 17/32	......	1⅝
1 1/32	8	..	..	53/64	.....	.....	.....	.....	....	1 29/32	4½	5	1 9/16	1 9/16	......	....
1 1/16	8	..	..	55/64	.....	.....	.....	.....	....	1 15/16	4½	5	1 19/32	1 19/32	......	....
1 3/32	8	..	..	57/64	.....	....	.....	.....	....	1 31/32	4½	5	1 ⅝	1 ⅝	......	....
										2	4½	..	1 21/32	......	1 23/32	....

o inches across the hole of 5.2%, ave 63,920 lbs., and 10.6% elonga-

for punches for use in metal of punch. This form is of great- rked is less than two thirds the

wing-press. Oberlin Smith nethods of finding the size of d consists simply in a series of roper one is found. This is for the cutting portions of the di- her work is done. The second nd then, knowing the weight of the diameter of a piece having ight. The third method is by + $4dh$ for sharp-cornered cup. of cup, h = height of cup. For l, say radius of corner less than + $4dh$) − r, about; r being the assumption that the thickness ing operation.

of the Drop-press. R. H. copper cylinders was prepared. jected to the action of press- of fall. Companion specimen- me amount, and measures were n, and of the amount of work lrop. Comparing one with the e hammer was 9% of the work ficiency. That is to say, [illegible] qual to that due the weight of

ht of drop × fall × efficiency

compression.

e mean area opposed to crush-

our. Frank. Inst., March, [illegible] gh iron blocks 1¾ inches thick. only 1 1/16 inch thick, and its the hole. Therefore, 6% of the into the block itself, increasing

KING FITS.

by Hydraulic Pressure. larger than the hole into which pressure of 30 to 35 tons. (Lec-

e driving-wheel, when the pin- ould be pressed in with a pres- he wheel fit. When the hole is t of shrinking the tire on the has been bored, or if the hole is e to be increased to 9 tons for m. Machinist.)

an Railway Master Mechanics' rinkage allowances for tires of ly heated by gas-flames, slipped ol. The centres are turned to ires are bored smaller by the n:

4	50	56	62	66
7	.053	.060	.066	.070

y 1/80 inch per foot, or 1/960. A odulus of elasticity of steel at

30,000,000, the strain caused by shrinkage would be 30,000 lbs. per sq... which is well within the elastic limit of machinery steel.

SCREWS, SCREW-THREADS, ETC.[1]

Efficiency of a Screw.—Let α = angle of the thread, ... angle whose tangent is the pitch of the screw divided by the circ... of a circle whose diameter is the mean of the diameters at the to... bottom of the thread. Then for a square thread

$$\text{Efficiency} = \frac{1 - f \tan \alpha}{1 + f \operatorname{cotan} \alpha},$$

in which f is the coefficient of friction. (For demonstration, see Cotterill & Slade, Applied Mechanics, p. 146.) Since cotan = 1 ÷ tan, we may sub... for cotan α the reciprocal of the tangent, or if p = pitch, and c = ci... cumference of the screw,

$$\text{Efficiency} = \frac{1 - f\dfrac{p}{c}}{1 + f\dfrac{c}{p}}.$$

EXAMPLE.—Efficiency of square-threaded screws of ½ in. pitch.

Diameter at bottom of thread, in....	1	2	3
" " top " " "....	1½	2½	3½
Mean circumference " " "....	3.927	7.069	10.21
Cotangent $\alpha = c \div p$............... =	7.854	14.14	20.42
Tangent $\alpha = p \div c$................. =	.1273	.0661	.0490
Efficiency if $f = .10$............ ... =	55.3%	41.2%	32.7%
" " $f = .15$................ =	45%	31.7%	24.4%

The efficiency thus increases with the steepness of the pitch.

The above formulæ and examples are for square-threaded screws ... consider the friction of the screw-thread only, and not the friction ... collar or step by which end thrust is resisted, and which further reduces ... efficiency. The efficiency is also further reduced by giving an inclination ... the side of the thread, as in the V-threaded screw. For discussion ... subject, see paper by Wilfred Lewis, Jour. Frank. Inst. 1880; also ... A. S. M. E., vol. xii. 784.

Efficiency of Screw-bolts.—Mr. Lewis gives the following ap... imate formula for ordinary screw-bolts (V threads, with collar... pitch of screw, d = outside diameter of screw, F = force applied at c... ference to lift a unit of weight, E = efficiency of screw. For an a... case, in which the coefficient of friction may be assumed at .15,

$$F = \frac{p + d}{3d}, \qquad E = \frac{p}{p + d}.$$

For bolts of the dimensions given above, ½-in. pitch, and outside diam... eters 1½, 2½, 3½, and 4½ in., the efficiencies according to this ... would be, respectively, .25, .167, .125, and .10.

James McBride (Trans. A. S. M. E., xii. 781) describes an experiment ... an ordinary 2-in. screw-bolt, with a V thread, 4½ threads per inch, ... a weight of 7500 lbs., the force being applied by turning the nut. ... power applied 89.8% was absorbed by friction of the nut on its ... washer and of the threads of the bolt in the nut. The nut was ... and had the flat side to the washer.

Prof. Ball in his "Experimental Mechanics" says: "Experiments ... in two cases respectively about ⅔ and ¾ of the power was lost."

Trautwine says: "In practice the friction of the screw (which ... heavy loads becomes very great) make the theoretical calculations ... little value."

Weisbach says: "The efficiency is from 19% to 30%."

Efficiency of a Differential Screw.—A correspondent ... *American Machinist* describes an experiment with a differential ... punch, consisting of an outer screw 2 in. diam., 3 threads per in., ... *inner screw 1½ in. diam., 3½ threads per inch*. The pitch of the outer ...

[1] For U. S. Standard Screw-threads, see page 894.

inner screw 2/7 in., the punch would ad-
/7 = 1/21 in. Experiments were made to de-
punch an 11/16-in. hole in iron ¼ in. thick, the
of a lever-arm of 47¾ in. The leverage would
mean force applied at the end of the lever
he punch, if there was no friction, would be
orce required to punch the iron, assuming a
lbs. per sq. in., would be 50,000 × 11/16 × π ×
ency of the punch would be 27,000 ÷ 598,500 =
rew only used as a punch the mean force at
ly 82 lbs. The leverage in this case was 47¾ ×
eferred to the punch, including friction, 900 ×
y 27,000 ÷ 73,800 = 36.7%. The screws were of
ricated with lard-oil and plumbago.

-thread.—A. M. Powell (*Am. Mach.*, Jan. 24,
w-thread to replace the square form of thread,
ter ease in making fits, and provision for "take
dimensions are the same as those of square-
tion that the sides of the thread, instead of
is of the screw, are inclined 14½° to such per-
ides of a thread are inclined 29° to each other.
of the thread are the following: Depth of
f top of thread = width of space at bottom =
ot of thread = width of space at top = .6293 ÷
number of threads to the inch.

RTS OF MACHINES IN A SERIES OF SIZES.

s Indicator, April, 1892.)

sed by Coleman Sellers while at William Sellers
ns of the parts of machines, based upon the
ge machine and a small one to any series of
sed in getting up the proportion-book and ar-
s from which any machine can be constructed
the largest and smallest of the series.

nstruction Formulæ.—Take difference
the largest and the smallest machines that
me construction. Take also the difference be-
rts on the largest and smallest machines se-
the former, and the result obtained will be a
y the nominal capacity of the intermediate
minished by a constant "increment," will give
To find the "increment:" Multiply the nomi-
ize by the factor obtained, and subtract the
part belonging to the machine of nominal ca-

e of a part of a 72-in. machine is 3 in., and the
in. machine is 1⅞, or 1.875 in.: then 72 − 42 =
n. = 1.125. 1.125 ÷ 30 = .0375 = the "factor,"
1.875 − 1.575 = .3 = the "increment" to be
capacity; then the formula will read: x =

875, or 1⅞, the size of one of the selected parts.
$aD + c = x$, in which D = nominal capacity in
nstant increment, a is the factor, and x = the

KEYS.

-gearing. (Trans. A. S. M. E., xiii, 229.)—E.
f key = ⅛ diam. of shaft, depth = 1/9 diam. of

ills: Keys of square section, side = ¼ diam. of
in even sixteenths of an inch.
th = ⅛ diam. of hole; depth of side

key for 1 to 1¼ in. shafts, 5/16
½ to 1¾ in. shafts, and so on.
arge end of splice, 4/5 width of

Unwin (Elements of Machine Design) gives: Width $= \frac{1}{4}d + \frac{1}{8}$ in. Thickness $= \frac{1}{8}d + \frac{1}{8}$ in., in which d = diam. of shaft in inches. When wheels or pulleys transmitting only a small amount of power are keyed on large shafts, he says, these dimensions are excessive. In that case, if H.P. = horse-power transmitted by the wheel or pulley, N = revs. per min, P = force acting at the circumference, in lbs., and R = radius of pulley in inches, take

$$d = \sqrt[3]{\frac{100\,\text{H.P.}}{N}} \quad \text{or} \quad \sqrt[3]{\frac{PR}{630}}.$$

Prof. Coleman Sellers (*Stevens Indicator*, April, 1892) gives the following: The size of keys, both for shafting and for machine tools, are the proportions adopted by William Sellers & Co., and rigidly adhered to during a period of nearly forty years. Their practice in making keys and fitting them is, that the keys shall always bind tight sidewise, but not top and bottom; that is, not necessarily touch either at the bottom of the key-seat in the shaft or touch the top of the slot cut in the gear-wheel that is fastened to the shaft; but in practice keys used in this manner depend upon the fit of the wheel upon the shaft being a forcing fit, or a fit that is so tight as to require screw-pressure to put the wheel in place upon the shaft.

Size of Keys for Shafting.

Diameter of Shaft, in.	Size of Key, in.
1¼ 1 7/16 1 11/16	5/16 × ⅜
1 15/16 2 3/16	7/16 × ½
2 7/16	9/16 × ⅝
2 11/16 2 15/16 3 3/16 3 7/16	11/16 × ¾
3 15/16 4 7/16 4 15/16	13/16 × ⅞
5 7/16 5 15/16 6 7/16	15/16 × 1
6 15/16 7 7/16 7 15/16 8 7/16 8 15/16	1 1/16 × 1⅛

Length of key-seat for coupling = 1½ × nominal diameter of shaft.

Size of Keys for Machine Tools.

Diam. of Shaft, in.	Size of Key, in. sq.	Diam. of Shaft, in.	Size of Key, sq. in.
15/16 and under	⅛	4 to 5 7/16	13/16
1 to 1 3/16	3/16	5½ to 6 15/16	15/16
1¼ to 1 7/16	¼	7 to 8 15/16	1 1/16
1½ to 1 11/16	5/16	9 to 10 15/16	1 3/16
1¾ to 2 3/16	7/16	11 to 12 15/16	1 5/16
2¼ to 2 11/16	9/16	13 to 14 15/16	1 7/16
2¾ to 3 15/16	11/16		

John Richards, in an article in *Cassier's Magazine*, writes as follows: There are two kinds or system of keys, both proper and necessary, but widely different in nature. 1. The common fastening key, usually made in width one fourth of the shaft's diameter, and the depth five eighths to one third the width. These keys are tapered and fit on all sides, or, as it is commonly described, "bear all over." They perform the double function in most cases of driving or transmitting and fastening the keyed-on member against movement endwise on the shaft. Such keys, when properly made, drive as a strut, diagonally from corner to corner.

2. The other kind or class of keys are not tapered and fit on their sides only, a slight clearance being left on the back to insure against wedge action or radial strain. These keys drive by shearing strain.

For fixed work where there is no sliding movement such keys are commonly made of square section, the sides only being planed, so the depth is more than the width by so much as is cut away in finishing or fitting.

For sliding bearings, as in the case of drilling-machine spindles, the depth should be increased, and in cases where there is heavy strain there should be two keys or feathers instead of one.

The following tables are taken from proportions adopted in practical use.

Flat keys, as in the first table, are employed for fixed work when the parts are to be held not only against torsional strain, but also against movement endwise; and in case of heavy strain the strut principle being the strongest and most secure against movement when there is strain each way, as in the case of engine cranks and first movers generally. The objections

s are, straining the work out of truth, the care
ing, and destroying the evidence of good or bad
When a wheel or other part is fastened with a
ere is no means of knowing whether the work is
reason such keys are not employed by machine-
e of accurate work of any kind, indeed, cannot
strain, and also the difficulty of inspecting com-

IONS OF FLAT KEYS, IN INCHES.

1¼	1½	1¾	2	2½	3	3½	4	5	6	7	8
5/16	⅜	7/16	½	⅝	¾	⅞	1	1⅛	1⅜	1½	1¾
3/16	¼	9/32	5/16	⅜	7/16	½	⅝	11/16	13/16	⅞	1

IONS OF SQUARE KEYS, IN INCHES.

1¼	1½	1¾	2	2½	3	3½	4
7/32	9/32	11/32	13/32	15/32	17/32	9/16	11/16
¼	5/16	⅜	7/16	½	9/16	⅝	¾

OF SLIDING FEATHER-KEYS, IN INCHES.

[illegible]	1¾	2	2¼	2½	3	3½	4	4½
[illegible]	5/16	5/16	⅜	⅜	½	9/16	9/16	⅝
[illegible]	7/16	7/16	½	½	⅝	¾	¾	⅞

following table of dimensions to the *Am. Machin-*
heavy work and very short hubs we put in two
t. With special long hubs, where we cannot use
should be thicker than the standard.

...th, ...es.	Thickness, in.	Diameter of Shafts, inches.	Width, inches.	Thickness, in.
16	3/16	3 7/16 to 3 11/16	⅞	⅝
16	¼	3 15/16 to 4 3/16	1	11/16
	5/16	4 7/16 to 4 11/16	1⅛	¾
	⅜	4⅞ to 5⅜	1¼	15/16
	½	5⅞ to 6⅜	1½	1
	9/16	6⅞ to 7⅜	1¾	1⅛

es, say 14 to 16″, 1/16″ thicker; keys longer than
″ thicker; and so on. Special short hubs to have

Woodruff system of keying, see circular of the
Modern Mechanism, page 455.

ER OF KEYS AND SET-SCREWS.

ling-power of Set-screws in Pulleys.
M. E., x. 230.)—These tests were made by using a
aft by two set-screws with the shaft keyed to the
quired at the rim of the pulley to cause it to slip
being multiplied by the number 6.037 (obtained by
he pulley one-half the diameter of the wire rope,
wice the radius of the shaft, since there were two
time) gives the holding-power of the set-screws.
of wrought-iron, ⅝ of an inch in diameter, and ten
shaft used was of steel and rather hard, the set-
impression upon it. They were set
d of a ten-inch monkey-wrench. T
marked respectively A, B, C, and D.

A, ends perfectly flat, 9/16-in. diameter,	1412 to 2294 lbs.;	average 2064.
B, radius of rounded ends about ½ inch,	2747 " 3079 "	" 2912.
C, " " " " " ¼ "	1902 " 3079 "	" 2573.
D ends cup-shaped and case-hardened,	1962 " 2958 "	" 2470.

REMARKS.—A. The set-screws were not entirely normal to the shaft; hence they bore less in the earlier trials, before they had become flattened by wear.

B. The ends of these set-screws, after the first two trials, were found to be flattened, the flattened area having a diameter of about ¼ inch.

C. The ends were found, after the first two trials, to be flattened, as in B.

D. The first test held well because the edges were sharp, then the holding-power fell off till they had become flattened in a manner similar to B, when the holding-power increased again.

Tests of the Holding-power of Keys. (Lanza.)—The load was applied as in the tests of set-screws, the shaft being firmly keyed to the holders. The load required at the rim of the pulley to shear the keys was determined, and this, multiplied by a suitable constant, determined in a similar way to that used in the case of set-screws, gives us the shearing strength per square inch of the keys.

The keys tested were of eight kinds, denoted, respectively, by the letters A, B, C, D, E, F, G and H, and the results were as follows: A, B, D and F, each 4 tests; E, 3 tests; C, G, and H, each 2 tests.

A, Norway iron, 2″ × ¼″ × 15/32″,	40,184 to 47,760 lbs.;	average, 42,726.
B, refined iron, 2″ × ¼″ × 15/32″,	36,482 " 39,254;	" 38,059.
C, tool steel, 1″ × ¼″ × 15/32″,	91,344 & 100,056.	
D, machinery steel, 2″ × ¼″ × 15/32″,	64,630 to 70,186;	" 66,875.
E, Norway iron, 1⅛″ × ⅜″ × 7/16″,	36,850 " 37,222;	" 37,036.
F, cast-iron, 2″ × ¼″ × 15/32″,	30,278 " 36,944;	" 33,034.
G, cast-iron, 1⅛″ × ⅜″ × 7/16″,	37,222 & 38,700.	
H, cast-iron, 1″ × ½″ × 7/16″,	29,814 & 38,978.	

In A and B some crushing took place before shearing. In E, the keys being only 7/16 in. deep, tipped slightly in the key-way. In H, in the first test, there was a defect in the key-way of the pulley.

DYNAMOMETERS.

Dynamometers are instruments used for measuring power. They are of several classes, as: 1. Traction dynamometers, used for determining the power required to pull a car or other vehicle, or a plough or harrow. 2. Brake or absorption dynamometers, in which the power of a rotating shaft or wheel is absorbed or converted into heat by the friction of a brake; and, 3. Transmission dynamometers, in which the power in a rotating shaft is measured during its transmission through a belt or other connection to another shaft, without being absorbed.

Traction Dynamometers generally contain two principal parts: (1) A spring or series of springs, through which the pull is exerted, the extension of the spring measuring the amount of the pulling force; and (2) a paper-covered drum, rotated either at a uniform speed by clockwork, or at a speed proportional to the speed of the traction, through gearing, on which the extension of the spring is registered by a pencil. From the average height of the diagram drawn by the pencil above the zero-line the average pulling force in pounds is obtained, and this multiplied by the distance traversed, in feet, gives the work done, in foot-pounds. The product divided by the time in minutes and by 33,000 gives the horse-power.

The Prony brake is the typical form of absorption dynamometer. (See Fig. 167, from Flather on Dynamometers and the Measurement of Power.)

Primarily this consists of a lever connected to a revolving shaft or pulley in such a manner that the friction induced between the surfaces in contact will tend to rotate the arm in the direction in which the shaft revolves. This rotation is counterbalanced by weights P, hung in the scale pan at the end of the lever. In order to measure the power for a given number of revolutions of pulley, we add weights to the scale-pan and screw up on bolts *bb*, until the friction induced balances the weights and the lever is maintained revolutions of shaft per minute remain

enerally omitted—the friction being mea-
p thrown over the pulley. Ropes or cords
se.
scale-pan, as in Fig. 167, the friction may be
this
eing
the

the
aft,
ner
ugh
the
eam
into
the
the

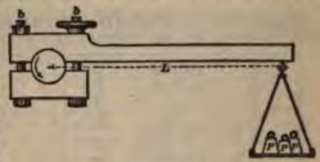


FIG. 167.

e action of gravity; a waste-pipe with its
rough that it acts as a scoop, and removes
sists of a flexible strap to which are fitted
ing-surface; the ends of the strap are con-
np, by means of which any desired tension

shaft is determined from the following:

power absorbed, per minute;
or weight in pounds, acting on lever-arm

feet from centre of shaft;
eet per minute at distance L, if arm were
the speed of the shaft;
per minute;

ave H.P. $= 2\pi LNP \div 33,000$.
$\frac{P}{00}$. $33 \div 2\pi$ is practically 5 ft. 3 in., a value
gth of arm.
nall, the resulting friction will show great
t of insufficient lubrication—the jaws be-
thus producing shocks and sudden vibra-

-tree, beech, poplar, or maple are all to be
brake-blocks. The rubbing-surface should
rease.

-dynamometer. (G. I. Alden, Trans.
00 and xiii. 429.)—This dynamometer is a
n quite moderate sizes of absorbing large
nd complete regulation. A smooth cast-
ing shaft. This is enclosed in a cast-iron
ring at their circumference, which is free
nterior of each of the sides of the shell is
between itself and the side a water-tight
om the city pipes is admitted into each of
er plate against the central disk. The
ed with oil. To the outer shell is fixed a
tendency of the shell to rotate with the
the plates against the central disk. Four
., were used in testing the experimental
y (Trans. A. S. M. E., xiii. 429). Each was
t of 10,500 foot-pounds with a wate

ith the copper plates on either
aving its outer radius equal to
ches. The apparent coefficient
as 3½%.

W. W. Beaumont (Proc. Inst. C. E. 1889) has deduced a formula by means of which the relative capacity of brakes can be compared, judging from the amount of horse-power ascertained by their use.

If W = width of rubbing-surface on brake-wheel in inches; V = vel. of point on circum. of wheel in feet per minute; K = coefficient; then

$$K = WV \div \text{H.P.}$$

Capacity of Friction-brakes.—Prof. Flather obtains the values of K given in the last column of the subjoined table:

Horse-power.	R. P. M. Brake-pulley.	Brake-pulley. Face, in inches.	Brake-pulley. Diameter, in feet.	Length of Arm.	Design of Brake.	Value of K.
21	150	7	5	33″	Royal Ag. Soc., compensating........	785
19	148.5	7	5	33.38″	McLaren, compensating	838
20	146	7	5	32.19″	" water-cooled and comp.....	802
40	180	10.5	5	32″	Garrett, " " "	741
33	150	10.5	5	32″	" " " "	749
150	150	10	9	......	Schoenheyder, water-cooled.........	292
24	142	12	6	38.31″	Balk......................	1385
180	100	24	5	126.1″	Gately & Kletsch, water-cooled......	209
475	76.2	24	7	191″	Webber, water-cooled	84.7
125 } 250 }	290 } 250 }	24	4	63″	Westinghouse, water-cooled.........	465
40 } 125 }	322 } 290 }	13	4	27¾″	" "	847

The above calculations for eleven brakes give values of K varying from 84.7 to 1385 for actual horse-powers tested, the average being $K = 655$.

Instead of assuming an average coefficient, Prof. Flather proposes the following:

Water-cooled brake, non-compensating, $K = 400$; $W = 400$ H.P. $\div V$.

Water-cooled brake, compensating, $K = 750$; $W = 750$ H.P. $\div V$.

Non-cooling brake, with or without compensating device, $K = 900$; $W = 900$ H.P. $\div V$.

Transmission Dynamometers are of various forms, as the Batchelder dynamometer, in which the power is transmitted through a "train-arm" of bevel gearing, with its modifications, as the one described by the author in Trans. A. I. M. E., viii. 177, and the one described by Samuel Webber in Trans. A. S. M. E., x. 514; belt dynamometers, as the Tatham; the Van Winkle dynamometer, in which the power is transmitted from a revolving shaft to another in line with it, the two almost touching, through the medium of coiled springs fastened to arms or disks keyed to the shafts; the Brackett and the Webb cradle dynamometers, used for measuring the power required to run dynamo-electric machines. Descriptions of the four last named are given in Flather on Dynamometers.

Much information on various forms of dynamometers will be found in Trans. A. S. M. E., vol. vii. to xv., inclusive, indexed under Dynamometers.

REFRIGERATING MACHINES.

borate discussion of the thermodynamic theory of uids used in the production of cold was published by *des Mines*, and translated in *Van Nostrand's Magn-*evised and additions made in the light of recent ex- nton, Jacobus, and Riesenberger, was reprinted in cience Series, No. 46.) The work is largely mathe- as much information of immediate practical value, matter given below is taken. Other references are s, Chap. V., and numerous papers by Professors , and Linde in Trans. A. S. M. E., vols. x. to xiv.; rticle on Refrigerating-machines; also *Eng'g*, June ril 1, 1887; June 15, 1888; July 31, Aug. 28, 1889; Sept. 6 and July 8, 1892. For properties of Ammonia and pers by Professors Wood and Jacobus, Trans. A. S.

describing refrigerating-machines, see *Am. Mach.*, , and *Mfrs. Record*, Oct. 7, 1892; also catalogues of Waynesboro, Pa.; De La Vergne Refrigerating-ma- d others.

efrigerating-machine.—Apparatus designed d upon the following series of operations:
or by means of some external force, then relieve it ish its volume; next, cause this compressed gas or produce mechanical work, and thus lower its tem- on of heat at this stage by the gas, in resuming its itutes the refrigerating effect of the apparatus.
ne is a heat-engine reversed.
etween heat-motors and freezing-machines it results duced from the mechanical theory of heat to deter- f the first, apply equally to the second.
upon the difference between the extremes of tem-

a refrigerating-machine depends upon the ratio eliminated and the work expended in compressing

ent of the nature of the body employed.
rs, the freezing-machine possesses the greatest effi- f temperature is small, and when the final tempera-

e the same, there is no theoretical advantage in em- n a vapor in order to produce cold.
ermediate body would be determined by practical the physical characteristics of the body, such as the r manipulating it, the extreme pressures required

dvantage that it is everywhere obtainable, and that igher pressures, independent of the temperature of produce a given useful effect the apparatus must than that required by liquefiable vapors.
re is determined by the temperature of the con- the volatile liquid; this pressure is often very high. ume of a saturated vapor is made under constant re remains constant. The addition or subtraction of change of volume, is represented by an increase or tity of liquid mixed with the vapor.
en vapors, even if saturated, are no longer in con- nd receive an addition of heat either through l force, or from some external source of hea rly in the same way as permanent gases,

perty, that refrigerating-machines using *s differing according* to the method of

and depending upon the state of the gas, whether it remains constantly s-
urated, or is superheated during a part of the cycle of working.

The temperature of the condenser is determined by local conditions. T
interior will exceed by 9° to 18° the temperature of the water furnished
the exterior. This latter will vary from about 52° F., the temperature
water from considerable depth below the surface, to about 92° F., the t
perature of surface-water in hot climates. The volatile liquid employed
the machine ought not at this temperature to have a tension above
which can be readily managed by the apparatus.

On the other hand, if the tension of the gas at the minimum temper
is too low, it becomes necessary to give to the compression-cylinder
dimensions, in order that the weight of vapor compressed by a single
of the piston shall be sufficient to produce a notably useful effect.

These two conditions, to which may be added others, such as the
pending upon the greater or less facility of obtaining the liquid, up
dangers incurred in its use, either from its inflammability or unhea
ness, and finally upon its action upon the metals, limit the choice to
number of substances.

The gases or vapors generally available are: sulphuric ether, sulp
oxide, ammonia, methylic ether, and carbonic acid.

The following table, derived from Regnault, shows the tensions
vapors of these substances at different temperatures between −22°
104°.

**Pressures and Boiling-points of Liquids availabl
Use in Refrigerating-machines.**

Temp. of Ebullition.	Tension of Vapor, in lbs. per sq. in., above Zero				
Deg. Fahr.	Sulphuric Ether.	Sulphur Dioxide.	Ammonia.	Methylic Ether.	Carbonic Acid.
−40	....	....	10.22	....	....
−31	....	....	13.23	....	....
−22	....	5.56	16.95	11.15	....
−13	....	7.23	21.51	13.85	251.6
−4	1.30	9.27	27.04	17.06	292.9
5	1.70	11.76	33.67	20.84	340.1
14	2.19	14.75	41.58	25.27	393.4
23	2.79	18.31	50.91	30.41	453.4
32	3.55	22.53	61.85	36.34	520.4
41	4.45	27.48	74.55	43.13	594.8
50	5.54	33.26	89.21	50.84	676.9
59	6.84	39.93	105.99	59.56	766.9
68	8.38	47.62	125.08	69.35	864.9
77	10.19	56.39	146.64	80.28	971.1
86	12.31	66.37	170.83	92.41	1085.6
95	14.76	77.64	197.83	....	1207.9
104	17.59	90.32	227.76	....	1338.[illegible]

The table shows that the use of ether does not readily lead to th
tion of low temperatures, because its pressure becomes then very

Ammonia, on the contrary, is well adapted to the production of
peratures.

Methylic ether yields low temperatures without attaining too gr
sures at the temperature of the condenser. Sulphur dioxide reac
temperatures of −14 to −5, while its pressure is only 3 to 4 atm
at the ordinary temperature of the condenser. These latter substa
lend themselves conveniently for the production of cold by
mechanical force.

The "Pictet fluid" is a mixture of 97% sulphur dioxide and 3%
acid. At atmospheric pressure it affords a temperature 14° lo
sulphur dioxide.

*Carbonic acid is as yet (1895) in use but to a limited extent, w
likely ... compactness of compressor that it requires, and its*

nmendation for service on shipboard, where

erated with a surplus of liquid present dur-
ating is prevented. This practice is known
ssion.
rding the application of methylic ether or
gene in practical refrigerating service. The
nd the cumbrousness of the compressor
.
-It is agreed that the term "ice-melting
d in an insulated bath of brine, on the as-
' represents one pound of ice, this being the
the heat required to melt a pound of ice at
ature.
, expressed in pounds or tons of "ice-melt-
hat the refrigerating-machine would make
but that the cold produced is equivalent to
t 32° to water of the same temperature.
er frozen is generally about 70° F. when sub-
of a machine; second, the ice is chilled from
t; third, there is a dissipation of cold, from
nd the manipulation of the ice-cans: there-
nade, multiplied by its latent heat of fusion,
nly about three fourths of the cold produced
; fluid per I.H.P. of the engine driving the
re is considerable fuel consumed to operate
condensing-water and feed-pumps, and to
steam from which the ice is frozen. This
n leakage and drip water, amounts to about
he main steam-engine. Hence the pounds
distilled water is just about half the equiv-
produced in the brine per indicated horse-

om natural water by means of the "plate
used with distilled water, is saved by avoid-
m expansively in a compound engine.
India, are said to have produced about 6
uel consumed.
because the density of the vapor of ether,
re, requires that the compressing-cylinder
n for sulphur dioxide, and 17 times larger

ut 1.2 times greater capacity of compress-
e, more cumbersome than ether machines,
oard. In using air the expansion must take
nstead of through a simple expansion-cock
nes. The work done in the expansion-cylin-
mpressor.
-machines.—*"Cold" vs. "Dry" Systems*
' system or "humid" system some of the
sion-cylinder is liquid, so that the heat de-
ed by the liquid and the temperature of the
he boiling-point due to the condenser-pres-
quired about the cylinder.
n all ammonia entering the compressor is
comes by compression several hundred de-
int due to the condenser-pressure. A water-
o permit the cylinder to be properly lubri-

**of Ammonia Compression- and
ssuming no Water to be En-
nia-gas in the Condenser.** (Denton
, xiii.)—It is assumed in the calculation for
l imparts 10,000 B.T.U. to the boiler. The

' ice is 144 thermal units (*Phil.* M
use 142. (Prof. Wood, Trans. A

...onia Compression-machines.—Ammonia gas possesses ... of sulphur dioxide for the same volume described by the piston.

The perfection of ammonia apparatus now renders it so convenient and reliable that no pra... r pressures afforded by sulphur dioxide.

The results of the calculations for ammonia are given in the table below:

PERFORMANCE OF AMMONIA COMPRESSION-MACHINES.

as superheated during compression as in ordinary practice. Temperature of condenser, 64.4° Fahr. Pressure in condenser, 117.44 lbs per sq. in. (Ledoux.)

Temperature Corresponding to Pressure of Vapor in Refrigerating-coils.	Absolute Pressure in Refrigerating-coils.	Temperature of Gas at End of Compression.	Per Cubic Foot of Piston Displacement.					Performance in British Thermal Units.		Ice-melting Capacity per Cubic Foot of Piston Displacement.	Ice-melting Capacity per lb. of Coal, assuming 3 lbs. of Coal per hour per H.P. of Steam-cylinder. With Friction.	Condensing-water. Per Ton of Ice-melting Capacity, assuming 30° F. Range of Temperature.
			Weight of Gas Compressed.	Heat Abstracted at Condenser.	Number of Negative Thermal Units Developed.	Work of Compression. Without Friction.	Work of Compression. With Friction, or Indicated Steam-power.	Per ft.-lb. of Work of Compression. With Friction.	Per hour per Horse-power. With Friction.			
t_2	$P_2 \div 144$	t_1	m	Q_1	Q	W_r	$1.15W_r$	$\frac{Q}{1.15W_r}$				
Deg. F.	Lbs. per sq. in.	Deg. F.	Lbs.	B.T.U.	B.T.U.	Ft.-lbs.	Ft.-lbs.			Tons.	Lbs.	Gals.
9.66	37.76	158.9	.1329	78.56	69.41	7070	8130	.00854	16,900	.000244	39.6	1290
5.00	33.67	170.1	.1206	71.98	62.77	7120	8190	.00766	15,170	.000221	35.6	1310
22.00	16.95	241.3	.0639	40.45	32.58	6080	6990	.00466	9,230	.000115	21.6	1410

... reasons that have been stated for sulphur dioxide. ... been determined experimentally by

Temp. Due to Press. Vapor in Condenser.	Absolute Pressure in Condenser.	Temperature at End of Compression.	Heat Carried away from Condenser.	Refrigerating Effect in Heat Units.	Ratio of Refrigerating Effect to Heat Expended.	Work of Compression, without Friction.	Work of Compression, with Friction, or Indicated Steam-power.	Refrigerating Effect in Heat Units. Per ft.-lb. of Work Expended, without Friction.	Refrigerating Effect in Heat Units. Per ft.-lb. of Work Expended, including Friction.	Refrigerating Effect in Heat Units. Per Hour per H.P., including Friction.	Ice-melting Capacity. Per Hour per H.P. Without Friction.	Ice-melting Capacity. Per Hour per H.P. With Friction.	Ice-melting Capacity. Per Pound of Coal. Without Friction.	Ice-melting Capacity. Per Pound of Coal. With Friction.	Ice-melting Capacity. Per cu. ft. of Piston Displacement.	Condensing-water. Per cu. ft. of Piston Displacement, assuming 30° Range of Temp.	Condensing-water. Per Ton of Ice-melting Capacity.	Condensing-water. Per Minute per Ton of Ice-melting Capacity in 24 hours.
Deg. F.	Lbs. per sq. in.	Deg. F.	B.T.U.	B.T.U.		Ft.-lbs.	Ft.-lbs.	B.T.U.	B.T.U.	B.T.U.	Lbs.	Lbs.	Lbs.	Lbs.	Tons.	Gals.	Gals.	Gals.
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)
59	106.0	154.6	71.81	63.47	7.61	6,450	7,410	.00984	.00857	16,960	137.0	119.3	45.8	39.8	.000223	.2872	1290	.89
68	125.1	179.9	72.05	62.31	6.40	7,530	8,660	.00827	.00719	14,250	115.2	100.2	38.4	33.4	.000219	.2882	1320	.92
77	146.6	205.1	72.26	61.13	5.49	8,600	9,890	.00711	.00618	12,240	99.0	86.1	33.0	28.7	.000215	.2890	1350	.94
86	170.8	230.3	72.46	59.93	4.78	9,680	11,130	.00619	.00538	10,660	86.2	75.0	28.7	25.0	.000211	.2898	1380	.96
95	197.8	255.4	72.61	58.70	4.22	10,750	12,360	.00546	.00475	9,400	76.0	66.1	25.3	22.0	.000206	.2904	1410	.98
104	227.8	280.2	72.74	57.45	3.76	11,820	13,590	.00486	.00423	8,380	67.7	58.9	22.6	19.6	.000202	.2910	1440	1.00

REFRIGERATING EFFECT OF 1 CU. FT. OR .06386 LBS. OF AMMONIA EXPANDED THROUGH A SIMPLE COCK TO 16.95 LBS. ABSOLUTE PRESSURE PER SQ. IN., AND TAKEN INTO THE COMPRESSOR AT THIS PRESSURE AND THE CORRESPONDING TEMPERATURE OF − 22° F.

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)
59	106.0	224.1	40.28	32.93	4.48	5,680	6,530	.00580	.00504	9,980	80.7	70.2	26.9	23.4	.000116	.1611	1,390	.97
68	125.1	252.2	40.50	32.31	3.95	6,330	7,280	.00510	.00444	8,790	71.0	61.8	23.7	20.6	.000114	.1620	1,420	.99
77	146.6	280.2	40.70	31.69	3.52	6,960	8,000	.00455	.00396	7,840	63.3	55.1	21.1	18.4	.000111	.1628	1,470	1.02
86	170.8	308.3	40.90	31.05	3.15	7,610	8,750	.00408	.00355	7,030	56.8	49.4	18.9	16.5	.000109	.1636	1,500	1.04
95	197.8	336.2	41.07	30.41	2.85	8,240	9,480	.00369	.00321	6,360	51.4	44.7	17.1	14.9	.000107	.1643	1,540	1.07
[illegible]	227.8	364.0	41.23	29.75	2.59	8,870	10,200	.00335	.00292	5,780	46.6	40.6	15.5	13.5	.000105	.1649	1,570	1.09

T	A	T	H	R	R	W	W	P	P	P	W	W	W	W	P	P	F	F
Deg. F.	Lbs. per sq. in.	Deg. F.	B.T.U.	B.T.U.		Ft.-lbs.	Ft.-lbs.	B.T.U.	B.T.U.	B.T.U.	Lbs.	Lbs.	Lbs.	Lbs.	Tons.	Gals.	Gals.	Gals.
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)
59	106.0	154.6	71.81	63.47	7.61	6,450	7,410	.00984	.00857	16,960	137.0	119.8	45.8	39.8	.000223	.2872	1290	.89
68	125.1	179.9	72.05	62.31	6.40	7,530	8,660	.00827	.00719	14,250	115.2	100.2	38.4	33.4	.000219	.2882	1320	.92
77	146.6	205.1	72.26	61.13	5.49	8,600	9,890	.00711	.00618	12,240	99.0	86.1	33.0	28.7	.000215	.2890	1350	.94
86	170.8	230.3	72.46	59.93	4.78	9,680	11,130	.00619	.00538	10,660	86.2	75.0	28.7	25.0	.000211	.2898	1380	.96
95	197.8	255.4	72.61	58.70	4.22	10,750	12,360	.00546	.00475	9,400	76.0	66.1	25.3	22.0	.000206	.2904	1410	.98
104	227.8	280.3	72.74	57.45	3.76	11,820	13,590	.00486	.00423	8,380	67.7	58.9	22.6	19.6	.000202	.2910	1440	1.00

REFRIGERATING EFFECT OF 1 CU. FT. OR .06386 LBS. OF AMMONIA EXPANDED THROUGH A SIMPLE COCK TO 16.95 LBS. ABSOLUTE PRESSURE PER SQ. IN., AND TAKEN INTO THE COMPRESSOR AT THIS PRESSURE AND THE CORRESPONDING TEMPERATURE OF − 22° F.

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)
59	106.0	224.1	40.28	32.93	4.48	5,680	6,530	.00580	.00504	9,980	80.7	70.2	26.9	23.4	.000116	.1611	1,390	.97
68	125.1	252.2	40.50	32.31	3.95	6,330	7,280	.00510	.00441	8,790	71.0	61.8	23.7	20.6	.000114	.1620	1,420	.99
[illegible]	146.6	280.2	40.70	31.69	3.52	6,960	8,000	.00455	.00396	7,840	63.3	55.1	21.1	18.4	.000111	.1628	1,470	1.02
[illegible]	0.8	308.3	40.90	31.05	3.15	7,610	8,750	.00408	.00355	7,030	56.8	49.4	18.9	16.5	.000109	.1636	1,500	1.04
[illegible]	7.8	336.2	41.07	30.41	2.85	8,240	9,480	.00369	.00321	6,360	51.4	44.7	17.1	14.9	.000107	.1643	1,540	1.07
[illegible]	7.8	364.0	41.23	29.75	2.59	8,870	10,200	.00335	.00292	5,780	46.6	40.6	15.5	13.5	.000105	.1649	1,570	1.09

The following is a comparison of the theoretical ice-melting capacity of an ammonia compression machine with that obtained in some of Prof. Schröter's tests on a Linde machine having a compression-cylinder 9½ in. bore and 16.5 in. stroke, and also in tests by Prof. Denton on a machine having two single-acting compression cylinders 12 in. × 30 in.:

No. of Test.		Temp. in Degrees F. Corresponding to Pressure of Vapor.		Ice-melting Capacity per lb. of Coal, assuming 3 lbs. per hour per Horse-power.		
		Condenser.	Suction.	Theoretical, Friction * included.	Actual.	Per Cent of Loss Due to Cylinder Superheating.
Schröter	1	72.3	26.6	50.4	40.6	19.4
	2	70.5	14.3	37.6	30.0	20.2
	3	69.2	0.5	29.4	22.0	25.2
	4	68.5	−11.8	22.8	16.1	29.4
Denton	24	84.2	15.0	27.4	24.2	11.7
	26	82.7	− 3.2	21.6	17.5	19.0
	25	84.6	−10.8	18.8	14.5	22.9

Refrigerating Machines using Vapor of Water. (Ledoux.) —In these machines, sometimes called vacuum machines, water, at ordinary temperatures, is injected into, or placed in connection with, a chamber in which a strong vacuum is maintained. A portion of the water vaporizes, the heat to cause the vaporization being supplied from the water not vaporized, so that the latter is chilled or frozen to ice. If brine is used instead of pure water, its temperature may be reduced below the freezing-point of water. The water vapor is compressed from, say, a pressure of one tenth of a pound per square inch to one and one half pounds, and discharged into a condenser. It is then condensed and removed by means of an ordinary air-pump. The principle of action of such a machine is the same as that of volatile-vapor machines.

A theoretical calculation for ice-making, assuming a lower temperature of 32° F., a pressure in the condenser of 1½ lbs. per square inch, and a coal consumption of 3 lbs. per I.H.P. per hour, gives an ice-melting effect of 3[illegible] lbs. per pound of coal, neglecting friction. Ammonia for ice-making conditions gives 40.9 lbs. The volume of the compressing cylinder is about [illegible] times the theoretical volume for an ammonia machine for these conditions.

Relative Efficiency of a Refrigerating Machine.—The efficiency of a refrigerating machine is sometimes expressed as the quotient of the quantity of heat received by the ammonia from the brine, that is, the quantity of useful work done, divided by the heat equivalent of the mechanical work done in the compressor. Thus in column 1 of the table of performance of the 75-ton machine (page 998) the heat given by the brine to the ammonia per minute is 14,776 B.T.U. The horse-power of the ammonia cylinder is 65.7, and its heat equivalent = 65.7 × 33,000 ÷ 778 = 2786 B.T.U. Then 14,776 ÷ 2786 = 5.304, efficiency. The apparent paradox that the efficiency is greater than unity, which is impossible in any machine, is thus explained. The working fluid, as ammonia, receives heat from the brine and rejects heat into the condenser. (If the compressor is jacketed, a portion is rejected into the jacket-water.) The heat rejected into the condenser is greater than that received from the brine; the difference (plus or minus a small difference radiated to or from the atmosphere) is heat received by the ammonia from the compressor. The work to be done by the compressor is not the mechanical equivalent of the refrigeration of the brine, but only that necessary to supply the difference between the heat rejected by the ammonia into the condenser and that received from the brine. If cooling water colder than the brine were available, the brine might transfer its heat directly into the cooling water, and there would be no need of ammonia or of a compressor; the

* *Friction taken at figures observed in the tests, which range from 14[illegible] [illegible] of the steam-cylinder.*

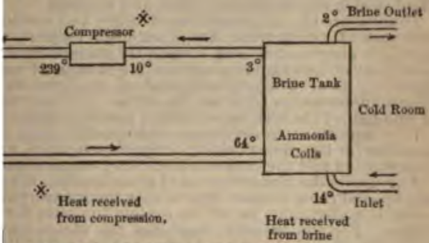
s not available, the brine rejects its heat into the
en the compressor is required to heat the ammonia
hat it may reject heat into the cooling water.
efrigerating plant referred to the amount of fuel

$$= \frac{\left\{\begin{array}{l}\text{Pounds circulated per hour} \\ \times \text{ specific heat} \times \text{range} \\ \text{of temperature}\end{array}\right\} \text{of brine or other circulating fluid.}}{142.2 \times \text{pounds of fuel used per hour.}}$$

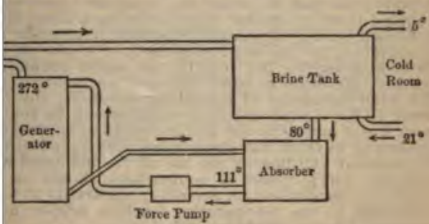
ity is expressed as follows:

$$= \frac{\left\{\begin{array}{l}24 \times \text{pounds} \\ \times \text{ specific heat} \\ \times \text{ range of temp.}\end{array}\right\} \text{of brine circulated per hour.}}{142.2 \times 2000}$$

a heat-engine and a refrigerating-machine is as fol-
receives heat from the boiler, converts a part of it
n the cylinder, and throws away the difference into
nmonia in a compression refrigerating-machine re-
ine-tank or cold-room, receives an additional amount
anical work done in the compression-cylinder, and
nto the condenser. The efficiency of the steam-engine
ceived from boiler. The efficiency of the refrigerat-
eived from the brine-tank or cold-room + heat re-
vork in the compression-cylinder. In the ammonia



F AMMONIA COMPRESSION MACHINE.



OF AMMONIA ABSORPTION MACHINE.

he ammonia receives heat from the brine-tank
boiler or generator, and rejects the sum i
cooling water supplied to the absorber. T
from the brine + heat received from the bo

$$\frac{AL}{Q'} = u, \text{ and } \frac{Q'}{Q} = \frac{T_o - T}{uT}.$$

It follows that the expenditure of heat Q' necessary for the produ the quantity of cold Q in a compression-machine will be the smal smaller the difference of temperature $T_o - T$.

Metering the Ammonia.—For a complete test of an amm frigerating-machine it is advisable to measure the quantity of amm culated, as was done in the test of the 75-ton machine described b Denton. (Trans. A. S. M. E., xii. 326.)

PROPERTIES OF SULPHUR DIOXIDE AND AMMONIA GAS.

Ledoux's Table for Saturated Sulphur-dioxide G

Heat-units expressed in B.T.U. per pound of sulphur dioxide.

Temperature of Ebullition in deg. F. t	Absolute Pressure in lbs. per sq. in. $P \div 144$	Total Heat reckoned from 32° F. λ	Heat of Liquid reckoned from 32° F. q	Latent Heat of Evaporation r	Heat Equivalent of External Work. APu	Internal Latent Heat. ρ	Increase of Volume during Evaporation. u
Deg. F.	Lbs.	B.T.U.	B.T.U.	B.T.U.	B.T.U.	B.T.U.	Cu. ft.
−22	5.56	157.43	−19.56	176.99	13.59	163.39	13.17
−13	7.23	158.64	−16.30	174.95	13.83	161.12	10.27
− 4	9.27	159.84	−13.05	172.89	14.05	158.84	8.12
5	11.76	161.03	− 9.79	170.82	14.26	156.56	6.50
14	14.74	162.20	− 6.53	168.73	14.46	154.27	5.25
23	18.31	163.36	− 3.27	166.63	14.66	151.97	4.29
32	22.53	164.51	0.00	164.51	14.84	149.68	3.54
41	27.48	165.65	3.27	162 38	15.01	147.37	2.93
50	33.25	166.78	6.55	160.23	15.17	145.06	2.45
59	39.93	167.90	9.83	158.07	15.32	142.75	2.07
68	47.61	168.99	13.11	155.89	15.46	140.43	1.75
77	56.39	170.09	16.39	153.70	15.59	138.11	1.49
86	66.36	171.17	19.69	151.49	15.71	135.78	1.27
95	77.64	172.24	22.98	149.26	15.82	133.45	1.09
104	90.31	173.30	26.28	147.02	15.91	131.11	91

Density of Liquid Ammonia. (D'Andreff, Trans. A. S. x. 641.)

At temperature C.......	− 10	− 5	0	5	10	15
" " F.....	+ 14	23	32	41	50	59
Density.........	.6492	.6429	.6364	.6298	.6230	.6160

These may be expressed very nearly by

$$\delta = 0.6364 - 0.0014t° \text{ Centigrade;}$$
$$\delta = 0.6502 - 0.000777T° \text{ Fahr.}$$

Latent Heat of Evaporation of Ammonia. (Wood, A. S. M. E., x. 641.)

$$h_e = 555.5 - 0.613T - 0.0002197T^2 \text{ (in B.T.U., Fahr. d}$$

Ledoux found $h_e = 583.33 - 0.5499T - 0.0001173T^2$.

For experimental values at different temperatures determined by Denton, see Trans. A. S. M. E., xii. 356. For calculated value vol. x. 646.

Density of Ammonia Gas.—Theoretical, 0.5894; experim 0.596. Regnault (Trans. A. S. M. E., x. 633).

Specific Heat of Liquid Ammonia. (Wood, Trans. A. S. x 615.)—The specific heat is nearly constant at different temperature about equal to that of water, or unity. From 0° to 100° F., it is

$$c = 1.096 - .0012T, \text{ nearly.}$$

In a later paper by Prof. Wood (Trans. A.S.M.E., xii. 136) he gives a hi value, viz., $c = 1.12136 + 0.000438T$.

achines) it will be possible to obtain for the
ance exhibiting small discrepancies only.
intended to be used for comparison with
ines will therefore have to embrace at least

er hour........ ....
efrigerator.................. ....
refrigerator .. ..t
° Fahr.) ....
....
d)..................................Q_e
gerator...................... .. ...

hour ... .
nser.......................... ....
enser.... ....t
...................................Q_1
enser.. ....
the condenser.... ....

	COMPRESSION-MACHINE.
	Compressor:
..	Indicated work........... ....Lt
1.	Temperature of gases at inlet..
d	Temperature of gases at exit..
.	*Steam-engine:*
'e	Feed-water per hour.......... ...
	Temperature of feed-water....
r	Absolute steam-pressure before steam-engine.................
.	Indicated work of steam-engine Le
$_3$	Condensing water per hour....
	Temperature of da.............
e	Total sum of losses by radiation and convection... ...... $\pm Q_3$
f	*Heat Balance:*
p	$Q_e + ALc = Q_1 \pm Q_3$.
n	
$_3$	

cy and for comparison of various tests, the
ared with the theoretical maximum of effi-

corresponding to the temperature range.

As temperatures (T and Tc) at which the
ator and imparted to the condenser, it is cor-
the brine leaving the refrigerator and that
condenser, because it is in principle impos-
essure higher than would correspond to the
reduce the condenser pressure below that
perature of the cooling water.
aximum theoretical efficiency of a com-
ssed by the formula

$$= \frac{T}{Tc - T},$$

bstracted (cold produced);
t of the mechanical work expended;
rk, and $A = 1 \div 778$;
ure of heat abstraction (refrigerator);
" " rejection (condenser).

equivalent of the mechanical work AI
ust be imparted to the motor to pr

$$\frac{AL}{Q'} = u, \text{ and } \frac{Q'}{Q} = \frac{T_e - T}{uT}.$$

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Deg. F.	Lbs.	B.T.U.	B.T.U.	B.T.U.	B.T.U.	B.T.U.	Cu. ft.	[illegible]
−22	5.56	157.43	−19.56	176.99	13.59	163.39	13.17	[illegible]
−13	7.23	158.64	−16.30	174.95	13.83	161.12	10.27	[illegible]
− 4	9.27	159.84	−13.05	172.89	14.05	158.84	8.12	[illegible]
5	11.76	161.03	− 9.79	170.82	14.26	156.56	6.50	[illegible]
14	14.74	162.20	− 6.53	168.73	14.46	154.27	5.25	[illegible]
23	18.31	163.36	− 3.27	166.63	14.66	151.97	4.29	[illegible]
32	22.53	164.51	0.00	164.51	14.84	149.68	3.54	[illegible]
41	27.48	165.65	3.27	162.38	15.01	147.37	2.93	[illegible]
50	33.25	166.78	6.55	160.23	15.17	145.06	2.45	[illegible]
59	39.93	167.90	9.83	158.07	15.32	142.75	2.07	[illegible]
68	47.61	168.99	13.11	155.89	15.46	140.43	1.75	[illegible]
77	56.39	170.09	16.39	153.70	15.59	138.11	1.49	[illegible]
86	66.36	171.17	19.69	151.49	15.71	135.78	1.27	[illegible]
95	77.64	172.24	22.98	149.26	15.82	133.45	1.09	[illegible]
104	90.31	173.30	26.28	147.02	15.91	131.11	.91	[illegible]

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At temperature C.......	−10	−5	0	5	10	15
" " F.....	+14	23	32	41	50	59
Density.........	.6492	.6429	.6364	.6298	.6230	.6160

These may be expressed very nearly by

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$$c = 1.096 - .0012T, \text{ nearly.}$$

In a ... by Prof. Wood (*Trans. A. S. M. E.*, xii. 136) he gives a l *value ... .22136 + 0.000438T.*

Condensing Pressure, lbs.	Suction-pressure, lbs.	Pounds of Ice-melting Effect with Engines—						B.T.U. per lb. of Steam with Engines—		
		Non-condensing.		Non-compound Condensing.		Compound Condensing.		Non-condensing.	Condensing.	Compound Condensing.
		Per lb. Coal.	Per lb. Steam.	Per lb. Coal.	Per lb. Steam.	Per lb. Coal.	Per lb. Steam.			
150	28	24	2.90	30	3.61	37.5	4.51	393	513	640
150	7	14	1.69	17.5	2.11	21.5	2.58	240	300	366
105	28	34.5	4.16	43	5.18	54	6.50	591	725	923
105	7	22	2.65	27.5	3.31	34.5	4.16	376	470	591

The non-condensing engine is assumed to require 25 lbs. of steam per horse-power per hour, the non-compound condensing 20 lbs., and the compound condensing 16 lbs., and the boiler efficiency is assumed at 8.3 lbs. of water per lb. coal under working conditions. The following conclusions were derived from the investigation:

[T]he capacity of the machine is proportional, almost entirely, to the ...t of ammonia circulated. This weight depends on the suction-...ure and the displacement of the compressor-pumps. The practical ...on-pressures range from 7 lbs. above the atmosphere, with which a ...erature of 0° F. can be produced, to 28 lbs. above the atmosphere, with ...h the temperatures of refrigeration are confined to about 28° F. At the ... pressure only about one half as much weight of ammonia can be cir-...ed as at the upper pressure, the proportion being about in accordance ...the ratios of the absolute pressures, 22 and 42 lbs. respectively. For each ... foot of piston-displacement per minute a capacity of about one sixth ...on of "refrigerating effect" per 24 hours can be produced at the lower ...ure, and of about one third of a ton at the upper pressure. No other ...nts practically affect the capacity of a machine, provided the cooling-...ce in the brine-tank or other space to be cooled is equal to about ... ft. per ton of capacity at 28 lbs. back pressure. For example, a differ-...of 100% in the rate of circulation of brine, while producing a propor-...l difference in the range of temperature of the latter, made no practical ...ence in capacity.

... brine-tank was 10½ × 13 × 1⁰⁄₁₂ ft., and contained 8000 lineal feet of ...pipe as cooling-surface. The condensing-tank was 12 × 10 × 10 ft., and ...ined 5000 lineal feet of 1-in. pipe as cooling-surface.

[T]he economy in coal-consumption depends mainly upon both the suc-...pressures and condensing-pressures. Maximum economy, with a given ... of engine, where water must be bought at average city prices, is ...ned at 28 lbs. suction-pressure and about 150 lbs. condensing-pressure. ...r these conditions, for a non-condensing steam-engine, consuming coal ... rate of 3 lbs. per hour per I.H.P. of steam-cylinders, 24 lbs. of ice-...erating effect are obtained per lb. of coal consumed. For the same ...nsing-pressure, and with 7 lbs. suction-pressure, which affords tem-...ures of 0° F., the possible economy falls to about 14 lbs. of "refrigerat-...fect" per lb. of coal consumed. The condensing-pressure is determined ...e amount of condensing-water supplied to liquefy the ammonia in the ...nser. If the latter is about 1 gallon per minute per ton of refrigerating ... per 24 hours, a condensing-pressure of 150 lbs. results, if the initial tem-...ure of the water is about 56° F. Twenty-five per cent less water causes ...ndensing-pressure to increase to 190 lbs. The work of compression is ...y increased about 20%, and the resulting "economy" is reduced to ... 18 lbs. of "ice effect" per lb. of coal at 28 lbs. suction-pressure and ... 7 lbs. If, on the other hand, the supply of water is made 3 gallons ...inute, the condensing-pressure may be confined to about 105 lbs. The ...f compression is thereby reduced about 25%, and a proportional increase ...nomy results. Minor alterations of economy depend on the initial ...rature of the condensing-water and variations of latent heat, but these ...nfined within about 5% of the gross result, the main element of control ... the work of compression, as affected by the back pressure and con-...g-pressure, or *both.* *If the steam-engine supplying the motive power ...e a condenser to secure a vacuum, an increase of economy of 25% is ...le over the above figures, making the lbs. of "ice effect" per lb. c...*

ce-melting Effect with Engines—				B.T.U. per lb. of Steam with Engines—		
Non-compound Condensing.		Compound Condensing.		Non-condensing.	Condensing.	Compound Condensing.
Per lb. Coal.	Per lb. Steam.	Per lb. Coal.	Per lb. Steam.			
30	3.61	37.5	4.51	393	513	640
17.5	2.11	21.5	2.58	240	300	366
43	5.18	54	6.50	591	725	923
27.5	3.31	34.5	4.16	376	470	591

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essure, the proportion being about in accordance
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cement per minute a capacity of about one sixth
effect" per 24 hours can be produced at the lower
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btained per lb. of coal consumed. For the same
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sible economy falls to about 14 lbs. of "refrigerat-
onsumed. The condensing-pressure is determined
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r alterations of economy depend on the initial
sing-water and variations of latent heat, but these
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ession, as affected by the back pressure and
If the steam-engine supplying the motive
ure a vacuum, an increase of economy of
gures, making the lbs. of "ice effect" y

Performance of a 75-ton Refrigerating-machi

	Maximum Capacity and Economy at 28 lbs. Back Pressure.	Maximum Capacity and Economy at Zero, Brine, and 8 lbs. Back Pressure.	Maximum Capacity and Economy for Zero, Brine, 13 lbs. Back Pressure.
Av. high ammonia press. above atmos.	151 lbs.	152 lbs.	147 lbs.
Av. back ammonia press. above atmos.	28 "	8.2 "	13 "
Av. temperature brine inlet	36.76°	6.27°	14.29
Av. temperature brine outlet	28.86°	2.03°	2.29
Av. range of temperature	7.9°	4.24°	12.00
Lbs. of brine circulated per minute	2281	2173	943
Av. temp. condensing-water at inlet	44.65°	56.65°	46.9°
Av. temp. condensing-water at outlet	83.66°	85.4°	85.46
Av. range of temperature	39.01°	28.75°	38.56
Lbs. water circulated p. min. thro' cond'ser	442	315	257
Lbs. water per min. through jackets	25	44	40
Range of temperature in jackets	24.0°	16.2°	16.4°
Lbs. ammonia circulated per min	*28.17	14.68	16.67
Probable temperature of liquid ammonia, entrance to brine-tank	*71.3°	*68°	*63.7°
Temp. of amm. corresp. to av. back press.	+14°	− 8°	− 5°
Av. temperature of gas leaving brine-tanks	34.2°	14.7°	3 0°
Temperature of gas entering compressor	*39°	25°	10.13
Av. temperature of gas leaving compressor	213°	263°	239°
Av. temp. of gas entering condenser	200°	218°	209°
Temperature due to condensing pressure	84.5°	84.0°	82.5°
Heat given ammonia:			
By brine, B.T.U. per miniute	14776	7186	8824
By compressor, B.T.U. per minute	2786	2320	2518
By atmosphere, B.T.U. per minute	140	147	167
Total heat rec. by amm., B.T.U. per min.	17702	9653	11409
Heat taken from ammonia:			
By condenser, B.T.U. per min	17242	9056	9910
By jackets, B.T.U. per min	608	712	656
By atmosphere, B.T.U. per min	182	338	250
Total heat rej. by amm., B.T.U. per min.	18032	10106	10816
Dif. of heat rec'd and rej., B.T.U. per min.	330	453	407
% work of compression removed by jackets.	22%	31%	26%
Av. revolutions per min	58.09	57.7	57.88
Mean eff. press. steam-cyl., lbs. per sq. in.	32.5	27.17	27.83
Mean eff. press. amm.-cyl., lbs. per sq. in.	65.9	53.3	59.86
Av. H.P. steam-cylinder	85.00	71.7	73.6
Av. H.P. ammonia-cylinder	65.7	54.7	59.37
Friction in per cent of steam H.P.	23.0	24.0	20.0
Total cooling water, gallons per min. per ton per 24 hours	*0.75	1.185	0.79
Tons ice-melting capacity per 24 hours	74.8	36.43	44.64
Lbs. ice-refrigerating eff. per lb. coal at 3 lbs. per H.P. per hour	24.1	14.1	17.27
Cost coal per ton of ice-refrigerating effect at $4 per ton	$0.166	$0.283	$0.231
Cost water per ton of ice-refrigerating effect at $1 per 1000 cu. ft.	$0.128	$0.200	$0.13[illegible]
Total cost of 1 ton of ice-refrigerating eff.	$0.294	$0.483	$0.46[illegible]

Figures marked thus (*) are obtained by calculation; all other figures are obtained from experimental data; temperatures are in Fahrenheit degrees.

the ice-melting capacity ranges from 46.29
f coal, according as the suction pressure
ove the atmosphere, this pressure being the
s the economy of compression-machines,
ealizing from 72% to 57% of theoretically per-
per cents appear to occur with the higher
greater loss from cylinder-heating (a phe-
er condensation in steam-engines), as the
the gas in the compression-cylinder is

n-machine, operating on the "dry system,"
ffect realized ranges from 69.5% to 62.6%.
the American machine. The latter's higher
refore, to more perfect displacement.
city" in the American machine is 24.16 lbs.
uction-pressures used in American practice
red in beer-storage cellars using the direct-
ns most nearly corresponding to American
ests are those in line 5, which give an "ice-

ial ice, the conditions of practice are those
26. In the former the condensing pressure
cooling water than is common in American
ity is therefore greater in the German ma-
against 17.55 and 14.52 for the American

Pictet Machines.—No records are available
elting capacity" of machines using pure
use in American machines, but in Europe
fluid," a mixture of about 97% of sulphur
The presence of the carbonic acid affords
grees lower than is obtained with pure sul-
ssure. The latent heat of this mixture has
sumed to be equal to that of pure sulphur

nditions, line 17, we have 26.24 lbs. "ice-
making conditions, line 13, the "ice-melt-
ese figures are practically as economical
cent of theoretical effect realized ranging
low temperatures, −15° Fahr., lines 14 and
as 42.5.
ompression-machines employing volatile
difference between the theoretical and the
the ammonia, by the warm cylinder walls,
npressor, thereby expanding it, so that to
a greater number of revolutions must be
s than corresponds to the density of the
e brine-tank.
rption-machine used in storage-ware-
New York and Brooklyn Bridge. (*Eng'g*,
id consisted of a solution of chloride of cal-
heat was found to be .827.
is for 24 hours was found by taking the
circulating through the pipes by the aver-
n the ingoing and outgoing currents, as
the specific heat of the brine (.827) and its
e final product, applying all allowances for
. amounted to 6,218,816 heat-units as the
qual to the melting of 43,565 lbs. of ice in

of the coal used in 24 hours was 27,000,000
f the apparatus was 23%. This is equivalent
s. per lb. of coal having a heating value of

machine in New Haven, Conn.,
, gave an ice-melting effect of
economy equivalent to 3 lbs. o
steam-engine. The ammonia
bove the atmosphere.

a Compression-machine.

TS OBTAINED AT THE MUNICH TESTS.

le, Trans. A. S. M. E., xiv. 1419.)

....	1	2	3	4	5
g F......	43.194	28.344	13.952	−0.279	28.251
t deg. F...	37.054	22.885	8.771	−5.879	23.072
....	0.861	0.851	0.843	0.837	0.851
r h., cu. ft.	1,039.38	908.84	623.89	414.98	800.93
r hour....	342,909	263,950	172,776	121,474	220,284
er h., c. ft.	338.76	260.83	187.506	139.99	97.76
linder (*Le*).	15.80	16.47	15.28	14.24	21.61
comp.-cyl.	24,813	18,471	12,770	10,140	11,151
steam-cyl.	21.703	16,026	11,307	8,530	10,194
am.........	1,100.8	785.6	564.9	435.82	512.12

ing the Cold. (M. C. Bannister, Liverpool most useful means for applying the cold to various ition of brine or chloride of magnesium, which ir. The brine is first cooled by being circulated in rator-tubes, and then distributed through coils of the substances requiring a reduction of tempera- s or rooms prepared for them; the air coming in es is immediately chilled, and the moisture in the s. It then falls, making room for warmer air, and ole room is at the temperature of the brine in the

it for refrigerating made by the Linde British Re- brine is circulated through a shallow trough, in of shafts, each geared together, and driven by me- shafts are fixed a number of wrought-iron disks, rine, which cool them down to the brine tempera- r these disks a rapid circulation of air is passed by ntact with the plates; then it is led into the cham- on, from which it is again drawn by the same fan; purities are removed from the chambers, and de- ucing the most perfect antiseptic atmosphere yet ; while the maximum efficiency of the brine tem- lable, the brine being periodically concentrated by

as the circulating medium. The ammonia-pipes oling-chamber, and large wooden conduits are used i it from the rooms to be cooled. An advantage of room may be refrigerated more quickly than by g air deposits its moisture in the form of snow on h is removed by mechanical brushes.

IAL ICE-MANUFACTURE.

ns, with condensing water at 70°, artificial ice-ma- about 190 lbs. above the atmosphere condenser- ion-pressure.
of machine the useful circulation of ammonia, cylinder-heating, is about 13 lbs. per hour per in- he steam cylinder. This weight of ammonia pro- at 15° from water at 70°. If the ice is made from "can system," the amount of the latter supplied 3% greater than the weight of ice obtained. This escaping to the atmosphere, from the re-boiler and fy the distilled water, or free it from air; also, the ips, and loss by melting of the ice in extr
l steam consumed per horse-power is.
. *About 7.0 lbs. of this covers the stea*
driving the brine circulating-pumps,

F. in a ton of 2240 lbs. For fresh water the divisor is 35.93. The U. S. register tonnage will equal the displacement when the entire internal cubic capacity bears to the displacement the ratio of 100 to 35.

The displacement or gross tonnage is sometimes approximately estimated as follows: Let L denote the length in feet of the boat, B its extreme breadth in feet, and D the mean draught in feet; the product of these three dimensions will give the volume of a parallelopipedon in cubic feet. Putting V for this volume, we have $V = L \times B \times D$.

The volume of displacement may then be expressed as a *percentage* of the volume V, known as the "*block coefficient*." This percentage varies for different classes of ships. In racing yachts with very deep keels it varies from 22 to 33; in modern merchantmen from 55 to 75; for ordinary small boats probably 50 will give a fair estimate. The volume of displacement in cubic feet divided by 35 gives the displacement in tons.

Coefficient of Fineness.—A term used to express the relation between the displacement of a ship and the volume of a rectangular prism or box whose lineal dimensions are the length, breadth, and draught of the ship.

Coefficient of fineness $= \dfrac{D \times 35}{L \times B \times W}$; D being the displacement in tons of 35 cubic feet of sea-water to the ton, L the length between perpendiculars, B the extreme breadth of beam, and W the mean draught of water, all in feet.

Coefficient of Water-lines.—An expression of the relation of the displacement to the volume of the prism whose section equals the midship section of the ship, and length equal to the length of the ship.

Coefficient of water-lines $= \dfrac{D \times 35}{\text{area of immersed water section} \times L}$. Seaton gives the following values:

	Coefficient of Fineness.	Coefficient of Water-lines.
Finely-shaped ships	0.55	0.63
Fairly-shaped ships	0.61	0.67
Ordinary merchant steamers for speeds of 10 to 11 knots	0.65	0.72
Cargo steamers, 9 to 10 knots	0.70	0.76
Modern cargo steamers of large size	0.78	0.83

Resistance of Ships.—The resistance of a ship passing through water may vary from a number of causes, as speed, form of body, displacement, midship dimensions, character of wetted surface, fineness of lines, etc. The resistance of the water is twofold: 1st. That due to the displacement of the water at the bow and its replacement at the stern, with the consequent formation of waves. 2d. The friction between the wetted surface of the ship and the water, known as skin resistance. A common approximate formula for resistance of vessels is

Resistance $=$ speed$^2 \times \sqrt[3]{\text{displacement}^2} \times$ a constant, or $R = S^2D^{\frac{2}{3}} \times C$.

If $D =$ displacement in pounds, $S =$ speed in feet per minute, $R =$ resistance in foot-pounds per minute, $R = CS^2D^{\frac{2}{3}}$. The work done in overcoming the resistance through a distance equal to S is $R \times S = CS^3D^{\frac{2}{3}}$; and if E is the efficiency of the propeller and machinery combined, the indicated horse-power I.H.P. $= \dfrac{CS^3D^{\frac{2}{3}}}{E \times 33{,}000}$.

If $S =$ speed in knots, $D =$ displacement in tons, and C a constant which includes all the constants for form of vessel, efficiency of mechanism, etc., I.H.P. $= \dfrac{S^3D^{\frac{2}{3}}}{C}$.

The wetted surface varies as the cube root of the square of the displacement; thus, let L be the length of edge of a cube just immersed, whose displacement is D and wetted surface W. Then $D = L^3$ or $L = \sqrt[3]{D}$, and $W = 5 \times L^2 = 5 \times (\sqrt[3]{D})^2$. That is, W varies as $D^{\frac{2}{3}}$.

cans	1 to 1.2
ensers to distilled water...........	26 to 1
lers per pound of coal......	8.085
mpressor-engines....	444
from cans........................	2.2

EAM IN PER CENTS OF TOTAL AMOUNT.

....	60.1
ndensers....	19.7
vator engines.........,....	7.6
ater....	6.5
ght at boilers.................	5.6
ans.........	0.5

the purity of the ice are thus described: he condenser is the accumulation of the and engines, together with an amount of y from the boilers. This last quantity is amount of water necessary to supply the the condensers is violently reboiled, and ugh a coil surface-cooler. It then passes ich it runs through three charcoal-filters nd containing 28 feet of charcoal. It next ich there is an electrical attachment for ests are also made for salt daily. From which are carefully covered so that the impurities.

NGINEERING.

mensions and Obtaining Ton- merican & Foreign Shipping. American he dimensions to be measured as follows: de of stem to the after side of stern-post per deck of all vessels, except those hav- tending right fore and aft, in which the nge of deck immediately below the hurri-

king forward, or receding stems, or rak- e distance of the fore side of stem from load water-line measured at middle line. aken as stern-post in screw-steamers.
over the widest frame at its widest part; th.
at the dead-flat frame and at middle line from the top of floor-plate to the upper ssels except those having a continuous fore and aft, and not intended for the the depth is to be the distance from top top of hurricane deck-beam and the top tely below hurricane-deck.
us hurricane deck, extending right fore erican coasting trade, the depth is to be te to top of deck-beam of deck immedi-

nnage.—Multiply together the length. luct by .75; divide the last product by 100;

$$\frac{L \times B \times D \times .75}{100} = \text{tonnage.}$$

Tonnage Law, May 6, 1864, provides vessel shall be her entire internal cubic ach." This measurement includes all the ever many there may be. Explicit direc- ts are given in the law.
Vessel (measured in tons of 2240 lb...
r which it displaces. For sea-wa...
l beneath the water-line, in cu...
number of cubic feet of sea-wa...

quantity in the parenthesis, which is known as the "coefficient of augmentation." The last term of the coefficient may be neglected in calculating the resistance of ships as too small to be practically important. In applying the formula, the mean of the squares of the sines of the angles of maximum obliquity of the water-lines is to be taken for $\sin^2 \theta$, and the rule will then read thus:

To obtain the resistance of a ship of good form, in pounds, multiply the length in feet by the mean immersed girth and by the coefficient of augmentation, and then take the product of this "augmented surface," as Rankine termed it, by the square of the speed in knots, and by the proper constant coefficient selected from the following:

For clean painted vessels, iron hulls........	$A = .01$
For clean coppered vessels..........	$A = .009$ to $.008$
For moderately rough iron vessels.........	$A = .011 +$

The net, or effective, horse-power demanded will be quite closely obtained by multiplying the resistance calculated, as above, by the speed in knots and dividing by 326. The gross, or indicated, power is obtained by multiplying the last quantity by the reciprocal of the efficiency of the machinery and propeller, which usually should be about 0.6. Rankine uses as a divisor in this case 200 to 260.

The form of the vessel, even when designed by skilful and experienced naval architects, will often vary to such an extent as to cause the above constant coefficients to vary somewhat; and the range of variation with good forms is found to be from 0.8 to 1.5 the figures given.

For well-shaped iron vessels, an approximate formula for the horse-power required is H.P. $= \frac{SV^3}{20,000}$, in which S is the "augmented surface." The expression $\frac{SV^3}{\text{H.P.}}$ has been called by Rankine the *coefficient of propulsion*. In the Hudson River steamer "Mary Powell," according to Thurston, this coefficient was as high as 23,500.

The expression $\frac{D^{\frac{2}{3}}V^3}{\text{H.P.}}$ has been called the *locomotive performance*. (See Rankine's Treatise on Shipbuilding, 1864; Thurston's Manual of the Steam-engine, part ii, p. 16; also paper by F. T. Bowles, U.S.N., Proc. U. S. Naval Institute, 1883.)

Rankine's method for calculating the resistance is said by Seaton to give more accurate and reliable results than those obtained by the older rules, but it is criticised as being difficult and inconvenient of application.

Dr. Kirk's Method.—This method is generally used on the Clyde. The general idea proposed by Dr. Kirk is to reduce all ships to so definite and simple a form that they may be easily compared; and the magnitude of certain features of this form shall determine the suitability of the ship for speed, etc.

The form consists of a middle body, which is a rectangular parallelopiped, and fore body and after body, prisms having isosceles triangles for bases, as shown in Fig. 168.

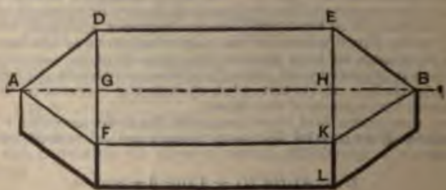

FIG. 168.

This is called a block model, and is such that its length is equal to that of the ship, the depth is equal to the mean draught, the capacity equal to the displacement volume, and its area of section equal to the area of the mersed midship section. The dimensions of the block model may be obtained as follows:

Let $AG = HB =$ length of fore- or after-body $= F$;
$GH =$ length of middle body $= M$;
$KL =$ mean draught $= H$;

$$EK = \frac{\text{area of immersed midship section}}{KL} = B.$$

Volume of block $= (F + M) \times B \times H$;
Midship section $= B \times H$;
Displacement in tons = volume in cubic ft. ÷ 35.

$$AH = AG + GH = F + M = \text{displacement} \times 35 \div (B \times H).$$

The wetted surface of the block is nearly equal to that of the ship of the same length, beam and draught; usually 2% to 5% greater. In exceedingly fine hollow-line ships it may be 8% greater.

Area of bottom of block $= (F + M) \times B$;
Area of sides $= 2M \times H$.

$$\text{Area of sides of ends} = 4\sqrt{F^2 + \left(\frac{B}{2}\right)^2} \times H;$$

$$\text{Tangent of half angle of entrance} = \frac{\frac{1}{2}B}{F} = \frac{B}{2F}.$$

From this, by a table of natural tangents, the angle of entrance may be obtained:

	Angle of Entrance of the Block Model.	Fore-body in parts of length.
Ocean-going steamers, 14 knots and upward.	18° to 15°	.3 to .36
" " 12 to 14 knots........	21 to 18	.26 to .3
" cargo steamers, 10 to 12 knots..	30 to 22	.22 to .26

E. R. Mumford's Method of Calculating Wetted Surfaces is given in a paper by Archibald Denny, *Eng'g*, Sept. 21, 1894. The following is his formula, which gives closely accurate results for medium draughts, beams, and finenesses:

$$S = (L \times D \times 1.7) + (L \times B \times C),$$

in which $S =$ wetted surface in square feet;
$L =$ length between perpendiculars in feet;
$D =$ middle draught in feet;
$B =$ beam in feet;
$C =$ block coefficient.

The formula may also be expressed in the form $S = L(1.7D + BC)$.
In the case of twin-screw ships having projecting shaft-casings, or in the case of a ship having a deep keel or bilge keels, an addition must be made for such projections. The formula gives results which are in general much more accurate than those obtained by Kirk's method. It underestimates the surface when the beam, draught, or block coefficients are excessive; but the error is small except in the case of abnormal forms, such as stern-wheel steamers having very excessive beams (nearly one fourth the length), and to very full block coefficients. The formula gives a surface about 6% too small for such forms.

To Find the Indicated Horse-power from the Wetted Surface. (Seaton.)—In ordinary cases the horse-power per 100 feet of wetted surface may be found by assuming that the rate for a speed of 10 knots is 5, and that the quantity varies as the cube of the speed. For example: To find the number of I.H.P. necessary to drive a ship at a speed of 15 knots, having a wetted skin of block model of 16,200 square feet:

The rate per 100 feet $= (15/10)^3 \times 5 = 16.875$.
Then I.H.P. required $= 16.875 \times 162 = 2734$.

of the block model may be obtained

after-body $= F$;
$= M$;
$= H$;
ship section $= B$.

$\times B \times H$;

me in cubic ft. $\div$ 35.

= displacement $\times$ 35 $\div$ ($B \times H$).

nearly equal to that of the ship of the usually 2% to 3% greater. In exceedingly reater.

ck $= (F + M) \times B$;
H.

$= 4\sqrt{F^2 + \left(\frac{B}{2}\right)^2} \times H$;

of entrance $= \frac{\frac{1}{2}B}{F} = \frac{B}{2F}$.

tangents, the angle of entrance may be

	Angle of Entrance of the Block Model.	Fore-body in parts of length.
upward.	18° to 15°	.3 to .36
s........	21 to 18	.26 to .3
knots..	30 to 22	.22 to .26

of Calculating Wetted Surfaces nny, *Eng'g*, Sept. 21, 1894. The following accurate results for medium draughts,

$1.7) + (L \times B \times C)$,

re feet;
diculars in feet;
;

d in the form $S = L(1.7D + BC)$.
ving projecting shaft-casings, or in the bilge keels, an addition must be made gives results which are in general much by Kirk's method. It underestimates t, or block coefficients are excessive; but of abnormal forms, such as stern-wheel ms (nearly one fourth the length), and e formula gives a surface about 6% too

orse-power from the Wetted cases the horse-power per 100 feet of uming that the rate for a speed of 10 ies as the cube of the speed. For ex cessary to drive a ship at a speed model of 16,200 square feet:

$= (15/10)^3 \times 5 = 16.875$,
$= 16.875 \times 162 = 2734$.

When the ship is exceptionally well-proportioned, the bottom quite clean, and the efficiency of the machinery high, as low a rate as 4 I.H.P. per 100 feet of wetted skin of block model may be allowed

The gross indicated horse-power includes the power necessary to overcome the friction and other resistance of the engine itself and the shafting, and also the power lost in the propellor. In other words, I.H.P. is no measure of the resistance of the ship, and can only be relied on as a means of deciding the size of engines for speed, so long as the efficiency of the engine and propellor is known definitely, or so long as similar engines and propellers are employed in ships to be compared. The former is difficult to obtain, and it is nearly impossible in practice to know how much of the power shown in the cylinders is employed usefully in overcoming the resistance of the ship. The following example is given to show the variation in the efficiency of propellers:

	Knots.		I.H.P.
H.M.S. "Amazon," with a 4-bladed screw, gave............	12.064	with	1940
H.M.S. "Amazon," with a 2-bladed screw, increased pitch, and less revolutions per minute............................	12.396	"	1663
H.M.S. "Iris," with a 4-bladed screw........................	16.577	"	7503
H.M.S. "Iris," with 2-bladed screw, increased pitch, less revolutions per knot.....................................	18.587	"	7556

Relative Horse-power Required for Different Speeds of Vessels. (Horse-power for 10 knots = 1.)—The horse-power is taken usually to vary as the cube of the speed, but in different vessels and at different speeds it may vary from the 2.8 power to the 3.5 power, depending upon the lines of the vessel and upon the efficiency of the engines, the propeller, etc.

Speed, knots.	4	6	8	10	12	14	16	18	20	22	24	26	28	30
HP $\propto$														
$S^{2.8}$	.0769	.239	.535	1.	1.666	2.565	3.729	5.185	6.964	9.095	11.60	14.52	17.87	21.67
$S^{2.9}$	.0701	.227	.524	1.	1.697	2.653	3.908	5.499	7.464	9.841	12.67	15.97	19.80	24.19
S^{3}	.0640	.216	.512	1.	1.728	2.744	4.096	5.832	8.	10.65	13.82	17.58	21.95	27.
$S^{3.1}$	.0584	.205	.501	1.	1.760	2.838	4.293	6.185	8.574	11.52	15.09	19.34	24.33	30.14
$S^{3.2}$	.0533	.195	.490	1.	1.792	2.935	4.500	6.559	9.189	12.47	16.47	21.28	26.97	33.63
$S^{3.3}$	.0486	.185	.479	1.	1.825	3.036	4.716	6.957	9.849	13.49	17.98	23.41	29.90	37.54
$S^{3.4}$	.0444	.176	.468	1.	1.859	3.139	4.943	7.378	10.56	14.60	19.62	25.76	33.14	41.90
$S^{3.5}$	.0405	.167	.458	1.	1.893	3.247	5.181	7.824	11.31	15.79	21.42	28.34	36.73	46.77

EXAMPLE IN USE OF THE TABLE.—A certain vessel makes 14 knots speed with 587 I.H.P. and 16 knots with 900 I.H.P. What I.H.P. will be required at 18 knots, the rate of increase of horse-power with increase of speed remaining constant? The first step is to find the rate of increase, thus: $14^x : 16^x :: 587 : 900$.

$$x \log 16 - x \log 14 = \log 900 - \log 587;$$
$$x(0.204120 - 0.146128) = 2.954243 - 2.768638,$$

whence x (the exponent of S in formula H.P. $\propto S^x$) = 3.2.

From the table, for $S^{3.2}$ and 16 knots, the I H.P. is 4.5 times the I.H.P. at 10 knots, ∴ H.P. at 10 knots = 900 ÷ 4.5 = 200.

From the table, for $S^{3.2}$ and 18 knots, the I.H.P. is 6.559 times the I.H.P. at 10 knots; ∴ H.P. at 18 knots = 200 × 6.559 = 1312 H P.

Resistance per Horse-power for Different Speeds. (One horse-power = 33,000 lbs. resistance overcome through 1 ft. in 1 min.)—The resistances per horse-power for various speeds are as follows: For a speed of 1 knot, or 6080 feet per hour = 101⅓ ft. per min., 33,000 ÷ 101⅓ = 325.658 lbs. per horse-power; and for any other speed 325.658 lbs. divided by the speed in knots; or for

1 knot	325.66 lbs.	6 knots	54.28 lbs.	11 knots	29.61 lbs.	16 knots	20.35 lbs
2 knots	162.83 "	7 "	46.52 "	12 "	27.14 "	17 "	19.16 "
3 "	108.55 "	8 "	40.71 "	13 "	25.05 "	18 "	18.09 "
4 "	81.41 "	9 "	36.18 "	14 "	23.26 "	19 "	17.14 "
5 "	65.13 "	10 "	32.57 "	15 "	21.71 "	20 "	[illegible] "

of Steam-vessels of Various Sizes.
eaton's Marine Engineering.)

	S.S. "Torpedo."	P.S. "John Penn."	S.S. "Africa."	P.S. "Mary Powell"	S.S. "Harrar."	R.M.P.S. "Connaught."
......	90′ 0″	171′ 9″	130′ 0″	286′ 0″	230′ 0″	3:7′ 0″
......	10′ 6″	18′ 9″	21′ 0″	34′ 3″	29′ 0″	35′ 0″
......	2′ 6″	6′ 9½″	8′ 10″	6′ 0″	13′ 6″	13′ 0″
......	29.73	280	370	800	1500	1900
n....	24?	99	148	200	340	336
......	903	3793	3754	8222	10,075	15,782
......	45′ 0″	72′ 00″	42′ 6″	143′ 0″	79′ 6″	129′ 0″
......	12° 40′	11° 30′	23° 50′	13° 21′	17° 0″	11° 26′
......	0.481	0.576	0.608	0.489	0.671	0.605
......	22 01	15.3	10.74	17.20	10.04	17.8
......	460	798	371	1490	503	4751
in....	50.9	21.04	9.88	18.12	5.00	30.00
in, re-	4.78	5.87	7.97	3.56	4.90	5.32
......	223	192	172.8	293.7	266	182
......	556?	445	495	683	690	399

	H.M.S. "Active."	H.M.S. "Iris."	H.M.S. "Iris."	S.S. "Garonne."	H.M.S. "Hecla."	R.M.S.S. "Britannic."
......	270′ 0″	300′ 0″	300′ 0″	370′ 0″	392 0″	450′ 0″
......	42′ 0″	46′ 0″	46′ 0″	41′ 0″	39 0″	45′ 2″
......	18′ 10″	18′ 2″	18′ 2″	18′ 11″	21′ 4″	23′ 7″
......	3057	3290	3:90	4635	5767	8500
......	632	700	700	656	738	926
......	16,008	18,168	18,168	22,633	26,235	32,578
......	101′ 0′	135′ 6″	135′ 6″	123′ 0″	118′ 0″	129′ 0″
......	18° 44′	16° 16′	16° 16′	16° 4′	16° 30′	17° 16′
......	0.629	0.548	0.548	0.668	0.698	0.714
......	14.966	18.573	15.746	13.80	12.054	15.045
......	4015	7714	3958	2500	1758	4900
in....	25.08	42.46	21.78	11.04	6.7	15.04
in, re-	7.49	6.634	5.58	4.20	3.83	4.42
......	175.8	183.7	218.2	292	[illegible]	[illegible]
......	527.5	581.4	690.5	689	[illegible]	[illegible]

Results of Progressive Speed Trials in Typical Ves
(*Eng'g*, April 15, 1892, p. 463.)

	Torpedo-boat.	Torpedo-gunboat, "Sharp-shooter" Class.	"Medusa," 3d-cl. Cruiser.	"Terpsichore," 2d-cl. Cruiser.	"Edgar," 1st-cl. Cruiser.	"Blenheim," 1st-cl. Cruiser.
Length (in feet)	135	230	265	300	360	[illegible]
Breadth " "	14	27	41	43	60	[illegible]
Draught (mean) on trial	5′ 1″	8′ 3″	16′ 6″	16′ 2″	23′ 9″	25′ [illegible]
Displacement (tons)	103	735	2800	3330	7390	[illegible]
I.H.P.—10 knots	110	450	700	800	1000	[illegible]
" 14 "	260	1100	2100	2400	3000	[illegible]
" 18 "	870	2500	6400	6000	7500	[illegible]
" 20 "	1130	3500	10000	9000	11000	[illegible]
Speed 10, Ratio of speed³ 1., Ratio of H.P. =	1	1	1	1	1	1
Speed 14, Ratio of speed³ 2.744, " " =	2.36	2.44	3	3	3	[illegible]
Speed 18, Ratio of speed³ 5.832, " " =	7.91	5.56	9.14	7.5	7.5	[illegible]
Speed 20, Ratio of speed³ 8., " " =	10.27	7.78	14.14	11.25	11	[illegible]
Admiralty coeff. $C = \frac{D^{\frac{2}{3}} \times S^3}{\text{I.H.P.}}$ 10 knots	200	181	284	279	380	[illegible]
14 "	232	203	259	255	347	[illegible]
18 "	147	190	181	217	295	[illegible]
20 "	156	186	159	198	276	[illegible]

The figures for I.H.P. are "round." The "Medusa's" figures fo
are from trial on Stokes Bay, and show the retarding effect of shall
The figures for the other ships for 20 knots are estimated for deep

More accurate methods than those above given for estim
horse-power required for any proposed ship are: 1. Estimations
from the results of trials of "similar" vessels driven at "corres
speeds; "similar" vessels being those that have the same ratio of
breadth and to draught, and the same coefficient of fineness, an
sponding" speeds those which are proportional to the square
the lengths of the respective vessels. Froude found that the resis
such vessels varied almost exactly as wetted surface × (speed)².

2. The method employed by the British Admiralty and by so
shipbuilders, viz., ascertaining the resistance of a model of the ve
20 ft. long, in a tank, and calculating the power from the results o

Speed on Canals.—A great loss of speed occurs when a ste
passes from open water into a more or less restricted channel. Th
speed of vessels in the Suez Canal in 1882 was only $5\frac{1}{2}$ statute miles
(*Eng'g*, Feb. 15, 1884, p. 139.)

Estimated Displacement, Horse-power, etc.—The
the next page, calculated by the author, will be found convenient
ing approximate estimates.

The figures in 7th column are calculated by the formula H.P. =
in which $c = 200$ for vessels under 200 ft. long when $C =$ .6
when $C = .55$; $c = 200$ for vessels 200 to 400 ft. long when $C =$ .7
$C = .65$, 240 when $C = .55$; $c = 230$ for vessels over 400 ft. long wh
250 when $C = .65$, 260 when $C = .55$.

The figures in the 8th column are based on 5 H.P. per 100 sq. ft
surface.

The diameters of screw in the 9th column are from fo
$3.31 \sqrt[5]{\text{I.H.P.}}$, and in the 10th column from formula $D = 2.71 \sqrt[5]{\text{I}}$

To find the diameter of screw for any other speed than 10 kn
tions being 100 per minute, multiply the diameter given in the t
5th root of the cube of the given speed ÷ 10. For any other rev
minute than 100, divide by the revolutions and multiply by 100.

To find the [illegible] *imate horse-power for any other speed th*
multiply [illegible] *wer given in the table by the cube of the*
[illegible] *the relative figure from table on p. 1008.*

ent, Horse-power, etc., of Steam-
s of Various Sizes.

Wetted Surface $L(1.7D + BC)$ sq. ft.	Estimated Horse-power at 10 knots.		Diam. of Screw for 10 knots speed and 100 revs. per minute.	
	Calc. from Displacem't.	Calc. from Wetted Surface.	If Pitch = Diam.	If Pitch = 1.4 Diam.
48	4.3	2.4	4.4	3.6
64	5.2	3.2	4.6	3.8
96	8.9	4.8	5.1	4.2
80	6.0	4.0	4.7	3.9
120	10.3	6.0	5.3	4.3
104	7.5	5.2	5	4.1
152	12.6	7.6	5.5	4.5
168	11.5	8.4	5.4	4.4
224	18.2	11.2	5.9	4.8
235	15.1	11.8	5.7	4.7
326	24.9	16.3	6.3	5.2
420	27.8	21.0	6.4	5.4
558	43.9	27.9	7.1	5.8
621	42.2	31.1	7.0	5.7
798	62.9	39.9	7.6	6.2
861	59.4	43.1	7.5	6.1
1082	85.1	54.1	8.1	6.6
1140	79.2	57.0	7.9	6.5
1408	111	70.4	8.5	7.0
1408	97	70.4	8.3	6.8
1854	147	92.7	9	7.3
1565	104	78.3	8.4	6.9
1910	143	95.5	8.9	7.3
2295	202	115	9.6	7.8
2046	131	102	8.8	7.2
2472	179	124	9.4	7.6
2946	250	147	10	8.2
2660	169	133	9.2	7.4
3185	227	159	9.8	8.0
3766	312	188	10.5	8.5
3264	203	168	9.6	7.8
3880	269	194	10.1	8.3
4560	368	228	10.8	8.8
4122	257	206	10.1	8.2
4869	337	243	10.6	8.7
5688	455	284	11.3	9.2
4800	257	240	10.1	8.2
5970	373	299	10.8	8.8
7260	526	363	11.6	9.5
7250	383	363	10.9	8.9
9450	592	473	11.9	9.7
11850	875	593	12.8	10.5
10380	548	519	11.7	9.6
13140	806	657	12.6	10.4
17140	1175	857	13.6	11.1
14455	769	723	12.5	10.2
17885	1111	894	13.5	11.0
21595	1562	1080	14.4	11.8
19200	1028	960	13.3	10.8
23360	1451	1168	14.2	11.6
27840	2006	1392	15.2	12.4
24515	1221	1226	13.7	11.2
29565	1616	1478	14.5	11.9
34875	2171	1744	15.4	12.6
29600	1454	1480	14.2	11.6
35200	1966	1760	15.1	12.4
41200	2543	2060	15.9	12.0
36245	1747	1812	14.7	[illegible]
42735	2266	2137	15.5	[illegible]
49665	2908	2483	16.[illegible]	[illegible]
42900	2065	2145	15[illegible]	[illegible]
50220	2656	2511	1[illegible]	[illegible]
58020	3489	2901	1[illegible]	[illegible]

THE SCREW-PROPELLER.

The "pitch" of a propeller is the distance which any point in a blade, describing a helix, will travel in the direction of the axis during one revolution, the point being assumed to move around the axis. The pitch of a propeller with a uniform pitch is equal to the distance a propeller will advance during one revolution, provided there is no slip. In a case of this kind, the term "pitch" is analogous to the term "pitch of the thread" of an ordinary single-threaded screw.

Let P = pitch of screw in feet, R = number of revolutions per second, V = velocity of stream from the propeller = $P \times R$, v = velocity of the ship in feet per second, $V - v$ = slip, A = area in square feet of section of stream from the screw, approximately the area of a circle of the same diameter, $A \times V$ = volume of water projected astern from the ship in cubic feet per second. Taking the weight of a cubic foot of sea-water at 64 lbs., and the force of gravity at 32, we have from the common formula for force of acceleration, viz.: $F = M\frac{v_1}{t} = \frac{W}{g}\frac{v_1}{t}$, or $F = \frac{W}{g}v_1$, when $t = 1$ second, v_1 being the acceleration.

$$\text{Thrust of screw in pounds} = \frac{64AV}{32}(V - v) = 2AV(V - v).$$

Rankine (Rules, Tables, and Data, p. 275) gives the following: To calculate the thrust of a propelling instrument (jet, paddle, or screw) in pounds, multiply together the transverse sectional area, in square feet, of the stream driven astern by the propeller; the speed of the stream relatively to the ship in knots; the real slip, or part of that speed which is impressed on that stream by the propeller, also in knots; and the constant 5.66 for sea-water, or 5.5 for fresh water. If S = speed of the screw in knots, s = speed of ship in knots, A = area of the stream in square feet (of sea-water),

$$\text{Thrust in pounds} = A \times S(S - s) \times 5.66.$$

The real slip is the velocity (relative to water at rest) of the water projected sternward; *the apparent slip* is the difference between the speed of the ship and the speed of the screw; i.e., the product of the pitch of the screw by the number of revolutions.

This apparent slip is sometimes negative, due to the working of the screw in disturbed water which has a forward velocity, following the ship. Negative apparent slip is an indication that the propeller is not suited to the ship.

The apparent slip should generally be about 8% to 10% at full speed in well-formed vessels with moderately fine lines; in bluff cargo boats it rarely exceeds 5%.

The effective area of a screw is the sectional area of the stream of water laid hold of by the propeller, and is generally, if not always, greater than the actual area, in a ratio which in good ordinary examples is 1.2 or thereabouts, and is sometimes as high as 1.4; a fact probably due to the stiffness of the water, which communicates motion laterally amongst its particles. (Rankine's Shipbuilding, p. 89.)

Prof. D. S. Jacobus, Trans. A. S. M. E., xi. 1028, found the ratio of the effective to the actual disk area of the screws of different vessels to be as follows:

Tug-boat, with ordinary true-pitch screw		1.42
" " screw having blades projecting backward		.57
Ferryboat "Bergen," with ordinary true-pitch screw	at speed of 12.09 stat. miles per hour.	1.53
	" " 13.4 " " " "	1.48
Steamer "Homer Ramsdell," with ordinary true-pitch screw		1.28

Size of Screw.—Seaton says: The size of a screw depends on so many things that it is very difficult to lay down any rule for guidance, and much must always be left to the experience of the designer, to allow for all the circumstances of each particular case. The following rules are given for ordinary cases. (Seaton and Rounthwaite's Pocket-book):

P = pitch of propeller in feet = $\frac{10133S}{R(100 - x)}$, in which S = speed in knots, R = revolutions per minute, and x = percentage of apparent slip.

For a slip of 10%, pitch = $\frac{112.6S}{R}$.

$\sqrt{\frac{\text{I.H.P.}}{\left(\frac{P \times R}{100}\right)^3}}$, K being a coefficient given

If $K = 20$, $D = 20000\sqrt{\frac{\text{I.H.P.}}{(P \times R)^3}}$.

les $= C\sqrt{\frac{\text{I.H.P.}}{R}}$, in which C is a coefficient

, given in Seaton's Marine Engineering, is

$=$ 737 for ordinary vessels, and 660 for slow-

nes.

$= \sqrt{\frac{d^3}{nb}} \times k$, in which d = diameter of tail-

f blades, b = breadth of blade in inches where
allel to the shaft axis; k = 4 for cast iron, 1.5
1.5 for high-class bronze.
ast iron $.04D + .4$ in.; cast steel $.03D + .4$ in.;
lass bronze $.02D + .3$ in., where D = diameter

eller Coefficients.

Number of Screws.	Number of Blades per Screw.	Values of K.	Values of C.	Usual Material of Blades.
One	4	17 –17.5	19 –17.5	Cast iron
"	4	18 –19	17 –15.5	" "
"	4	19.5–20.5	15 –13	C. I. or S.
Twin	4	20.5–21.5	14.5–12.5	" " "
One	4	21 –22	12.5–11	G. M. or B
Twin	3	22 –23	10.5– 9	" " "
"	4	21 –22.5	11.5–10.5	" " "
"	3	22 –23.5	8.5– 7	" " "
One	3	25	7– 6	B. or F. S.

etal; B., bronze; S., steel; F. S., forged steel.

$\sqrt{\frac{\text{I.H.P.}}{(P \times R)^3}}$ and $P = \frac{737}{R}\sqrt[3]{\frac{\text{I.H.P.}}{D^2}}$, if $P = D$

$\sqrt{400 \times \text{I.H.P.}} = 3.31\sqrt[5]{\text{I.H.P.}}$

n $D = \sqrt[5]{145.8 \times \text{I.H.P.}} = 2.71\sqrt[5]{\text{I.H.P.}}$
figures for diameter of screw in the table on
l. They may be used as rough approximations
rew for any given horse-power, for a speed of
r minute.
evolutions per minute multiply the figures in
the given number of revolutions. For any
e the I.H.P. varies approximately as the cube
r of the screw as the 5th root of the I.H.P.,
for 10 knots by the 5th root of the cube of one
multiply by the following factors:

8	9	11	12	13	14
.875	.939	1.059	1.116	1.170	1.224

Speed:	17	18	19	20	21	22	23	24	25	26	27	28
$\sqrt[5]{(S+10)^3}$ =	1.375	1.423	1.470	1.515	1.561	1.605	1.648	1.691	1.733	1.774	1.815	1.8

For more accurate determinations of diameter and pitch of screw, t formulæ and coefficients given by Seaton, quoted above, should be used.

Efficiency of the Propeller.—According to Rankine, if the slip the water be s, its weight W, the resistance R, and the speed of the ship

$$R = \frac{Ws}{g}; \quad Rv = \frac{Wsv}{g}.$$

This impelling action must, to secure maximum efficiency of propeller, effected by an instrument which takes hold of the fluid without shock disturbance of the surrounding mass, and, by a steady acceleration, gives the required final velocity of discharge. The velocity of the propeller ov coming the resistance R would then be

$$\frac{v + (v + s)}{2} = v + \frac{s}{2};$$

and the work performed would be

$$R\left(v + \frac{s}{2}\right) = \frac{Wvs}{g} + \frac{Ws^2}{2g},$$

the first of the last two terms being useful, the second the minimum l work; the latter being the wasted energy of the water thrown backwa The efficiency is

$$E = v \div \left(v + \frac{s}{2}\right);$$

and this is the limit attainable with a perfect propelling instrument, wh limit is approached the more nearly as the conditions above prescribed the more nearly fulfilled. The efficiency of the propelling instrumen probably rarely much above 0.60, and never above 0.80.

In designing the screw-propeller, as was shown by Dr. Froude, the b angle for the surface is that of 45° with the plane of the disk; but as parts of the blade cannot be given the same angle, it should, where pra cable, be so proportioned that the "pitch-angle at the centre of effo should be made 45°. The maximum possible efficiency is then, accord to Froude, 77%.

In order that the water should be taken on without shock and dischar with maximum backward velocity, the screw must have an axially incre ing pitch.

The true screw is by far the more usual form of propeller, in all steame both merchant and naval. (Thurston, Manual of the Steam-engine, part p. 176.)

The combined efficiency of screw, shaft, engine, etc., is generally tak at 50%. In some cases it may reach 60% or 65%. Rankine takes the effect H.P. to equal the I.H.P. ÷ 1.63.

Pitch-ratio and Slip for Screws of Standard Form.

Pitch-ratio.	Real Slip of Screw.	Pitch-ratio.	Real Slip o Screw
.8	15.55	1.7	21.3
.9	16.22	1.8	21.8
1.0	16.88	1.9	22.4
1.1	17.55	2.0	22.9
1.2	18.2	2.1	23.5
1.3	18.8	2.2	24.0
1.4	19.5	2.3	24.5
1.5	20.1	2.4	[illegible]
1.6	20.7	2.5	[illegible]

archer on the efficiency of screw-propel-
tions, in a paper read before section G of
n, 1890. He states that the following gen-
hed:
of real slip at which, and at which only,
ned with a screw of any given type, and
-ratio. The slip-ratio proper to a given
et discovered and tabulated for a screw
table on page 1002:

besides being less efficient in themselves,
y an amount bearing some proportion to
amount of rotation left in the race.
ably between 1.1 and 1.5.
sel, the less the pitch-ratio should be.
ld be placed further from the stern than

natural result of abnormal proportions of

rred for high-speed vessels, but when the
ur or even more may be advantageously

an ellipse having a minor axis equal to

ews should increase from forward to aft,
tory results when the blades are narrow,
tion should be a function of the width of

f screw-shaft produces vibration, and with
g outwards, if the shafts are inclined at
ards from the propellers.
screw-propellers, see F. C. Marshall, Proc.
rans. Institution of Naval Architects, 1886;
of Naval Architects 1887; and S. W. Bar-
vol. cii.
lts deduced from experiments on model
e practically equal efficiencies throughout
nd in surface-ratio; so that great latitude
the form of the propeller. Another im-
hese experiments are not a direct guide to
propeller for a particular ship, they sup-
rformances of screws fitted to vessels, and
at are likely to be the best dimensions of
se results are known. Thus a great ad-
ethod of trial upon the ship itself, which
onceivable erroneous view respecting the
., July, 1891.)

DDLE-WHEEL.

dial Floats. (Seaton's Marine En-
er of a radial wheel is usually taken from
ut it is difficult to say what is absolutely
on the form of float, the amount of dip,
the wheel. The slip of a radial wheel is
on the size of float.

$$\text{e float} = \frac{\text{I.H.P.}}{D} \times C.$$

feet, and C is a multiplier, varying from
light steamers.
lly about ¼ its length, and its thickness
r of floats varies directly with the diam-
for every foot of diameter.
of the radial wheel, see Thurston, Manual
2.)
eels. (Seaton.) — The diam
s: The amount of slip varies
ats are small or the resistan

is as high as 25 per cent: a well-designed wheel on a well-formed ship should not exceed 15 per cent under ordinary circumstances.
If K is the speed of the ship in knots, S the percentage of slip, and R revolutions per minute,

$$\text{Diameter of wheel at centres} = \frac{K(100 + S)}{3.14 \times R}.$$

The diameter, however, must be such as will suit the structure of the ship, so that a modification may be necessary on this account, and revolutions altered to suit it.
The diameter will also depend on the amount of "dip" or immersion of float.
When a ship is working always in smooth water the immersion of the edge should not exceed 1/6 the breadth of the float; and for general service at sea an immension of 1/2 the breadth of the float is sufficient. If the ship is intended to carry cargo, the immersion when light need not be more than 2 or 3 inches, and should not be more than the breadth of float when at deepest draught; indeed, the efficiency of the wheel falls off rapidly with the immersion of the wheel.

$$\text{Area of one float} = \frac{\text{I.H.P.}}{D} \times C.$$

C is a multiplier, varying from 0.3 to 0.35; D is the diameter of the wheel to the float centres, in feet.

The number of floats $= \frac{1}{2}(D + 2)$.
The breadth of the float $= 0.35 \times$ the length.
The thickness of floats $= 1/12$ the breadth.
Diameter of gudgeons $=$ thickness of float.

Seaton and Rounthwaite's Pocket-book gives:

$$\text{Number of floats} = \frac{60}{\sqrt{R}},$$

where R is number of revolutions per minute.

$$\text{Area of one float (in square feet)} = \frac{\text{I.H.P.} \times 33000 \times K}{N \times (D \times R)^3},$$

where N = number of floats in one wheel.
For vessels plying always in smooth water $K = 1200$. For sea-going steamers $K = 1400$. For tugs and such craft as require to stop and start frequently in a tide-way $K = 1600$.
It will be quite accurate enough if the last four figures of the value of $(D \times R)^3$ be taken as ciphers.
For illustrated description of the feathering paddle-wheel see Seaton's Marine Engineering, or Seaton and Rounthwaite's Pocket-book. The diameter of a feathering-wheel is about one half that of a radial wheel for equal efficiency. (Thurston.)

Efficiency of Paddle-wheels.—Computations by Prof. T... of the efficiency of propulsion by paddle-wheels give for light river steamers with ratio of velocity of the vessel, v, to velocity of the paddle centre of pressure, V, or $\frac{v}{V}$, $= \frac{3}{4}$, with a dip $= 3/20$ radius of the wheel, a slip of 25 per cent, an efficiency of .714; and for ocean steamers with the same slip and ratio of $\frac{v}{V}$, and a dip $= \frac{1}{2}$ radius, an efficiency of ...

JET-PROPULSION.

Numerous experiments have been made in driving a vessel by the reaction of a jet of water pumped through an orifice in the stern, but they have all resulted in commercial failure. Two jet-propulsion boats, the "Waterwitch," 1100 tons, and the "Squirt," a small torpedo-boat, were built by the British Government. The former was tried in 1867 and gave an efficiency of apparatus of only 18 per cent. The latter gave a speed of 12 knots, as against 17 knots attained by a sister-ship having a screw and equal steam-power. The mathematical theory of the efficiency of the jet was discussed by Rankine in The Engineer, Jan. 11, 1867, and he showed the greater the quantity of water operated on by a jet-propeller, the ...

nce both of the theory and of the results of earlier the opinions of many naval engineers, more than 8-90 in New York upon two experimental boats, the Evolution," in which the jet was made of very small only ⅝-inch diameter, and with a pressure of 2500 had been predicted, the vessel was a total failure. in *Mechanics*, March, 1891.)

propeller is similar to that of the screw-propeller. in square feet, V its velocity with reference to the d, v = the velocity of the ship in reference to the the jet (see Screw-propeller, *ante*) is $2AV(V-v)$. essel is $2AV(V-v)v$, and the work wasted on the the jet is $\frac{1}{2} \times 2AV(V-v)^2$. The efficiency is $\overline{-v)^2} = \frac{2v}{V+v}$. This expression equals unity when velocity of the jet with reference to the earth, or thrust of the propeller is also 0. The greater the with v, the less the efficiency. For $V = 20v$, as was ion," the efficiency of the jet would be less than 10 be further reduced by the friction of the pumping ater in pipes.

ropulsion may be summed up in Rankine's words: best, other things being equal, which drives astern r at the lowest velocity."

ible to devise any system of hydraulic or jet propul- favorably, under these conditions, with the screw

t.—If a jet of water issues horizontally from a ves- side of the vessel opposite the orifice is equal to the water the section of which is the area of the orifice, he head.

n jet-propulsion is the reaction of the stream issuing it is the same whether the jet is discharged under , or against a solid wall. For proof, see account of , Jr., given by Prof. J. Burkitt Webb, Trans. A. S. M.

RACTICE IN MARINE ENGINES.

. Blechynden on Marine Engineering during the past cade, Proc. Inst. M. E., July, 1891.)

stage-expansion engine has become the rule, and the en increased to 160 lbs. and even as high as 200 lbs. per ge-expansion engines of various forms have also been

t has become the rule in all vessels for naval service, ommon in both passenger and cargo vessels. By this nsiderably to augment the power obtained from a ong as it is kept within certain limits it need result in , but when pushed too far the increase is sometimes able cost.

nomy of forced draught, an examination of the ap- 8) will show that while the mean consumption of coal king under natural draught is 1.573 lbs. per indicated it is only 1.336 lbs. in those fitted with forced draught. n economy of 15%. Part of this economy, however, other heat-saving appliances with which the latter

aterial for boilers, iron is now a thing of the past, ble that it will continue yet awhile to be the material s can be procured at 132 square feet superficial area For purely boiler work a punching-mach *ne-engine work.*

res of steam have also caused attentio *e led to the adoption of various artific* *nd spiral flues, with the object of gi* *e without abnormally increasing th* *ce-plate is viewed by many engin*

e added to each pound of the feed-water
would be 1005 units against 1125 units
expenditure of only 89.4% of the heat
e the expenditure of heat in relation
, equivalent to a heat economy of 3.6%.
en from the low-pressure receiver, the

fitted with Twin Screws.

	Beam.	Cylinders, two sets in all. Diameters.	Stro.	Boiler-pressure per sq. in.	Indicated Horse-power
	Feet	Inches	In.	Lbs.	I.H.P.
	63¼	45, 71, 113	60	150	20,000
	58	43, 68, 110	60	180	18,000
	57½	40, 67, 106	66	160	11,500
8½	55½	41, 66, 101	66	160	12,500
40	51	32, 51, 82	54	160	10,125
15	48	34, 54, 85	51	160	10,000
60	54½	34½, 57½, 92	60	170	11,656

of Working of Marine Engines, , 1881, and 1891.

d Coal.	1872.	1881.	1891.
....	52.4	77.4	158.5
ver, sq. ft.......	4.410	3.917	3.275
....	55.67	59.76	63.75
....	376	467	529
r, lbs	2.110	1.828	1.522

ge - expansion Engines in Nine n to Indicated Horse-power and .

Relative Weight of Machinery. Indicated Horse-power. ne-n.	Boiler-room.	Total	Engine-room per cu. ft. of Cylinder-capacity.	Boiler-room per 100 sq. ft. of Heating-surface.	Type of Machinery.
.	lbs.	lbs.	tons.	tons.	
	220	446	1.30	3.75	Mercantile
	251	510	1.46	4.10	"
	198	405	1.23	3.23	"
	203	373	1.29	3.30	"
	162	329	1.41	3.44	"
	202	343	1.37	3.3[illegible]	"
	108	185	1.21	[illegible]	
	116	194	1.11		
.5	102	165	0.82		

Particulars of Three-stage-expansion Engines in Twenty-eight Steamers. (A. Blechynden.)

No.	Cylinders. Diameter.			Cylinders. Stroke.	Condenser. Cooling Surface.	Propeller. Diameter.		Propeller. Pitch.		Boilers. Heating-surface.	Boilers. Fire-grate Area.	Boilers. Steam-pressure.	Results of Trials. Revolutions per minute.	Piston-speed, Ft. per min.	Indicated H.P.	Heating-surface. Per I.H.P.	Heating-surface. Per lb. of Coal per hour.	I.H.P. per sq. ft. of grate.	Coal burnt per sq. ft. of grate per hour.	Coal burnt per I.H.P. per hour.	Remarks.
	ins.			ins.	sq. ft.	ft.	in.	ft.	in.	sq. ft.	sq. ft.	lbs.	revs.	ft.	I.H.P.	sq. ft.	sq. ft.	I.H.P.	lbs.	lbs.	
1	40	66	100	72	11,586	22	0	28	6	17,640	626	155	52.2	627	4295	4.11	2.46	6.86	11.45	1.67	H
2	40	66	100	72	11,586	22	0	28	6	17,640	626	155	51.3	616	4402	4.04	2.55	7.03	11.14	1.584	H
3	39	61	97	66	11,000	20	10	26	0	15,107	540	155	57.3	630	3587	4.21	2.22	6.65	12.60	1.896	H
4	39	61	97	66	11,000	20	10	26	0	15,107	540	155	57.4	631	3822	3.95	2.14	7.08	13.02	1.841	H
5	23	38	61	42	2,008	16	0	17	6	3,972	133	160	61	427	1120	3.54	2.02	8 43	14.75	1.75	
6	25½	42	70	51	3,209	16	6	20	0	6,162	193	180	61.3	521	1700	3.62	2.40	8.82	13.25	1.505	
7	21	34	55½	36	1,447	14	0	17	6	3,350	99	160	64	384	900	3.72	2.31	9.09	14.67	1.612	H
8	22	35	59	39	1,430	15	6	15	6	3,324	102	160	70	455	1065	3.12	2.38	10 42	13.70	1.312	
9	29	45	74	54	3,900	19	6	20	0	6,875	240	160	56	504	2250	3.055	2.04	9.38	14.00	1.494	H
10	31	48	82	54	4,150	19	0	19	0	8,000	260	160	61.5	553	2600	3.075	2.04	10.00	15.10	1.505	H
11	25	41	67	48	2,800	......		......		4,645	142	160	58	464	1300	3.57	2.26	9.16	14.46	1.580	
12	21½	36	59	42	2,000	15	0	16	6	3,852	122	160	67	469	1100	3.50	2.29	9.02	13.79	1.529	
13	32	51	82	54	12,562	16	6	23	0	20,192	710	160	58.5	526	3670	5.50	3.64	5.17	7.80	1.510	H
14	27	44	71	48	2,800	17	9	17	6	6,164	220	150	63	504	1680	3.67	2.12	7.65	13.18	1.723	H
15	29	45	74	60	4,020	19	0	24	0	6,950	196	150	53.8	538	2360	2.94	1.78	12.03	19.85	1.650	
16	29	45	74	54	3,850	18	0	21	0	6,960	216	160	64	576	2550	2.73	1.82	11.80	17.70	1.500	
17	23	37	64	48	2,400	16	6	18	0	4,715	144	180	62	496	1500	3.14	2.00	10 40	16.31	1.568	
18	28	44	74	51	3,700	17	9	22	9	8,000	2 4	150	62	527	1727	4.63	2.85	10.53	17.06	1.620	
19	23	36½	58	36	2,218	15	0	15	6	3,271	126	160	76	456	1269	2.58	1.84	10.07	14.10	1.400	
20	17.17	38	60	42	2,900	15	6	15	6	4,400	168	150	75	525	1530	2.875	1.96	9.11	13.32	1.464	
21	25	39	62	36	2,700	14	0	16	3	4,000	150	160	73	[illegible]	1250	3.20	2.40	8.35	11.13	1.330	
22	31	46	72	51	3,713	16	3	22	6	5,076	110	150	72	612	2513	2.02	1.34	22 85	33.95	1.488	D H
23	22½	35½	58½	39	1,750	14	7	16	6	2,338	50	160	76	494	1350	1.73	1.28	27.00	36.42	1.350	D H
24	25	42	68½	48	2,763	16	10	17	9	4,346	84	160	65	520	1800	2.41	1 94	21.42	26.62	1.242	D H
25	22¼	35¾	58½	48	3,530	15	6	18	0	3,486	63	160	69.5	552	1360	2.56	1.91	21.59	28.90	1.338	D H
26	31	50	83	60	6,860	19	0	23	0	6,438	154	150	59	590	2600	2.435	1.78	16.88	23.05	1.365	D H
27	32	53	87½	60	7,500	19	0	23	9	8,751	210	160	66	660	3400	2.52	2.04	16.18	19.97	1.234	D H
28	28½	46	75	42	3,450	16	0	21	0	6,618	188	160	73	511	2058	3.215	2.05	17.10	10.92	1.565	
	Average of all twenty-eight															3.275	2.14	11.22	17.08	1.522	
	" " natural draught															3.560	2.25	8.91	13.92	1.573	
	" " forced draught															2.412	1.72	20.98	28.15	1.336	

d Horse-power, and Cylinder-stage-expansion Engines in Nine

oke	Revolutions per minute.	Boiler-pressure per sq. in.	Indicated Horse-power.	Cylinder-capacity.	Heating-surface.	
					Total.	Per I.H.P.
ns.	revs.	lbs.	I.H.P.	cu. ft.	sq. ft.	sq. ft.
2	64.5	160	6751	522	17,640	2.62
6	67.8	160	5525	436	15,107	2.73
2	83	160	1450	109	3,973	2.73
4	90	150	510	30	1,403	2.75
4	88	160	9625	508	20,193	2.10
7	113	150	1194	55	3,200	2.68
4	191	145	1265	36.3	2,227	1.76
4	182.5	140	2105	66.2	3,928	1.87
9	145	150	9400	319	15,882	1.62

TION OF BUILDINGS.*

ling Laws of the City of New York, 1893.)

es, Stores, Factories, and Stables.—en walls, not less than 12 in. to height of 40 ft.; ss than 16 in. to 40 ft., and 12 in. thence to top;
" " 20 " 25 " 16 " "
" " 24 " 20 ft.; 20 in. to 60 ft., and 16 in.

less than 28 in. to 25 ft.; 24 in. to 50 ft.; 20 in p;

additional 25 ft. in height, or part thereof, next e increased 4 inches in thickness, the upper 100 e as specified for a wall of that weight.

part, the bearing-walls shall be 4 inches thicker ry 12½ feet or fraction thereof that said walls

loors, Roofs, and Supports.

	Floors calculated to bear safely per sq. ft., in addition to their own weight.
t, apartment-house or hotel, not	70 lbs.
less than....	100 "
ilding, not less than............	120 "
ehouse, etc., not less than... ...	150 "
ss than...........	50 "

ficient strength to bear safely the weight to be n to the weight of the materials of which the

—The strength of all columns and posts shall ordon's formulæ, and the crushing weights in f section, for the following-named materials, nts in said formulæ, namely: Cast iron, 80,000;

forbid any extended treatment of this subject. upon it will be found in Trautwine's Civil Engi- idder's Architect's and Builder's Pocket-book. tions the following works of reference: "Notes vols., Rivingtons, publishers, Boston; "Building Clark (J. R. Osgood & Co., Boston.); "The " by R. G. Hatfield; "Graphical Analysis of Greene; "The Fire Protection of Mills," b- Drainage and Water Service," by Ja- e and Estimator's Price-book," and " by *Fred. T. Hodgson*; "Foundations a *ilding*," by E. Dobson, Weale's Series, wrought or rolled iron, 40,000; rolled steel, 48,000; white pine and s 3500; pitch or Georgia pine, 5000; American oak, 6000. The breaking st of wooden beams and girders shall be computed according to the fo in which the constants for transverse strains for central load shal follows, namely: Hemlock, 400; white pine, 450; spruce, 450; pitch or G pine, 550; American oak, 550; and for wooden beams and girders carr uniformly distributed load the constants will be doubled. The fac safety shall be as one to four for all beams, girders, and other pieces to a transverse strain; as one to four for all posts, columns, an vertical supports when of wrought iron or rolled steel; as one to other materials, subject to a compressive strain; as one to six rods, tie-beams, and other pieces subject to a tensile strain. Goo natural earth shall be deemed to safely sustain a load of four ton superficial foot, or as otherwise determined by the superintendent ings, and the width of footing-courses shall be at least sufficient to m requirement. In computing the width of walls, a cubic foot of br shall be deemed to weigh 115 lbs. Sandstone, white marble, gra other kinds of building-stone shall deemed to weigh 160 lbs. per cu The safe-bearing load to apply to good brickwork shall be taken per superficial foot when good lime mortar is used, 11½ tons per su foot when good lime and cement mortar mixed is used, and 15 tons erficial foot when good cement mortar is used.

Fire-proof Buildings—Iron and Steel Columns.— iron, wrought-iron, or rolled-steel columns shall be made true an at both ends, and shall rest on iron or steel bed-plates, and hav steel cap-plates, which shall also be made true. All iron or steel beams, headers, and tail-beams shall be suitably framed and conn gether, and the iron girders, columns, beams, trusses, and all other of all floors and roofs shall be strapped, bolted, anchored, and con gether, and to the walls, in a strong and substantial manner. Whe are framed into headers, the angle-irons, which are bolted to the ta shall have at least two bolts for all beams over 7 inches in depth, bolts for all beams 12 inches and over in depth, and these bolts sh less than ¾ inch in diameter. Each one of such angles or knees, wh to girders, shall have the same number of bolts as stated for the The angle-iron in no case shall be less in thickness than the heade mer to which it is bolted, and the width of angle in no case shall be one third the depth of beam, excepting that no angle-knee shall be 2½ inches wide, nor required to be more than 6 inches wide. All iron or rolled-steel beams 8 inches deep and under shall have beari to their depth, if resting on a wall; 9 to 12 inch beams shall have of 10 inches, and all beams more than 12 inches in depth shall have of not less than 12 inches if resting on a wall. Where beams re supports, and are properly tied to the same, no greater bearings sh quired than one third of the depth of the beams. Iron or steel fl shall be so arranged as to spacing and length of beams that the supported by them, together with the weights of the materials u construction of the said floors, shall not cause a deflection of the s of more than 1/30 of an inch per linear foot of span; and they sh together at intervals of not more than eight times the depth of th

Under the ends of all iron or steel beams, where they rest on th stone or cast-iron template shall be built into the walls. Said tem be 8 inches wide in 12-inch walls, and in all walls of greater thic template shall be 12 inches wide; and such templates, if of stone, n in any case less than 2½ inches in thickness, and no template sh than 12 inches long.

No cast-iron post or column shall be used in any building of a l thickness of shaft than three quarters of an inch, nor shall it h supported length of more than twenty times its least lateral dim diameter. No wrought-iron or rolled-steel column shall have an un length of more than thirty times its least lateral dimension or dia shall its metal be less than one fourth of an inch in thickness.

Lintels, Bearings and Supports.—All iron or steel li *have bearings proportionate to the weight to be imposed there lintel used to span any opening more than 10 feet in width shall h ing less than 12 inches at each end, if resting on a wall; but if re ron post, such lintel shall have a bearing of at least 6 inches* ... *of the wall to be supported*

... Girders and Rivets.—Rolled iron or ste

girders used as lintels or as girders,
hall be so proportioned that the loads
not produce strains in tension or com-
than 12,000 lbs. for iron, nor more than
of the gross section of each of such
he web-plate of more than 6000 lbs. per
-plate, if of iron, nor more than 7000
plate shall be less than ¼ inch in
hall not be less than ⅝ inch in diameter,
inches apart in any case. They shall be
ns shall not exceed 9000 lbs. per square
by the thickness of the plates through
girders shall be proportioned upon the
ord strains are resisted entirely by the
he shearing strains are resisted entirely
eb shall be estimated as flange area, nor
of the angle-iron which lies against the
tres of gravity of the flange areas will
of the girder.
New York contain a great amount of de-
e, and penalties are provided for viola-
rtment of Buildings, etc., Chapter 275,
lished by Baker, Voorhies & Co., New

OAD ON FLOORS.

Maximum load per square foot of floor
ense crowd. Considerable variation is
any authorities, as the following table

	Weight of Crowd, lbs. per sq. ft.
ne and Stoney	41
0)	70
wine........	84
ay bridges according to	
... .	100
Palace,....................	120
, at Melbourne	126 / 143.1
On Stresses," p. 617)	147.4

by crowding a number of persons pre-
the men being tightly packed so as to
occurs on the stairways and platforms
.

OF FLOORS.

anufacturers' Mutual Insurance Co.)
d by C. J. H. Woodbury, for determining
observed to select the figure giving the
entration of load as the one which may
-beams; and in no case should beams be
ose specified, unless a lower factor of
of a competent engineer.
ms or heavy timbers are made use of in
ehouse, they should not be painted, var-
pervious concrete, air-proof plastering,
t fermentation should destroy them by

floor-beams in two parts, with a small
ventilation may be secured, even if the
ted or filled.
oads, but the first can be used in respect
ted loads by using half the figure given
twice as much load when evenly distrib-
he load was concentrated in the centre

deducted from the figure given in
d which may be placed upon any
at 36 lbs. per cubic foot, and t

Southern-pine beams. From this should be deducted the w
which would amount to 17½ lbs. per square foot, leaving 1
foot as a safe load to be carried upon such a floor. If t
spruce, the result of 147½ lbs. would be multiplied by 0.78,
to 115 lbs. The weight of the floor, in this instance amou
would leave the safe net load as 90 lbs. per square foot for

Table II applies to the design of floors whose strength
of that necessary to sustain the weight, in order to meet
delicate or rapidly moving machinery, to the end that the
tortion of the floor may be reduced to the least practicable

In the table the limit is that of load which would cause
beams to a curve of which the average radius would be 12

This table is based upon a modulus of elasticity obtain
tions upon the deflection of loaded storehouse floors, and is
lbs. for Southern pine; the same table can be applied t
modulus of elasticity is taken as 1,200,000 lbs., if six tenth
Southern pine is taken as the proper load for spruce; or,
designing, the load should be increased one and two third
dimension of timbers for this increased load as found in th
used for spruce.

It can also be applied to beams and floor-timbers which
each end and in the middle, remembering that the defle
supported in that manner is only four tenths that of a bea
which rests at each end; that is to say, the floor-planks
half times as stiff, cut two bays in length, as they would b
bay in length. When a floor-plank two bays in length
three sixteenths of the load on the plank is sustained by t
end of the plank, and ten sixteenths by the beam under t
plank; so that for a completed floor three eighths of the lo
tained by the beams under the joints of the plank, and five e
by the beams under the middle of the plank: this is the rea
tance of breaking joints in a floor-plank every three feet i
beam shall receive an identical load. If it were not so, thr
whole load upon the floor would be sustained by every othe
eighths of the load by the corresponding alternate beams.

Repeating the former example for the load on a mill fl
pine beams 10 × 14 inches, and 20 feet span, laid 8 feet on
II a 1 × 14 inch beam should receive 61 lbs. per foot of sp
sq. ft. of floor, for Southern-pine beams. Deducting the w
17½ lbs. per sq. ft., leaves 57 lbs. per sq. ft. as the advisab

If the beams are of spruce, the result of 75 lbs. should be

.oads upon Southern-pine Beams
Inch in Width.

. J. H. Woodbury.)

at the centre of the span, the beams will sus in the table.)

epth of Beam in inches.

7	8	9	10	11	12	13	14	15	16
in pounds per foot of Span.									
470	614	778	960						
327	427	540	667	807					
240	314	397	490	593	705	828			
184	240	304	375	454	540	634	735		
145	190	240	296	359	427	501	581	667	759
118	154	194	240	290	346	406	470	540	614
97	127	161	198	240	286	335	389	446	508
82	107	135	167	202	240	282	327	375	474
70	90	115	142	172	205	240	278	320	364
60	78	99	123	148	176	207	240	276	314
52	68	86	107	129	154	180	209	240	273
46	60	76	94	113	135	158	184	211	240
41	53	67	83	101	120	140	163	187	217
36	47	60	74	90	107	125	145	167	190
..	43	54	66	80	96	112	130	150	170
..	38	49	60	73	86	101	118	135	154
.	....	44	54	66	78	92	107	122	139
..	...	...	50	60	71	84	97	112	127
..	..	....	45	55	65	77	89	102	116
..	..	....	....	50	60	70	82	94	107
...	...	..	..	46	55	65	75	86	98

upon Southern-pine Beams suffi-
Standard Limit of Deflection.

. J. H. Woodbury.)

h of Beam in inches.

8	9	10	11	12	13	14	15	16	Deflection, inches.
pounds per foot of Span.									
182	259								.0300
126	180	247							.0432
93	132	181	241						.0588
71	101	139	185	240	305				.0768
56	80	110	146	190	241	301			.0972
46	65	89	118	154	195	244	300		.1200
38	54	73	98	127	161	202	248	301	.1452
32	45	62	82	107	136	169	208	253	.1728
27	38	53	70	91	116	144	178	215	.2028
23	33	45	60	78	100	124	153	186	.2352
20	29	40	53	68	87	108	133	162	.2700
18	25	35	46	60	76	95	117	147	.3072
16	22	31	41	53	68	84	104	126	.3468
...	20	27	37	47	60	75	93	112	.3888
...	18	25	33	43	54	68	83	101	.4332
..	...	22	30	38	49	61	75	91	.4800
...	...	20	27	35	44	55	68	83	.5[illegible]
...	....	...	24	32	40	50	62	75	
...	...	....	22	29	37	46	57	69	
...	...	....	....	27	34	42	52	63	
..	...	...		25	31	39	48	58	

Unit of force = 1 dyne = $\frac{1}{981}$ gramme = $\frac{\ldots}{981}$ lb. =

A dyne is that force which, acting on a mass of one gra
second, will give it a velocity of one centimetre per second
one gramme in latitude 40° to 45° is about 980 dynes, at the e
and at the poles nearly 984 dynes. Taking the value of g,
due to gravity, in British measures at 32.185 feet per second
metre = 39.37 inches, we have

$$1 \text{ gramme} = 32.185 \times 12 \div .3937 = 981.00 \text{ dyne}$$

Unit of work = 1 erg = 1 dyne-centimetre = .000000073
Unit of power = 1 watt = 10 million ergs per second,
= .7378 foot-pound per second,

$$= \frac{.7378}{550} = \frac{1}{746} \text{ of 1 horse-power} =$$

C.G.S. Unit of magnetism = the quantity which attrac
equal quantity at a centimetre's distance with the force of

C.G.S. Unit of electrical current = the current which, flo
length of 1 centimetre of wire, acts with a force of 1 dyn
magnetism distant 1 centimetre from every point of the wi
the commercial unit of current, is one tenth of the C.G.S. u

The Practical Units used in Electrical Cal

Ampere, the unit of current strength, or rate of flow, repr

Volt, the unit of electro-motive force, electrical pressure,
potential, represented by E.

Ohm, the unit of resistance, represented by R.

Coulomb (or ampere-second), the unit of quantity, Q.

Ampere-hour = 3600 coulombs, Q'.

Watt (ampere-volt, or volt-ampere), the unit of power, P.

Joule (volt-coulomb), the unit of energy or work, W.

Farad, the unit of capacity, represented by K.

Henry, the unit of induction, represented by L.

Using letters to represent the units, the relations betwe
expressed by the following formulæ, in which t represent
T one hour:

$$C = \frac{E}{R}, \quad Q = Ct, \quad Q' = CT, \quad K = \frac{Q}{E}, \quad W = QE,$$

As these relations contain no coefficient other than unity

as adopted at the International Electrical
d as established by Act of Congress of the
as follows:
1 to 10^9 (or 1,000,000,000) units of resistance
esented by the resistance offered to an un-
umn of mercury at 32° F., 14.4521 grammes
tional area, and of the length of 106.3 centi-

it of current of the C.G.S. system, and is the
nvarying current which when passed through
in water in accordance with standard speci-
rate of .001118 gramme per second.
ive force that, steadily applied to a conductor
will produce a current of one ampere, and is
/1434 (or .6974) of the electro-motive force be-
s of a Clark's cell at a temperature of 15° C.,
escribed in the standard specifications.
of electricity transferred by a current of one

of a condenser charged to a potential of one
icity.
),000 units of work in the C.G.S. system, and is
energy expended in one second by an ampere

00 units of power in the C.G.S. system, and is
work done at the rate of one joule per second.
in a circuit when the electro-motive force in-
lt, while the inducing current varies at the rate

e defined, are called the "international" ohm,
from the "legal" ohm, B.A. unit, etc.
mined by a committee of the British Associa-
unit, was the resistance of a certain piece of
ndon. The so-called "legal" ohm, as adopted
of Electricians in Paris in 1884, was a correc-
vas defined as the resistance of a column of
in section and 106 centimetres long, at a tem-

2 B.A. units, 1 B.A. unit = 0.9889 legal ohm;
6 " " 1 " " = 0.9866 int. ohm;
3 legal ohm, 1 legal ohm = 0.9977 " "

DERIVED UNITS.

= 1 million ohms;
= 1 millionth of an ohm;
ere = 1/1000 of an ampere;
ad = 1 millionth of a farad.

ONS OF VARIOUS UNITS.

= 1 coulomb per second;
= 1 watt = 1 volt-coulomb per second;
= .7373 foot-pound per second,
= .0009477 heat-units per second (Fahr.),
= 1/746 of one horse-power;
= .7373 foot-pound,
= work done by one watt in one second,
= .0009477 heat-unit;
= 1055.2 joules;
= 737.3 foot-pound per second,
= .9477 heat-units per second,
= 1000/746 or 1.3405 horse-powers;
= 1.3405 horse-power hours,
= 2,654,200 foot-pounds,
= 3416 heat-units;
= 746 watts = 746 volt-amperes,
= 33,000 foot-pounds per minute.

are defined in terms of one another as follows:
uctor through which a current of one amper
tive force is one volt. Ampere, the quantit

Equivalent Values of Electrical and Mechanical Units.

Unit.	Equivalent Value in Other Units.
1 K. W. Hour =	1,000 watt hours. 1.34 horse-power hours. 2,654,200 ft.-lbs. 3,600,000 joules. 3,412 heat-units. 367,000 kilogram metres. .235 lb. carbon oxidized with perfect efficiency. 3.53 lbs. water evap. from and at 212° F. 22.75 lbs. of water raised from 62° to 212° F.
1 H.P. Hour =	.746 K. W. hours. 1,980,000 ft.-lbs. 2,545 heat-units. 273,740 k. g. m. .175 lb. carbon oxidized with perfect efficiency. 2.64 lbs. water evaporated from and at 212° F. 17.0 lbs. water raised from 62° F. to 212° F.
1 Kilo-watt =	1,000 watts. 1.34 horse-power. 2,654,200 ft.-lbs. per hour. 44,240 ft.-lbs. per minute. 737.3 ft.-lbs. per second. 3,412 heat-units per hour. 56.9 heat-units per minute. .948 heat-unit per second. [illegible]

Unit.	Equivalent Value in Other Units.
1 H.P. =	746 watts. .746 K. W. 33,000 ft.-lbs. per minute. 550 ft.-lbs. per second. 2,545 heat-units per hour. 42.4 heat-units per minute. .707 heat-units per second. .175 lbs. carbon oxidized per hour. 2.64 lbs. water evap. per hour from and at 212° F.
1 Joule =	1 watt second. .000000278 K. W. hour. .102 k. g. m. .0009477 heat-units. .7373 ft.-lb.
1 Ft.-lb. =	1.356 joules. .1383 k. g. m. .000000377 K. W. hours. .001285 heat-units. .0000005 H.P. hour.
1 Watt =	1 joule per second. .00134 H.P. 3.412 heat-units per hour. .7373 ft.-lbs. per second. .0035 lbs. water evap. per hr. 44.24 ft.-lbs. per minute.
1 Watt [illegible]	8.19 heat-units per sq. ft. per minute.

Unit.	Equivalent Value in Other Units.
1 Heat-unit =	1,055 watt seconds. 778 ft.-lbs. 107.6 kilogram metres. .000293 K. W. hour. .000393 H.P. hour. .0000688 lbs. carbon oxidized. .001036 lbs. water evap. from and at 212° F.
1 Heat-unit per Sq. Ft. per min. =	.122 watts per square in. .0176 K. W. per sq. ft. .0236 H.P. per sq. ft.
1 Kilogram Metre =	7.233 ft.-lbs. .00000365 H.P. hour. .00000272 K. W. hour. .0093 heat-units.
1 lb. Carbon Oxidized with perfect Efficiency =	14,544 heat-units. 1.11 lb. Anth'cite coal ox. 2.5 lbs. dry wood oxidized. 21 cu. ft. illuminating-gas. 4.26 K. W. hours. 5.71 H.P. hours. 11,315,000 ft.-lbs. 15 lbs. of water evap. from and at 212° F.
1 lb. Water Evapor'ed from and at	.283 K. W. hour. .379 H.P. hour. 965.7 heat-units. 103,900 k. g. m. 1,019,000 joules. [illegible]

igh a resistance of one ohm when the electro-
the electro-motive force required to cause a
hrough a resistance of one ohm.

Circuit.—(See Electro-magnets, page 1058.)

ig Electrical Measurements, Test-
imieson's Pocket-Book of Electrical Rules,
son's Dynamo-Electric Machinery; and works

and Mechanical Units.—H. Ward
trical Engineer. Feb. 25, 1895, a table of use-
l mechanical units, from which the table on
nodifications.

N THE FLOW OF WATER AND ECTRICITY.

	ELECTRICITY.
t. in., in	Volts; electro-motive force; difference of potential or of pressure; E. or E.M.F.
, etc., , with ; de- tional id de- nplex r.	Ohms, resistance, R. The resistance increases directly as the length of the conductor or wire and inversely as its sectional area, $R \propto l \div s$. It varies with the nature or quality of the conductor. Conductivity is the reciprocal of specific resistance.
cond. olume iining d in	Amperes; current; current strength; intensity of current; rate of flow; 1 ampere = 1 coulomb per second. $\text{Amperes} = \frac{\text{volts}}{\text{ohms}}$; $C = \frac{E}{R}$; $E = CR$.
cubic quiva- ie, as many	Coulomb, unit of quantity, Q, = rate of flow × time, as ampere-seconds. 1 ampere-hour = 3600 coulombs.
foot- fall- ll; in ity in n lbs. h the	Joule, volt-coulomb, W, the unit of work, = product of quantity by the electro-motive force = volt-ampere-second. 1 joule = .7373 foot-pound. If C (amperes) = rate of flow, and E (volts) = difference of pressure between two points in a circuit, energy expended = CEt, = C^2Rt, since $E = CR$.
er,ft.- 33,000. ng in owing e feet ig the	Watt, unit of power, P, = volts × amperes, = current or rate of flow × difference of potential. 1 watt = .7373 foot-pound per second = 1/746 of a horse-power.

Ampere and the Miner's Inch.
ier's inch is defined as the quantity of water
rture an inch square in a board two inches
six inches. Here, as in the case of the am-
any abstract quantity, such as gallons or
to time. It is simply a rate of flow. We
r, six inches, as the representative of electri-
ie aperture restricting the flow of water may
sistance of one ohm; the flow through a re-
pressure of one volt is one amp... w
-inch hole two inches long und...
of the opening is one miner'...
e correct analogue of the am...
of water, 0.194 gallon; the...
ib.

ELECTRICAL RESISTANCE.

Laws of Electrical Resistance.—The resistance, R, of any conductor varies directly as its length, l, and inversely as its sectional area, s, or $R \propto \frac{l}{s}$.

EXAMPLE.—If one foot of copper wire .01 in. diameter has a resistance of .10323 ohm, what will be the resistance of a mile of wire .3 in. diam. at the same temperature? The sectional areas being proportional to the squares of the diameters, the ratio of the areas is $.3^2 : .01^2 = 900$ to 1. The lengths are as 5280 to 1. The resistances being directly as the lengths and inversely as the sectional areas, the resistance of the second wire is $.10323 \times 5280 \div 900 = .6056$ ohm.

Conductance, c, is the inverse of resistance. $R = \frac{l}{sc}$, $c = \frac{l}{sR}$. If c and c_1 represent the conductances, and R and R_1 the respective resistance of two substances of the same length and section, then $c : c_1 :: R_1 : R$.

Equivalent Conductors.—With two conductors of length l, l_1, of conductances c, c_1, and sectional areas s, s_1, we have the same resistance, and one may be substituted for the other when $\frac{l}{cs} = \frac{l_1}{c_1 s_1}$.

The specific resistance, also called resistivity, a, of a material of unit length and section is its resistance as compared with the resistance of a standard conductor, such as pure copper. Conductivity, or specific conductance, is the reciprocal of resistivity.

$$R = \frac{l}{sc}, \quad R = \frac{al}{s}.$$

If two wires have lengths l, l_1, areas s, s_1, and specific resistances a, a_1, their actual resistances are $R = \frac{al}{s}$, $R_1 = \frac{a_1 l_1}{s_1}$, and $\frac{R}{R_1} = \frac{a l s_1}{a_1 l_1 s}$.

Electrical Conductivity of Different Metals and Alloys.—Lazare Weiller presented to the Société Internationale des Électriciens the results of his experiments upon the relative electrical conductivity of certain metals and alloys, as here appended:

1. Pure silver	100	17. Phosphor tin	17.7
2. Pure copper	100	18. Alloy of gold and silver (50%)	16.12
3. Refined and crystallized copper	99.9	19. Swedish iron	16
4. Telegraphic silicious bronze	98	20. Pure Banca tin	15.45
5. Alloy of copper and silver (50%)	86.65	21. Antimonial copper	12.7
6. Pure gold	78	22. Aluminium bronze (10%)	12.6
7. Silicide of copper, 4% Si	75	23. Siemens steel	12
8. Silicide of copper, 12% Si	54.7	24. Pure platinum	10.6
9. Pure aluminium	54.2	25. Copper with 10% of nickel	10.6
10. Tin with 12% of sodium	46.9	26. Cadmium amalgam (15%)	10.2
11. Telephonic silicious bronze	35	27. Dronier mercurial bronze	10.14
12. Copper with 10% of lead	30	28. Arsenical copper (10%)	9.1
13. Pure zinc	29.9	29. Pure lead	8.88
14. Telephonic phosphor-bronze	29	30. Bronze with 20% of tin	8.4
15. [illegible] 25% zinc	26.49	31. Pure nickel	7.89
16. [illegible] zinc	21.5	32. Phosphor-bronze, 10% tin	6.5
		33. Phosphor-copper, 9% phos.	4.9
		34. Antimony	3.88

[illegible] above resistances may be reduced to ohms on the basis [illegible] copper one millimetre in diameter at a temperature of [illegible] of .02057 international ohms per metre; or a wire .001 [illegible] resistance of 9.59 international ohms per foot.

es of Different Metals at 0° and C. (Matthiessen.)

vities.	Metals.	Conductivities.	
At 100° C. " 212° F.		At 0° C. " 32° F.	At 100° C. " 212° F.
71.56	Tin	12.36	8.67
70.27	Lead	8.32	5.86
55.90	Arsenic..........	4.76	3.33
20.67	Antimony......	4.62	3.26
16.77	Mercury, pure..	1.60	....
....	Bismuth	1.245	0.878
....			

nsulators in Order of their Value.

	Insulators (Non-conductors).	
	Dry Air	Ebonite
	Shellac	Gutta-percha
	Paraffin	India-rubber
	Amber	Silk
	Resins	Dry Paper
	Sulphur	Parchment
	Wax	Dry Leather
ns	Jet	Porcelain
	Glass	Oils
	Mica	

resistance of distilled water is 6754 million times

with Temperature.—For every degree Cen-pper increases about 0.4%, or for every degree F. pper wire having a resistance of 10 ohms at 32° 11.11 ohms at 82° F.
s the amount of resistance of a few substances purposes by which 1 ohm is increased by a rise C.

	Rise of R. of 1 Ohm when Heated—	
	1° F.	1° C.
....	.00013	.00021
....	.00018	.00031
below)	.00024	.00044
....	.00036	.00065
. ...	.00044	.00080
....	.00222	.00400

ree of hardness or softness of a metal or alloy istance is lessened by annealing. Matthiessen conductivities for copper and silver, the com-ure silver at 100° C.:

	Temp. C.	Hard.	Annealed.
....	11°	95.31	97.83
....	14.6°	95.36	103.33

he conductivities of copper, silver, and brass with h the following results:

	Hard.	Annealed.
....	52.207	55.253
....	56.252	64.380
....	11.439	13.502

Electrical Congress 1893, p. 179) says t lepends on its composition. Matthies *per, with a temperature coefficient* c *r, has found* copper-nickel-zinc al silver) which had a resistance of nearly 28 times that of copper, and a perature coefficient of about one half that given by Matthiessen. Ke and Fessenden (Proc. Elec. Cong., p. 186) find that copper has a u temperature coefficient of 0.406% per degree C., between the limits of 250° C.

Standard of Resistance of Copper Wire. (Trans. A. I. Sept. and Nov. 1890.)—Matthiessen's standard is: A hard-drawn copp 1 metre long, weighing 1 gramme has a resistance of 0.1469 B.A. 0° C. (1 B.A. unit = 0.9889 legal ohm = 0.9866 international ohm.) ance of hard copper = 1.0226 times that of soft copper. Relative con power (Matthiessen): silver, 100; hard or unannealed copper, 99.95; annealed copper, 102.21. Conductivity of copper at other temperatur 0° C.,

$$Ct = C_0(1 - .00387t + .000009009t^2).$$

The resistance is the reciprocal of the conductivity, and is

$$Rt = R_0(1 + .00387t + .00000597t^2).$$

A committee of the Am. Inst. Electrical Engineers recommend the ing as the most correct form of the Matthiessen standard, taking 8. sp. gr. of pure copper:

A soft copper wire 1 metre long and 1 mm. diam. has an electrica ance of .02057 B.A. unit at 0° C. From this the resistance of a soft wire 1 foot long and .001 in. diam. (mil-foot) is found to be 9.720 B at 0° C.

Standard Resistance at 0° C.	B.A. Units.	Legal Ohms.	I
Metre-millimetre, soft copper.........	.02057	.02034	.
Cubic centimetre " "	.000001616	.000001598	.
Mil-foot " "	9.720	9.612	9.
1 mil-foot, of soft copper at 10°.22 C. or 50°.4 F...	10		9.
" " " " " " 15°.5 " 59°.9 F...	10.20		10
" " " " " " 23°.9 " 75° F...	10.53		10

For tables of the resistance of copper wire, see pages 218 to pp. 1034, 1035.

Taking Matthiessen's standard of pure copper as 100%, some refi has exhibited an electrical conductivity equivalent to 103%.

Matthiessen found that impurities in copper sufficient to de density from 8.94 to 8.90 produced a marked increase of electrical r

ELECTRIC CURRENTS.

Ohm's Law.—This law expresses the relation between the t damental units of resistance, electrical pressure, and current. It i

$$\text{Current} = \frac{\text{electrical pressure}}{\text{resistance}}; \quad C = \frac{E}{R}; \quad \text{whence} \quad E = CR, \text{ and}$$

In terms of the units of the three quantities,

$$\text{Amperes} = \frac{\text{volts}}{\text{ohms}}; \quad \text{volts} = \text{amperes} \times \text{ohms}; \quad \text{ohms} = \frac{\text{vo}}{\text{amp}}$$

EXAMPLES: *Simple Circuits.*—1. If the source has an effective pressure of 100 volts, and the resistance is two ohms, what is the c

$$C = \frac{E}{R} = \frac{100}{2} = 50 \text{ amperes.}$$

2. What pressure will give a current of 50 amperes through a res 2 ohms? $E = CR = 50 \times 2 = 100$ volts.

3. What resistance is required to obtain a current of 50 amperes pressure is 100 volts? $R = \frac{E}{C} = \frac{100}{50} = 2$ ohms.

The following examples are from R. E. Day's "Electric Light Ari

1. The internal resistance of a certain Brush dynamo-machine is 1 and the external resistance is 73 ohms; the electro-motive force o chine being 839 volts. Find the strength of the current flowing in t

$$E = 839; \quad R = 73 + 10.9 = 83.9 \text{ ohms};$$
$$C = E \div R = 839 \div 83.9 = 10 \text{ amperes.}$$

a resistance of 9.36 ohms, while the re-
ohm, and that of the dynamo is 2.8 ohms.
ve force of the machine when the strength
peres.

13.26 ohms; $C = 14.8$ amperes;
$.8 = 196.3$ volts.

data the average resistance of each of
The electro-motive force of the machine
ohms, while that of the leading wires is 2
through each lamp is 21 amperes.
nce in ohms of each lamp, then the total
$+ 2 + 3.7$.
$. 3x + 5.7 = 244/21 = 11.61$ ohms, whence

mps were placed in series. The average
was 39.3 ohms, and that of the dynamo
at electro-motive force was required to
through this circuit?

$+ 11.2 = 129.1$ ohms, and
re;

$1.2 \times 129.1 = 154.9$ volts.

certain Brush lamp was 3.8 ohms when a
through it. What was the electro-motive

$10 \times 3.8 = 38$ volts.

alvanic cells, each of which had an aver-
were joined up in series to one incandes-
nd produced a current of 0.112 amperes.
rent produced by a series of 30 such cells
resistance?
he problem enable us to determine the
ch cell of the battery. Let this be repre-

$(25 \times 15 + 70) = .112 \times 445;$

$\frac{445}{} = 2$ volts, nearly.

l part of the problem, we have, by Ohm's

$\overline{30} = \frac{60}{510} = 0.118$ ampere.

rcuit has two paths, the total current in
resistances.
f the two branches, and C and C_1 the cur-
$\frac{R_1}{R}$, whence

$; \; R = \frac{C_1 R_1}{C}; \; R_1 = \frac{CR}{C_1}.$

, one circuit is said to be in *shunt* to the
e arc or in parallel.
conductors are arranged on
nd the total resistance is the
R_2.
simple circuit we have
e internal parts of the s

small as is commercially practicable, so
ay be wasted in heating the wire. The
e fuel burned in the boiler to the electric

o mechanical energy by means of the boiler

med into electrical energy in the dynamo.
ed into heat in the electric light.
uctor is the equivalent of the energy causing
diture of energy in watts = electro-motive
eres = EC, and the energy in joules = watts
t = $C^2Rt = ECt$.

s. (From Kapp's Electrical Transmission
ter of great importance to determine before-
is to be expected in each given case, and if
greater than appears safe, provision must be
hich heat is carried off. This can generally
erficial area of the conductor. Say we have
uare inch area, and find that with 1000 amperes
hot. Now by splitting up this conductor into
enth of a square inch cross-sectional area, we
ount of energy transformed into heat, but we
xposed to the cooling action of the surrounding
therefore the ten thin wires can dissipate more
compared with the single thick wire.

**Subaqueous and Aerial Cables (In-
Gutta-percha).** (Prof. Forbes.)

ble ÷ Diameter of conductor = 4.
ure of air = 20° C. = 68° F.
mperature of conductor over air.

Current in amperes.

$t = 9°$ C. $= 16.2°$ F.	$t = 25°$ C. $= 45°$ F.	$t = 49°$ C. $= 92.2°$ F.	$t = 81°$ C. $= 145.8°$ F.
11.0	17.8	24.0	29.5
27.0	43.8	59.0	72.5
44.4	72.1	97.3	119
62.5	102	137	168
81.0	131	177	218
100	164	219	268
119	192	259	319
137	223	301	369
157	253	342	420
175	285	384	472
367	595	803	988
559	908	1225	1503
753	1221	1646	2021
945	1534	2068	2523
1138	1846	2491	3058
1330	2158	2846	3575
1525	2472	3335	4094
1716	2785	3755	4611
1909	3097	4178	5130

insulated wire carries a greater current without
e if the diameter be not too great. Assuming
be twice the diam. of the conductor, a greater
ulated wires than in bare wires up to 1.9 inch
. of cable = 4 times diam. of conduc
of conductor.

–The table on pages 1034 and 103
ommittee on Units and Standards
gineers (Trans. Oct. 1893).

Weights, Lengths, and Resistances of Cool, Warm, and Hot Copper Wire.

B. & S. Gauge	B. W. G. (Stubbs')	Diameter, inches.	Area, Circular mils.	Weight: Lbs. per Foot.	Weight: Lbs. per Ohm at 20° C., 68° F.	Length: Feet per Lb.	Length: Ft. per Ohm at 20° C., 68° F.	Resistance [illegible]	Resistance [illegible]	Resistance [illegible]	Resistance [illegible]
		0.460	211,600	0.6405	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	0000	0.454	206,100	0.6239	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	000	0.425	180,600	0.5468	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
000		0.4096	167,800	0.5080	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	00	0.380	144,400	0.4371	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
00		0.3648	133,100	0.4028	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	0	0.340	115,600	0.3499	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
0		0.3249	105,500	0.3195	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	1	0.3000	90,000	0.2724	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
1		0.2893	83,690	0.2533	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	2	0.2840	80,660	0.2441	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	3	0.2590	67,080	0.2031	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
2		0.2576	66,370	0.2009	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	4	0.2380	56,640	0.1715	[illegible]	5.833	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
3		0.2294	52,620	0.1593	[illegible]	6.278	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	5	0.2200	48,400	0.1465	[illegible]	6.826	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
4		0.2043	41,740	0.1264	[illegible]	7.914	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	6	0.2030	41,210	0.1247	[illegible]	8.017	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
5		0.1819	33,100	0.1002	[illegible]	9.980	3,197	[illegible]	[illegible]	[illegible]	[illegible]
	7	0.1800	32,400	0.09808	[illegible]	10.20	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	8	0.1650	27,230	0.08241	[illegible]	12.13	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
6		0.1620	26,250	0.07946	[illegible]	12.58	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	9	0.1480	21,900	0.06630	140.3	15.08	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
7		0.1443	20,820	0.06302	126.7	15.87	2,011	[illegible]	[illegible]	[illegible]	[illegible]
	10	0.1340	17,960	0.05435	94.26	18.40	[illegible]	0.01061	[illegible]	[illegible]	[illegible]
8		0.1285	16,510	0.04998	79.69	20.01	[illegible]	0.01255	[illegible]	[illegible]	[illegible]
	11	0.1200	14,400	0.04359	60.69	22.94	[illegible]	0.01648	[illegible]	[illegible]	[illegible]
9		0.1144	13,090	0.03963	50.12	25.23	[illegible]	0.01995	[illegible]	[illegible]	[illegible]
	12	0.1090	11,880	0.03596	41.27	27.81	1,147	0.02423	[illegible]	[illegible]	[illegible]
10		0.1019	10,380	0.03143	31.52	31.82	1,003	0.03173	[illegible]	[illegible]	[illegible]
	13	0.0950	9,025	0.02732	23.81	36.60	871.7	0.04199	0.001147	[illegible]	[illegible]
11		0.09074	8,234	0.02493	19.82	40.12	795.3	0.05045	0.001257	[illegible]	[illegible]
	14	0.08300	6,889	0.02085	13.87	47.95	665.4	0.07207	0.001503	[illegible]	[illegible]
12		0.08081	6,530	0.01977	12.47	50.59	630.7	0.08019	0.001586	[illegible]	[illegible]
	15	0.07200	5,184	0.01569	7.857	63.73	500.7	0.1273	0.001997	[illegible]	[illegible]
13		0.07196	5,178	0.01568	7.840	63.79	500.1	0.1276	0.001999	[illegible]	[illegible]
	16	0.06500	4,225	0.01279	5.219	78.19	408.1	0.1916	0.002451	[illegible]	[illegible]
14		0.06408	4,107	0.01243	4.931	80.44	396.6	0.2028	0.002521	[illegible]	[illegible]
	17	0.0580	3,364	0.01018	3.308	98.23	324.9	0.3023	0.003078	[illegible]	[illegible]
15		0.05707	3,257	0.009858	3.101	101.4	314.5	0.3225	0.003179	[illegible]	[illegible]
16		0.05082	2,583	0.007818	1.950	127.9	249.4	0.5128	0.004009	[illegible]	[illegible]
	18	0.04900	2,401	0.007268	1.685	137.6	231.9	0.5932	0.004318	[illegible]	[illegible]

Gauges.		Diameter, inches.	Area, Circular mils.	Weight.		Length.		Resistance in International Ohms.			
A. W. G. B. & S.	B. W. G. Stubbs'.			Lbs. per Foot.	Lbs. per Ohm, at 20° C., 68° F.	Feet per Lb.	Ft. per Ohm, at 20° C., 68° F.	Ohms per Lb. at 20° C., 68° F.	Ohms per ft. at 20° C., 68° F.	Ohms per ft. at 50° C., 122° F.	Ohms per ft. at 80° C., 176° F.
17		0.04526	2,048	0.006200	1.226	161.3	197.8	0.8153	0.005055	0.005648	0.006259
	19	0.04200	1,764	0.005340	0.9097	187.3	170.4	1.099	0.005870	0.006558	0.007267
18		0.04030	1,624	0.004917	0.7713	203.4	156.9	1.296	0.006374	0.007122	0.007892
19		0.03589	1,288	0.003899	0.4851	256.5	124.4	2.061	0.008038	0.008980	0.009952
	20	0.03500	1,225	0.003708	0.4387	269.7	118.3	2.279	0.008452	0.009443	0.01047
	21	0.03200	1,024	0.003100	0.3066	322.6	98.90	3.262	0.01011	0.01130	0.01252
20		0.03196	1,022	0.003092	0.3051	323.4	98.66	3.278	0.01014	0.01132	0.01255
21		0.02846	810.1	0 002452	0.1919	407.8	78.24	5.212	0.01278	0.01428	0.01583
	22	0.02800	784.0	0.002373	0.1797	421.4	75.72	5.565	0.01321	0.01475	0.01635
22		0.02535	642.4	0.001945	0.1207	514.2	62.05	8.287	0.01612	0.01801	0.01996
	23	0.0250	625.0	0.001892	0.1142	528.6	60.36	8.756	0.01657	0.01851	0.02051
23		0.02257	509.5	0.001542	0.07589	648.4	49.21	13.18	0.02032	0.02271	0.02516
	24	0.0220	484.0	0 001465	0.06849	682.6	46.75	14.60	0.02139	0.02390	0.02649
24		0.02010	404.0	0.001223	0.04773	817.6	39.02	20.95	0.02563	0.02863	0.03173
	25	0.0200	400.0	0.001211	0.04678	825.9	38.63	21.38	0.02588	0.02892	0.03205
	26	0.0180	324.0	0.0009808	0.03069	1,020	31.29	32.58	0.03196	0.03570	0.03957
25		0.01790	320.4	0.0009699	0.03002	1,031	30.95	33.32	0.03231	0.03610	0.04001
	27	0.0160	256.0	0.0007749	0.01916	1,290	24.73	52.19	0.04045	0.04519	0.05008
26		0.01594	254.1	0.0007692	0.01888	1,300	24.54	52.97	0.04075	0.04552	0.05045
27		0.0142	201.5	0.0006100	0.01187	1,639	19.46	84.23	0.05138	0.05740	0.06362
	28	0.0140	196.0	0.0005933	0.01123	1,685	18.93	89.04	0.05283	0.05902	0.06541
	29	0.0130	169.0	0.0005116	0.008350	1,955	16.32	119.8	0.06127	0.06845	0.07586
28		0.01264	159.8	0.0004837	0.007466	2,067	15.43	133.9	0.06479	0.07239	0.08022
	30	0.0120	144.0	0.0004359	0.006062	2,294	13.91	165.0	0.07190	0.08033	0.08903
29		0.01126	126.7	0.0003836	0.004696	2,607	12.24	213.0	0.08170	0.09128	0.1012
30		0.01003	100.5	0.0003042	0.002953	3,287	9.707	338.6	0.1030	0.1151	0.1276
	31	0.0100	100.0	0.0003027	0.002924	3,304	9.658	342.0	0.1035	0.1157	0.1282
	32	0.0090	81.0	0.0002452	0.001918	4,078	7.823	521.3	0.1278	0.1428	0.1583
31		0.008928	79.70	0.0002413	0.001857	4,145	7.698	538.4	0.1299	0.1451	0.1608
	33	0.0080	64.0	0.0001937	0.001197	5,162	6.181	835.1	0.1618	0.1807	0.2003
32		0.007950	63.21	0.0001913	0.001168	5,227	6.105	856.2	0.1638	0.1830	0.2028
33		0.007080	50.13	0.0001517	0.0007346	6,591	4.841	1,361	0.2066	0.2308	0.2558
	34	0.0070	49.0	0.0001483	0.0007019	6,742	4.733	1,425	0.2113	0.2361	0.2616
34		0.006305	39.75	0.0001203	0.0004620	8,311	3 839	2,165	0.2605	9.2910	0.3225
35		0.005615	31.52	0.00009543	0.0002905	10,480	3.045	3,441	0.3284	0.3669	0.4067
36	35	0.0050	25.0	0.00007568	0.0001827	13,210	2.414	5,473	0.4142	0.4627	0.5129
37		0.004453	19.83	0.00006001	0.0001149	16,660	1.915	8,702	0.5222	0.5835	0.6466
	36	0.0040	16.	0.00004843	0.00007484	20,650	1.545	13,360	0.6471	0.7230	0.8011
		0.003965	15.72	0.00004759	0.00007210	21,010	1.519	13,870	0.6585	0.7357	0.8154
38		0.003531	12.47	0.00003774	0.00004545	26,500	1 204	22,000	0.8304	0.9277	1.028
39		0.003145	9.888	0.00002993	0.00002858	33,410	0.9550	34,980	1.047	1.170	1.296

The data from which the foregoing table has been computed are as follows: Matthiessen's standard resistivity, Matthiessen's temperature coefficients, specific gravity of copper = 8.89. Resistance in terms of the international ohm.

Matthiessen's standard 1 metre-gramme of hard-drawn copper = 0.1469 B. A. U. @ 0° C. Ratio of resistivity hard to soft copper 1.0226.

Matthiessen's standard 1 metre-gramme of soft-drawn copper = 0.1436 B. A. U. @ 0° C. One B. A. U. = 0.9866 international ohm.

Matthiessen's standard 1 metre-gramme of soft-drawn copper = 0.1417 international ohm @ 0° C.

Temperature coefficients of resistance for 20° C., 50° C., and 80° C., 1.07968, 1.20625, and 1.33681 respectively. 1 foot = 0.3048028 metre, 1 pound = 453.59256 grammes.

Heating of Coils.—To calculate the heating of a coil, given the radiating surface and its resistance. (Forbes.)

Let ρ = the resistance of a coil in ohms at the permissible temperature (the resistance (cold) must be increased by 1/5 of its value to give ρ);

S = the surface exposed to the air measured in square centimetres (1 square cm. = .155 square inch; 1 sq. in. = 6.45 square cm.);

t = the rise in temperature, centigrade scale;

C = the current in amperes.

$$.24C^2\rho = \text{heat generated} = etS.$$

where e is McFarlane's constant, varying from .0002 to .0003. The latter value may be taken. If 50° C. be the permissible rise in temperature,

$$C = \sqrt{\frac{.0003 \times 50 \times S}{.24 \times \rho}} = .25\sqrt{\frac{S}{\rho}}.$$

EXAMPLE.—The resistance of the field-magnets of a dynamo is 1.5 ohms cold, and the surface exposed to the air is 1 square metre; find the current to heat it not more than 50° C.

Here $S = 10{,}000$; $\rho = 1.8$ ohms; and $C = .25\sqrt{\frac{10{,}000}{1.8}} = 18.5$ amperes.

For the heating of coils of field-magnets Mr. C. Hering gives 1 watt of energy dissipated for every 223 square inches of cooling-surface for each degree F. of difference between the temperature of the coil and the surrounding air.

$W = CE = 1/223TS = 0.004476TS$, in which W = watts lost in coil, T = degrees Fahr., and S = square inches.

$C = \frac{TS}{223E}$ is the greatest current which can be used in the magnet coils of a shunt machine having a certain pressure in order that they do not heat above a certain temperature. Thus for a rise of temperature of 50° F. above the surrounding air,

$C = \frac{50S}{223E} = .224\frac{S}{E}$. Substituting for E its equivalent CR, we get

$$C = \sqrt{.224\frac{S}{R}}.$$

If 80° F. is the maximum difference of temperature,

$$C = \frac{80S}{223E} = .36\frac{S}{E} = .60\sqrt{\frac{S}{R}}.$$

The formula can be used for series machines when C is known, for which

$$C^2R = 1/223TS, \quad \text{we get } R = \frac{TS}{223C^2}.$$

With a permissible rise of 50° F. or 80° F., we have respectively,

$$R = \frac{.224S}{C^2}; \text{ and } R = .36\frac{S}{C^2}.$$

The surface area of the coil in square inches may be found from

$$S = \frac{223W}{T} = \frac{223CE}{T} = \frac{223C^2R}{T}.$$

50° F. or 80° F., respectively, the surface will

$.46W$; and $S = \frac{223W}{80} = 2.8W.$

H. Preece gives a formula for the current re-
ent metals, viz.: $C = ad^{\frac{3}{2}}$, in which d is the
efficient whose value for different metals is as
num 7585; platinum 5172; German silver 5230;
1642; lead, 1379; alloy of 2 lead and 1 tin, 1318.

Wires which will be Fused by a
iven Current.

42 for tin = 1379 for lead = 10244 for copper =

	Lead Wire.		Copper Wire.		Iron Wire.	
	Diam. inches.	Approx. S.W.G.	Diam. inches.	Approx. S.W.G.	Diam. inches.	Approx. S.W.G.
	.0081	35	.0021	47	.0047	40
	.0128	30	.0034	43	.0074	36
	.0168	27	.0044	41	.0097	33
	.0208	25	.0053	39	.0117	31
	.0236	23	.0062	38	.0136	29
	.0375	20	.0098	33	.0216	24
	.0491	18	.0129	30	.0283	22
	.0595	17	.0156	28	.0343	20.5
	.0690	15	.0181	26	.0398	19
	.0779	14	.0205	25	.0450	18.5
5	.0864	13.5	.0227	24	.0498	18
5	.0944	13	.0248	23	.0545	17
	.1021	12	.0268	22	.0589	16.5
5	.1095	11.5	.0288	22	.0632	16
	.1237	10	.0325	21	.0714	15
	.1371	9.5	.0360	20	.0791	14
5	.1499	8.5	.0394	19	.0864	13.5
	.1621	8	.0426	18.5	.0935	13
5	.1789	7	.0457	18	.1003	12
	.1964	6	.0516	17.5	.1133	11
	.2176	5	.0572	17	.1255	10
	.2379	4	.0625	16	.1372	9.5
	.2573	3	.0676	16	.1484	9
5	.2760	2	.0725	15	.1592	8
5	.3208	0	.0841	13.5	.1848	6.5
	.3617	00.5	.0950	12.5	.2086	5

es Required to Fuse Wires According
the Formula $C = ad^{\frac{3}{2}}$.

$d^{\frac{3}{2}}$.	Tin. $a = 1642.$	Lead $a = 1379.$	Copper $a = 10244$	Iron. $a = 3148.$
22627	37.15	31.20	231.8	71.22
16191	26.58	22.32	165.8	50.96
10516	17.27	14.50	107.7	33.10
06831	11.22	9.419	69.97	21.50
04685	7.692	6.461	48.00	14.75
03263	5.357	4.499	33.43	10.97
02415	3.965	3.330	24.74	[illegible]
01801	2.956	2.483	18.44	
01381	2.267	1.904	14.15	
01122	1.843	1.548	11.50	

ELECTRIC TRANSMISSION.

Cross-section of Wire Required for a Given Curren *Constant Current (Series) System.*—The cross-sectional area of co necessary in any circuit for a given constant current depends on the di ence between the pressure at the generating station and the maxim pressure required by all the apparatus on the circuit, and on the total le of the circuit. The following formulæ are given in "Practical Elect Engineering:"

If V = pressure in volts at generators;
v = sum of all the pressures (in volts) required by apparatus sup in the circuit;
n = total length (going and return) of circuit in miles;
C = current in amperes;
r = resistance of 1 mile of copper-conductor of 1 square inch secti area in ohms;
a = required cross-sectional area of copper in square inches,—

$$a = \frac{nrC}{V - v}.$$

If we take the temperature of the conductor when the current has flowing for some time through it, as 80° F.,

$$r = 0.0455 \text{ ohm, and } a = \frac{0.0455nC}{V - v}.$$

It generally happens, however, that we are not tied down to a part value of V, as the pressure at the generators can be varied by a few v suit requirements. In this case it is usual to fix upon a current densi determine the cross-sectional area of copper in accordance with it.

If D = current density in amperes per square inch determined upo

$$a = \frac{C}{D}.$$

The current density is frequently taken at 1000 amperes to the squa but should in general be determined by economical consideratio every case in question.

Allowable Current Density in Insulated Cables. — Experime insulated cables in casing gave the results shown below, but they ne firmation or correction of the current densities permissible in differe of insulated cables run underground. C and D are the current in a and the current density in amperes per square inch, respectively, wh raise the temperature of the conductor by the number of degree indicated by the suffix.

No. Strands.	S.W.G.* of each Wire.	Area of Strand in square inches.	C_{18}	D_{18}	C_{50}
7	20	0.0072	18	2,500	28
7	14	0.0357	59	1,400	95
19	14	0.0975	126	1,300	205
37	14	0.191	210	1,100	339

Constant Pressure (Parallel System).—To determine the pressure in a feeder of given size in the case of two-wire parallel distr

Let a = cross-sectional area of copper of one conductor of the fe square inches;
n = length of feeder (going and return) in miles;
C = current in amperes;
$V - v$ = loss of pressure in feeder in volts;
r = resistance of 1 mile of copper conductor of 1 square in tional area in ohms.

$$V - v = \frac{nrC}{a}.$$

* Standard (British) Wire-gauge.

nductor with this current flowing in it is

$$\text{and} \quad V - v = \frac{0.0455nC}{a}.$$

the case of a three-wire feeder, let $p_1 q_1$ and
nductors, and let $p'q'$ represent the middle
he feeding-point and q_1, q', q_2 at the generat-

each of the outer conductors in square inches;
middle conductor;
a conductor of feeder;
nd p' in volts at generating station;
nd p_2 in volts at generating station;
nd q' in volts at feeding-point;
nd q_2 in volts at feeding-point;
peres;
nperes;
of copper conductor of 1 square inch sectional

$$\frac{- C_2}{a'} \Big\}; \qquad V_2 - v_2 = nr \Big\{ \frac{C_2}{a} - \frac{C_1 - C_2}{a'} \Big\}.$$

$= v_2$, and if C_1 is greater than C_2, V_1 is greater
pressure in the middle wire; this result shows
in circuit with the two outer conductors.
a; then, if the greatest want of balance between
s of the three-wire system is m* per cent of the
heavily loaded section, and if C_1 is the maximum
er conductors of the feeder under consideration,
$\left(1 - \frac{m}{100}\right)$, and consequently $C_1 - C_2$ will not be

$$\frac{200 + m}{200}; \qquad V_2 - v_2 = \frac{nrC_1}{a} \times \frac{200 - m}{200};$$

equal to V—the pressure required to be main-
ng-point—we can calculate V_1 and V_2 for given
loying the value of m, which we estimate should
e.

ow that the difference in the pressures required
o sections of a three-wire feeder increases with
eeder; hence the regulators on each of the outer
ivalent to a variable resistance having at least

rea of the middle conductor one half of that of
s, but this is not invariably the case.

rom the law $C = \frac{E}{R}$ it is seen that with any pres-
ecome very great if R is made very small. In
ce becomes small and the current therefore great.
-circuiting a current.

c Transmission. (R. G. Blaine, *Eng'g*, June
rule for the most economical section of conductor

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1 to m may vary from 10 to 25, according to the
customers to one section or the other, or both
nditions. At a certain station suppl
ure system to about 25,000 8-c.p. lam
exceeded 7 or 8.

is that for which the "annual interest on capital outlay is equal to annual cost of energy wasted," and its practical outcome is that the

the copper conductor should be such that its resistance per mile

(C being the current in amperes).

Tables have been compiled by Professor Forbes and others in accord with modifications of Sir W. Thomson's rule. For a given entering power the question is merely one as to what current density, or how many amperes per square inch of conductor, should be employed. Sir W. Thomson's rule gives about 393 amperes per square inch, and Professor Forbes's tables—for a medium cost of one electrical horse-power per hour—current density of about 380 amperes per square inch as most econ

When a given horse-power is to be delivered at a given distance, the is somewhat different, and Professors Ayrton and Perry (*Electrician*, 1886) have shown that in that case both the current and resistance variables, and that their most economical values may be found from the following formulæ:

$$C = \frac{w}{P}(1 + \sin\phi), \quad \text{and} \quad r = \frac{P^2}{nw}\,\frac{\sin\phi}{(1 + \sin\phi)^2},$$

in which C = the proper current in amperes; r = resistance in ohms per mile which should be given to the conductor; P = pressure at entrance in volts; n = number of miles of conductor; w = power delivered in watts; ϕ = such an angle that $\tan\phi = nt \div P$, t being a constant depending on the price of copper, the cost of one electrical horse-power, interest, etc., may be taken as about 17.

In this case the current density should not remain constant, but should diminish as the length increases, being in all cases less than that calculated by Sir W. Thomson's rule.

EXAMPLE.—If the current for an electric railway is sent in at 200 volts horse-power being delivered, find the waste of power in heating the conductor, the distance being 5 miles and there being a return conductor.

Here $n = 10$, $t = 17$, $P = 200$; $\tan\phi = 170 \div 200 = .85$, $\phi = 40° 22'$, sin .6477.

Hence most economical resistance

$$r = \frac{200^2}{10 \times 74600} \times \frac{.6477}{1.6477^2} = .01279 \text{ ohm per mile,}$$

or .1279 ohm in its total length.

The most economical current, $C = \frac{74600}{200} \times 1.6477 = 614.58$ amperes,

the power wasted in heat, $= \frac{C^2R}{746} = \frac{614.58^2 \times .1279}{746} = 64.75$ horse-power

The following tables show the power wasted as heat in the conduct

HORSE-POWER WASTED IN TRANSMITTING POWER ELECTRICALLY TO A DISTANCE, THE ENTERING POWER BEING FIXED. PRESSURE AT ENT 200 VOLTS. CURRENT DENSITY, 380 AMPERES PER SQUARE INCH.

Horse-power sent in.*	Horse-power Wasted, the Distance to which the Power is Transmitted being one Mile (there being a Return Conductor).	Horse-power Wa Distance Five M
10	1.663	8.318
20	3 327	16 636
40	6.654	33.27
50	8.318	41.59
80	13.308	66.54
100	16.636	83.18
200	33.272	166.36

* horse-power at the generator terminals.

ENTRANCE, 2000 VOLTS.

Horse-power Wasted. Distance Five Miles.	Horse-power Wasted. Distance Ten Miles.	Horse-power Wasted. Distance Twenty Miles.
8.318	16.636	33.27
16.636	33.272	66.54
33.272	66.54	133.08
41.59	83.18	166.36
66.54	133.08	266.17
83.18	166.36	332.72
166.36	332.72	665.44

numbers that when the current density is fixed ional to the entering horse-power and the length versely proportional to the potential. For a nay be simply stated as

$$W = 16.6358 \frac{E}{P} \times l,$$

and P the pressure at entrance, and l the length of

IN ELECTRIC TRANSMISSION TO A GIVEN DISTANCE, LIVERED AT THE DISTANT END BEING FIXED. PRES-00 VOLTS. CURRENT AND RESISTANCE CALCULATED Y'S RULES.

power Wasted, stance to which er is Transmitted One Mile (there eing a Return Conductor).	Horse-power Wasted. Distance Five Miles.	Horse-power Wasted. Distance Ten Miles.
1.676	6.476	8.620
3.352	12.952	17.24
6.704	25.904	34.48
8.38	32.38	43.10
13.408	51.808	68.96
16.76	64.86	86.20
33.52	129.52	172.4

PRESSURE AT ENTRANCE, 2000 VOLTS.

Horse-power asted. Distance One Mile.	Horse-power Wasted. Distance Five Miles.	Horse-power Wasted. Distance Ten Miles.
1.716	8.484	16.763
3.432	16.968	33.526
6.864	33.938	67.052
8.58	42.42	83.815
13.728	67.87	134.104
17.16	84.84	167 63
34.32	169.68	335.26

er sent in, w = power delivered in watts, ance in ohms per mile, P = pressure ber of miles of conductor,

$$+ C^2r) \div 746 = H; \quad w = 746H - C^2r$$

and the formulæ for best current and resistance become

$$C = \frac{746H - C^2r}{P}(1 + \sin\phi); \quad r = \frac{P^2}{n(746H - C^2r} \times \frac{\sin\phi}{1+\sin\phi}$$

Energy wasted as heat in watts per mile $= C^2r = \dfrac{746H \sin\phi}{n + \sin\phi}$

Horse-power wasted per mile $= W_1 = \dfrac{H \sin\phi}{n + \sin\phi}$.

(ϕ = angle whose tangent $= nt \div P$, and the value of t corresp[illegible] current density of 380 amperes per sq. in. is 16 636.)

TABLE OF ELECTRICAL HORSE-POWE[illegible]

Formula: $\dfrac{\text{Volts} \times \text{Amperes}}{746} = \text{H.P.}$, or 1 volt-ampere = .00[illegible]

Read amperes at top and volts at side, or *vice versa.*

Amperes or Volts.	Volts or Amperes.										
	1	10	20	30	40	50	60	70	80	90	100
1	.00134	.0134	.0268	.0402	.0536	.0670	.0804	.0938	.1072	.1206	.1341
2	.00268	.0268	.0536	.0804	.1072	.1341	.1609	.1877	.2145	.2413	.2681
3	.00402	.0402	.0804	.1206	.1609	.2011	.2413	.2815	.3217	.3619	.4022
4	.00536	.0536	.1072	.1609	.2145	.2681	.3217	.3753	.4290	.4826	.5362
5	.00670	.0670	.1341	.2011	.2681	.3351	.4022	.4692	.5362	.6032	.6703
6	.00804	.0804	.1609	.2413	.3217	.4022	.4826	.5630	.6434	.7239	.8043
7	.00938	.0938	.1877	.2815	.3753	.4692	.5630	.6568	.7507	.8445	.9384
8	.01072	.1072	.2145	.3217	.4290	.5362	.6434	.7507	.8579	.9652	1.072
9	.01206	.1206	.2413	.3619	.4826	.6032	.7239	.8445	.9652	1.086	1.206
10	.01341	.1341	.2681	.4022	.5362	.6703	.8043	.9383	1.072	1.206	1.341
11	.01475	.1475	.2949	.4424	.5898	.7373	.8847	1.032	1.180	1.327	1.475
12	.01609	.1609	.3217	.4826	.6434	.8043	.9652	1.126	1.287	1.448	1.609
13	.01743	.1743	.3485	.5228	.6970	.8713	1.046	1.220	1.394	1.568	1.743
14	.01877	.1877	.3753	.5630	.7507	.9384	1.126	1.314	1.501	1.689	1.877
15	.02011	.2011	.4022	.6032	.8043	1.005	1.206	1.408	1.609	1.810	2.011
16	.02145	.2145	.4290	.6434	.8579	1.072	1.287	1.501	1.716	1.930	2.145
17	.02279	.2279	.4558	.6837	.9115	1.139	1.367	1.595	1.823	2.051	2.279
18	.02413	.2413	.4826	.7239	.9652	1.206	1.448	1.689	1.930	2.172	2.413
19	.02547	.2547	.5094	.7641	1.019	1.273	1.528	1.783	2.037	2.292	2.547
20	.02681	.2681	.5362	.8043	1.072	1.340	1.609	1.877	2.145	2.413	2.681
21	.02815	.2815	.5630	.8445	1.126	1.408	1.689	1.971	2.252	2.533	2.815
22	.02949	.2949	.5898	.8847	1.180	1.475	1.769	2.064	2.359	2.654	2.949
23	.03083	.3083	.6166	.9249	1.233	1.542	1.850	2.158	2.467	2.775	3.083
24	.03217	.3217	.6434	.9652	1.287	1.609	1.930	2.252	2.574	2.895	3.217
25	.03351	.3351	.6703	1.005	1.341	1.676	2.011	2.346	2.681	3.016	3.351
26	.03485	.3485	.6971	1.046	1.394	1.743	2.091	2.440	2.788	3.137	3.485
27	.03619	.3619	.7239	1.086	1.448	1.810	2.172	2.534	2.895	3.257	3.619
28	.03753	.3753	.7507	1.126	1.501	1.877	2.252	2.627	3.003	3.378	3.753
29	.03887	.3887	.7775	1.166	1.555	1.944	2.332	2.721	3.110	3.499	3.887
30	.04022	.4022	.8043	1.206	1.609	2.011	2.413	2.815	3.217	3.619	4.022
31	.04156	.4156	.8311	1.247	1.662	2.078	2.493	2.909	3.324	3.740	4.156
32	.04290	.4290	.8579	1.287	1.716	2.145	2.574	3.003	3 432	3.861	4.290
33	.04424	.4424	.8847	1.327	1.769	2.212	2.654	3.097	3.539	3.981	4.424
34	.04558	.4558	.9115	1.367	1.823	2.279	2.735	3.190	3.646	4.102	4.558
35	.04692	.4692	.9384	1.408	1.877	2.346	2.815	3.284	3.753	4.223	4.692
36	.04826	.4826	.9652	1.448	1.930	2.413	2.895	3.378	3.861	4.343	4.826
37	.04960	.4960	.9920	1.488	1.984	2.480	2.976	3.472	3.968	4.464	4.960
38	.05094	.5094	1.019	1.528	2.038	2.547	3.056	3.566	4.075	4.585	5.094
39	.05228	.5228	1.046	1.568	2.091	2.614	3.137	3.660	4.182	4.705	5.228
40	.05362	.5362	1.072	1.609	2.145	2 681	3.217	3.753	4.290	4.826	5.362
41	.05496	.5496	1.099	1.649	2.198	2.748	3.298	3.847	4.397	4.946	5.496
42	.05630	.5630	1.126	1.689	2.252	2.815	3.378	3.941	4.504	5.067	5.630
43	.05764	.5764	1.153	1.729	2.306	2.882	3.458	4.035	4.611	5.187	5.764
44	.05898	.5898	1.180	1.769	2.359	2.949	3.539	4.129	4.719	5.308	5.898
45	.06032	.6032	1.206	1.810	2.413	3.016	3.619	4.223	4.826	5.429	6.032
46	.06166	.6166	1.233	1.850	2.467	3.083	3.700	4.316	4.933	5.550	6.166
47	.06300	.6300	1.260	1.890	2.520	3.150	3.780	4.410	5.040	5.670	6.300
48	.06434	.6434	1.287	1.930	2.574	3.217	3.861	4.504	5.148	5.791	6.434
49	.06568	.6568	1.314	1.970	2.627	3.284	3.941	4.598	5.255	5.912	6.568
[illegible]	[illegible]	.6703	1.341	2.011	2.681	3.351	4.022	4.692	5.362	6.032	6.703

ICAL HORSE-POWERS–(*Continued.*)

Volts or Amperes.

	40	50	60	70	80	90	100	110	120
	2.949	3.686	4.424	5.161	5.898	6.635	7.373	8.110	8.847
	3.217	4.022	4.826	5.630	6.434	7.239	8.043	8.847	9.652
	3.485	4.357	5.228	6.099	6.970	7.842	8.713	9.584	10.46
	3.753	4.692	5.630	6.568	7.507	8.445	9.384	10.32	11.26
	4.021	5.027	6.032	7.037	8.043	9.048	10.05	11.06	12.06
	4.290	5.362	6.434	7.507	8.579	9.652	10.72	11.80	12.87
	4.558	5.697	6.836	7.976	9.115	10.26	11.39	12.53	13.67
	4.826	6.032	7.239	8.445	9.652	10.86	12.06	13.27	14.48
	5.094	6.367	7.641	8.914	10.18	11.46	12.73	14.01	15.28
	5.362	6.703	8.043	9.384	10.72	12.06	13.41	14.75	16.09
	10.72	13.41	16.09	18.77	21.45	24.13	26.81	29.49	32.17
	16.09	20.11	24.13	28.15	32.17	36.19	40.22	44.24	48.26
	21.45	26.81	32.17	37.53	42.90	48.26	53.62	58.98	64.34
	26.81	33.51	40.22	46.92	53.62	60.32	67.03	73.73	80.43
	32.17	40.22	48.26	56.30	64.34	72.39	80.43	88.47	96.52
5	37.53	46.92	56.30	65.68	75.07	84.45	93.84	103.2	112.6
7	42.90	53.62	64.34	75.07	85.79	96.52	107.2	118.0	128.7
9	48.26	60.32	72.39	84.45	96.52	108.6	120.6	132.7	144.8
22	53.62	67.03	80.43	93.84	107.2	120.6	134.1	147.5	160.9
43	107.2	134.1	160.9	187.7	214.5	241.3	268.1	294.9	321.7
6	160.9	201.1	241.3	281.5	321.7	361.9	402.2	442.4	482.6
9	214.5	268.1	321.7	375.3	429.0	482.6	536.2	589.8	643.4
1	268.1	335.1	402.2	469.2	536.2	603.2	670.3	737.3	804.3
3	321.7	402.2	482.6	563.0	643.4	723.9	804.3	884.7	965.2
.5	375.3	469.2	563.0	656.8	750.7	844.5	938.4	1032	1126
.7	429.0	536.2	643.4	750.7	857.9	965.2	1072	1180	1287
.9	482.6	603.2	723.9	844.5	965.2	1086	1206	1327	1448
.2	536.2	670.3	804.3	938.3	1072	1206	1341	1475	1609

e wire table on the following page (from a circular of & Mfg. Co.) shows at a glance the size of wire neces- ion of any given current over a known distance with rop, for 100-volt and 500-volt circuits, with varying by which this table has been calculated is

$$\frac{D \times 1000}{C \times 2L} = R,$$

volts drop in electro-motive force, C the current, L the amo to the point of distribution, and R the line resist- usand feet.

red the size of wire necessary to carry a current of 60 f 650 feet with a loss of 5% at 100 volts.

ble, under 60 amperes, we find the given distance, 650 rizontal line and under 5% drop at 100 volts, we find No. size required.

size will be required for 10 amperes 2000 feet, with a s.

find 1930—the nearest figure to 2000—and in the same 10% at 500 volts find No. 11, the size required.

ulæ for Incandescent Lighting. (W. D. *d*, Oct. 15, 1892.)—A formula for calculating wiring

$$A = \frac{2150W}{aE^2}LN, \quad \text{or,} \quad A = \frac{2150LC}{aE},$$

circular mils; W = watt rating of lamps; E = volt- centre of distribution, in feet; N = number of lamps; op; C = current in amperes.

0; amperes, 100; feet to centre of distribution, 100;

$$\frac{0 \times 100 \times 100}{2 \times 50} = 215,000 \text{ circular mils,}$$

gauge.

and the formulæ for best current and resistance become

$$C = \frac{746H - C^2r}{P}(1 + \sin\phi); \quad r = \frac{P^2}{n(746H - C^2r} \times \frac{\sin\phi}{1 + \sin\phi}.$$

Energy wasted as heat in watts per mile $= C^2r = \dfrac{746H\sin\phi}{n + \sin\phi}$.

Horse-power wasted per mile $= W_1 = \dfrac{H\sin\phi}{n + \sin\phi}$.

(ϕ = angle whose tangent $= nt + P$, and the value of t corresponding to a current density of 380 amperes per sq. in. is 16.636.)

TABLE OF ELECTRICAL HORSE-POWERS.

Formula: $\dfrac{\text{Volts} \times \text{Amperes}}{746} = \text{H.P.}$, or 1 volt-ampere = .0013405 H.P.

Read amperes at top and volts at side, or *vice versa.*

Amperes or Volts.	Volts or Amperes.												
	1	10	20	30	40	50	60	70	80	90	100	110	120
1	.00134	.0134	.0268	.0402	.0536	.0670	.0804	.0938	.1072	.1206	.1341	.1475	.160
2	.00268	.0268	.0536	.0804	.1072	.1341	.1609	.1877	.2145	.2413	.2681	.2949	.32
3	.00402	.0402	.0804	.1206	.1609	.2011	.2413	.2815	.3217	.3619	.4022	.4424	.48
4	.00536	.0536	.1072	.1609	.2145	.2681	.3217	.3753	.4290	.4826	.5362	.5898	.64
5	.00670	.0670	.1341	.2011	.2681	.3351	.4022	.4692	.5362	.6032	.6703	.7373	.80
6	.00804	.0804	.1609	.2413	.3217	.4022	.4826	.5630	.6434	.7239	.8043	.8847	.96
7	.00938	.0938	.1877	.2815	.3753	.4692	.5630	.6568	.7507	.8445	.9384	1.032	1.12
8	.01072	.1072	.2145	.3217	.4290	.5362	.6434	.7507	.8579	.9652	1.072	1 180	1.28
9	.01206	.1206	.2413	.3619	.4826	.6032	.7239	.8445	.9652	1.086	1.206	1.327	1.44
10	.01341	.1341	.2681	.4022	.5362	.6703	.8043	.9383	1.072	1.206	1.341	1.475	1.60
11	.01475	.1475	.2949	.4424	.5898	.7373	.8847	1.032	1.180	1.327	1.475	1.622	1.76
12	.01609	.1609	.3217	.4826	.6434	.8043	.9652	1.126	1.287	1.448	1.609	1.769	1.93
13	.01743	.1743	.3485	.5228	.6970	.8713	1.046	1.220	1.394	1.568	1.743	1.917	2.09
14	.01877	.1877	.3753	.5630	.7507	.9384	1.126	1.314	1.501	1.689	1.877	2.064	2.25
15	.02011	.2011	.4022	.6032	.8043	1.005	1.206	1.408	1.609	1.810	2.011	2.212	2.41
16	.02145	.2145	.4290	.6434	.8579	1.072	1.287	1.501	1.716	1.930	2.145	2.359	2.57
17	.02279	.2279	.4558	.6837	.9115	1.139	1.367	1.595	1.823	2.051	2.279	2.507	2.73
18	.02413	.2413	.4826	.7239	.9652	1.206	1.448	1.689	1.930	2.172	2.413	2.654	2.89
19	.02547	.2547	.5094	.7641	1.019	1.273	1.528	1.783	2.037	2.292	2.547	2.801	3.05
20	.02681	.2681	.5362	.8043	1.072	1.340	1.609	1.877	2.145	2.413	2.681	2.949	3.21
21	.02815	.2815	.5630	.8445	1.126	1.408	1.689	1.971	2.252	2.533	2.815	3.097	3.37
22	.02949	.2949	.5898	.8847	1.180	1.475	1.769	2.064	2.359	2.654	2.949	3.244	3.53
23	.03083	.3083	.6166	.9249	1.233	1.542	1.850	2.158	2.467	2.775	3.083	3.391	3.70
24	.03217	.3217	.6434	.9652	1.287	1.609	1.930	2.252	2.574	2.895	3.217	3 539	3.86
25	.03351	.3351	.6703	1.005	1.341	1.676	2.011	2.346	2.681	3.016	3.351	3.686	4.02
26	.03485	.3485	.6971	1.046	1.394	1.743	2.091	2.440	2.788	3.137	3.485	3.834	4.18
27	.03619	.3619	.7239	1.086	1.448	1.810	2.172	2.534	2.895	3.257	3.619	3.981	4.34
28	.03753	.3753	.7507	1.126	1.501	1.877	2.252	2.627	3.003	3.378	3.753	4.129	4.50
29	.03887	.3887	.7775	1.166	1.555	1.944	2.332	2.721	3.110	3.499	3.887	4.276	4.66
30	.04022	.4022	.8043	1.206	1.609	2.011	2.413	2.815	3.217	3.619	4.022	4.424	4.82
31	.04156	.4156	.8311	1.247	1.662	2.078	2.493	2.909	3.324	3.740	4.156	4.571	4.98
32	.04290	.4290	.8579	1.287	1.716	2.145	2.574	3.003	3 432	3.861	4.290	4.719	5.14
33	.04424	.4424	.8847	1.327	1.769	2.212	2.654	3.097	3.539	3.986	4.424	4.866	5.30
34	.04558	.4558	.9115	1.367	1.823	2.279	2.735	3.190	3.646	4.102	4.558	5.013	5.46
35	.04692	.4692	.9384	1.408	1.877	2.346	2.815	3.284	3.753	4.223	4.692	5.161	5.63
36	.04826	.4826	.9652	1.448	1.930	2.413	2.895	3.378	3.861	4.343	4.826	5.308	5.79
37	.04960	.4960	.9920	1.488	1.984	2.480	2.976	3.472	3.968	4.464	4.960	5.456	5.95
38	.05094	.5094	1.019	1.528	2.038	2.547	3.056	3.566	4.075	4.585	5.094	5.603	6.11
39	.05228	.5228	1.046	1.568	2.091	2.614	3.137	3.660	4.182	4.705	5.228	5.751	6.27
40	.05362	.5362	1.072	1.609	2.145	2 681	3.217	3.753	4.290	4.826	5.362	5.898	6.43
41	.05496	.5496	1.099	1.649	2.198	2.748	3.298	3.847	4.397	4.946	5.496	6.046	6.59
42	.05630	.5630	1.126	1.689	2.252	2.815	3.378	3.941	4.504	5.067	5.630	6.193	6.75
43	.05764	.5764	1.153	1.729	2.306	2.882	3.458	4.035	4.611	5.187	5.764	6.341	6.91
44	.05898	.5898	1.180	1.769	2.359	2.949	3.539	4.129	4.719	5.308	5.898	6.488	7.07
45	.06032	.6032	1.206	1.810	2.413	3.016	3.619	4.223	4.826	5.439	6.032	6.635	7.23
46	.06166	.6166	1.233	1.850	2.467	3.083	3.700	4.316	4.933	5.550	6.166	6.783	7.40
47	.06300	.6300	1.260	1.890	2.520	3.150	3.780	4.410	5.040	5.670	6.300	6.930	7.56
48	.06434	.6434	1.287	1.930	2.574	3.217	3.861	4.504	5.148	5.791	6.434	7.078	7.72
49	.06568	.6568	1.314	1.970	2.627	3.284	3.941	4.598	5.255	5.912	6.568	7.225	7.88
50	.06703	.6703	1.341	2.011	2.681	3.351	4.022	4.692	5.362	6.032	6.703	7.373	8.04

acy of a motor being given, the size of the con-
an be found from the following formula:

$$\frac{0,400,000 \times H.P. \times L}{aE^2 \times \text{efficiency}}.$$

0; volts, 500; drop, 3%; feed to dis ributing
r, 75%.

= 17,109 circular mils, or about No. 8 B. & S.

r Long-distance Transmission.

nghouse El. & Mfg. Co.)

OR THE DELIVERY OF ONE MECHANICAL HORSE-
ITH 1000, 2000, 3000, 4000, 5000, AND 10,000 VOLTS
AT TERMINALS OF LOWERING TRANSFORMERS.

ors (drop), equals 20%.
nty miles.

mile of single distance, 11,000 feet, to allow for

ents per pound.

3000 v.	4000 v.	5000 v.	10,000 v.
$0.23	$0.13	$0.08	$0.02
0.93	0.52	0.33	0.08
2.08	1.17	0.75	0.19
3.70	2.08	1.33	0.33
5.78	3.25	2.08	0.52
8.32	4.68	3.00	0.75
11.30	6.37	4.08	1.02
14.80	8.32	5.33	1.33
18.70	10.50	6.74	1.69
23.14	13.01	8.33	2.08
28.00	15.75	10.08	2.52
33.30	18.70	12.00	3.00
39.00	22.00	14.08	3.52
45.30	25.50	16.32	4.08
52.00	29.25	18.72	4.68
59.00	33.30	21.32	5.33
67.00	37.60	24.00	6.00
75.00	42.20	27.00	6.75
83.50	47.00	30.00	7.50
92.60	52.00	33.32	8.33

od of calculating leads for wiring for electric
d Hering in Trans. A. I. E. E., 1891. He furnishes
sets of diagonal straight-line diagrams so con-
under the general formula for wiring may be
by simply locating three points in succession on

pon which the chart is based is that for any
variable quantities, one of which is the product
r two, the "curves" representing their relative
presented by a series of straight diagonal lines
or zero-point. Such a set of lines will therefore
calculations graphically for that formula. For
olts × amperes; the constant 746 does not con-
es of diagonal lines properly spaced will there-
he horse-power, the volts, or the amperes, when

other horizontal, and the diagonal
t one unit (or a number of units
es" straight lines the diagonals

COST OF COPPER REQUIRED TO DELIVER ONE MECHANICAL HORSE-POWE MOTOR-SHAFT WITH VARYING PERCENTAGES OF LOSS IN CONDUCTORS THE ASSUMPTION THAT THE POTENTIAL AT MOTOR TERMINALS IS IN E CASE 3000 VOLTS.

Distances equal one to twenty miles.
Motor efficiency equals 90%.
Length of conductor per mile of single distance, 11,000 feet, to allow sag.
Cost of copper equals 16 cents per pound.

Miles.	10%	15%	20%	25%	30%
1	$0.52	$0.33	$0.23	$0.17	$0.13
2	2.08	1.31	0.93	0.69	0.54
3	4.68	2.95	2.08	1.55	1.21
4	8.32	5.25	3.70	2.77	2.15
5	13.00	8.20	5.78	4.33	3.37
6	18.70	11.75	8.32	6.23	4.80
7	25.50	16.00	11.30	8.45	6.60
8	33.30	21.00	14.80	11.00	8.60
9	42.20	26.60	18.75	14.00	10.90
10	52.05	32.78	23.14	17.31	13.50
11	63.00	39.75	28.00	21.00	16.30
12	75.00	47.20	33.30	24.90	19.40
13	88.00	55.30	39.00	29.20	22.80
14	102.00	64.20	45.30	33.90	26.40
15	117.00	73.75	52.00	38.90	30.30
16	133.00	83.80	59.00	44.30	34.50
17	150.00	94.75	67.00	50.00	39.00
18	169.00	106.00	75.00	56.20	43.80
19	188.00	118.00	83.50	62.50	48.70
20	208.00	131.00	92.60	69.25	54.00

to represent one of the two quantities which is equal to the quotient of other two, and not the one which is equal to the product of the other because the curves would then be hyperbolas. In the example given diagonals must represent volts or amperes, but not horse-powers. The stants in such formulæ affect only the positions of the diagonals; alth they increase considerably the work of arithmetically calculating the resu they do not affect in the least the graphical calculations after the diagra are once drawn.

The general formula for wiring is :

$$\text{Cross-section} = \frac{\text{current for one lamp} \times \text{No. of lamps} \times \text{distance} \times \text{consta}}{\text{loss in volts}}$$

containing six quantities only, one of which is always constant, being eq to twice the mil-foot resistance of copper, if the cross-section is in circu mils. Calculations involving three of these five quantities may readily made graphically by means of a single set of diagonal lines.

In Mr. Hering's method the formula is split up into three smaller each of which contains no more than three variable quantities. E formula can then be calculated separately by a simple diagram, as scribed, thus permitting the whole formula to be calculated graphically.

To do this, let the first diagram perform the calculation,

$$x = \frac{\text{current for one lamp}}{\text{loss in volts}},$$

in which x is a mere auxiliary quantity. Let a second similar diag perform the next calculation,

$$y = x \times \text{number of lamps};$$

and a third [illegible] the final calculation,

$$\textit{cross-section} = y \times \text{distance}.$$

bined with any one of these, it is immaterial ation may at first seem to complicate matters antities, x and y. These, however, are easily vice of placing the three diagrams together, side hat the two x scales coincide, and similarly the is one has merely to pass directly from one set of erform the successive steps of the calculation, bout the intermediate auxiliary quantities. These rrespond, and are equal to the successive products ained in the successive arithmetical multiplications ive quantities in the formula, which cannot, of aking the calculations arithmetically.

r required for Long-distance Trans-sson (Trans. Tech. Socy. of the Pacific Coast, vol. ng formula:

$$V = \frac{D^2}{E^2} \text{H.P.} \frac{(100 - L)}{L} 266.5,$$

copper wire in pounds; D, the distance in miles; tor in hundreds of volts; H.P., the horse-power L, the per cent of line loss.

horse-power ten miles with 10 per cent loss, and tor, we have

$$\times 200 \times \frac{(100 - 10)}{10} \times 266.5 = 53,300 \text{ lbs.}$$

ng-distance Transmission. (F. R. Hart, nechanical efficiency of a system is the ratio of the ynamo-electric machines at one end of the line to he electric motors at the distant end. The com- namo or motor varies with its load. The maximum ines should not be under 90% and is seldom above orable conditions, then, we must expect a loss of l 9% in the motor. The loss in transmission, due to or "drop" in the line, is governed by the size of itions remaining the same. For a long-distance vill vary from 5% upwards. With a loss of 5% in the f transmission will be slightly under 79%. With a efficiency would be slightly under 75%. We may t of the efficiency with the apparatus of to-day. listance power transmission by electricity may be al classes: (1) Those using continuous current; (2) urrent; and (3) regenerating or "motor-dynamo" ns of each of these general classes are tabulated as

Low voltage		One machine. Machines in parallel.
High voltage		One machine. Machines in parallel. Machines in series.
		2 machines in series. Machines in multiple series.
le-wire		Machines in series.
ating single phase		Without conversions. With conversions.
ating multiphase		Without conversions. With conversions.

ating continuous.

ating converter; line converter; alternating con- ous.

uous-continuous.

l reconversion of any system.

s of these systems vary with each parti *general way may be tabulated as bel*

	System.		Advantages.	Disadvantages.
Continuous.	2-wire	Low voltage.	Safety, simplicity.	Expense for copper.
		High voltage.	Economy, simplicity.	Danger, difficulty building machines.
	3-wire.		Low voltage on machines and saving in copper.	Not saving enough copper for long tances. Necessity "balanced" system
	Multiple-wire.		Low voltage at machines and saving in copper.	
Alternating.	Single phase.		Economy of copper.	Cannot start under lo Low efficiency.
	Multiphase.		Economy of copper, synchronous speed unnecessary; applicable to very long distances.	Complexity. Lower ficiency of termi apparatus. Not as y "standard."
	Motor-dynamo.		High-voltage transmission. Low-voltage delivery.	Expensive. Low efficiency.

There are many factors which govern the selection of a system. For ea problem considered there will be found certain fixed and certain unfix conditions. In general the fixed factors are: (1) capacity of source power; (2) cost of power at source; (3) cost of power by other means at poi of delivery; (4) danger considerations at motors; (5) operation conditio (6) construction conditions (length of line, character of country, etc.). T partly fixed conditions are: (7) power which must be delivered, i.e., the ef ciency of the system; (8) size and number of delivery units. The variab conditions are: (9) initial voltage; (10) pounds of copper on line; (11) ori nal cost of all apparatus and construction; (12) expenses, operating (fi charges, interest, depreciation, taxes, insurance, etc.); (13) liability of trou and stoppages; (14) danger at station and on line; (15) convenience in op ating, making changes, extensions, etc. Assuming that the cost of dyn mos, motors, etc., will be approximately the same whatever the init pressure, the great variation in the cost of wire at different pressures shown by Mr. Hart in the following figures, giving the weights of copp required for transmitting 100 horse-power 5 miles:

Voltage.	Drop 10 per cent.	Drop 20 per cent.
2,000	16,800 lbs.	8,400 lbs.
2,000	7,400 "	3,700 "
10,000	620 "	310 "

Efficiency of a Combined Engine and Dynamo. — A co pound double-crank Willans engine mounted on a single base with dynamo of the Edison-Hopkinson type was tested in 1890, with results follows: The low-pressure cylinder is 14 in. diam., 16 in. stroke; stea pressure 120 lbs. It is coupled to a dynamo constructed for an output of amperes at 110 volts when driven at 430 revolutions per minute. The arm ture is of the bar construction, is plain shunt-wound, and is fitted wit commutator of hard-drawn copper with mica insulation. Four brushes carried on each rocker-arm.

Resistance of magnets	16.	ohms
Resistance of armature	0.0055	"
I.H.P	83.3	
E.H.P	72.2	
Total efficiency	86.7 per cent	
Consumption of water per I.H.P. hour	21.6 pounds	
Consumption of water per E.H.P. hour	25	"

The engine and dynamo were worked above their full normal outp *which fact would tend to slightly increase* the efficiency.

The elec[illegible] were: Loss in magnet coils, 756 watts, equal to 1. *[illegible] 1886 watts, equal to* 2.6%; so that the electrical efficien esistance alone was 96%. The remainder of se-power, is due to friction of engine and e.

f a Generator and Motor.—A twelve- Bodie, Cal., is described by T. H. Leggett gle-phase alternating current is used. The 0 K. W. constant-potential 12-pole machine, The motor is a synchronous constant-po- wer. It is brought up to speed by a 10-H.P. the electrical efficiency of the generator and ts:

T ON GENERATOR.

	Amperes	Volts.	Watts.
....	15.8	60	948
....	18.2	78	1419.6
ohms.			
....	....	....	664.72
....	....	....	3032.32
....	20	3414	68280

efficiency of generator, 95.559%.

'EST ON MOTOR.

	Amperes	Volts.	Watts.
....	52	62.4	3244.8
ns.			
... .	....	....	560.0
....	....	....	3804.08
....	20	3110	62200

al efficiency of motor, 93.883%.

trical Pumping-plant. (*Eng. & M.* g-plant at a mine at Normanton, England, elow:

f 20½ × 48-in. engines running at 20 revs. per s giving 690 volts and 59 amperes. The cur- ed into the mine by an insulated cable about connected to two 50-h.p. motors which oper- pumps, with rams 6 in. and 4½ in. diam. and gainst which the pumps operate is 890 feet. s there is also a set of gearing for driving a rope system, and a set of three-throw ram- -inch stroke can also be thrown into gear. at either motor can operate any or all three escribed. Indicator-diagrams gave the fol-

....	6.9 H.P.	9.4%
....	4.8 "	6.5%
....	6.7 "	9.4%
umps empty..........	10.2 "	14.0%
gh 890 feet..........	31.5 "	43.1%
nd rising main.......	12.9 "	17.6%
	73.0 H.P.	100.0%

vere obtained the total efficiency of the plant ose to 47%.

Distribution.—Kapp, Electric T ransmission Handbook; Martin an plications; Hospitalier, Polyphas ere the load is spasmodic, with variations at most but a few minutes. The stored heat rmous in quantity, and responds instantly to a to say, the boiler is an immense reservoir of ain upon it is not continued too long, it will its nominal capacity, and without any effect

t will depend upon the type of engine. With ed by Mr. Church, running non-condensing, an ter actually evaporated per I.H.P. per hour will encies. The engine duty under an average uni- thing from the duty under a variable load rep- Under the uniform load, 23 pounds of water performance, and the boiler could be propor- figure. Under the violent fluctuations of rail- ty of the engine will rise to about 28 pounds, e load is taken, and the boiler proportioned for ufficient margin. Other compound engines not h secures uniformity of duty will range up to at loads, and often to 60 pounds, and represent an an 35 to 40 pounds. The same is true of every engine, whether high speed or low speed, both of falling back of fuel duty under variable load.

CTRIC LIGHTING.

y required to produce Light.—According ity of energy, measured in watts, required to pro- e candle-power, measured by the light given out s follows for different light-giving substances:

4 watts	Coal gas............	68 watts.
4 "	Cannel gas..........	48 "
6 "	Incandescent lamp..	15 "
0 "	Arc lamp............	3 "
7 "		

production are about 1 for the arc lamp; 6 for the e mineral-oil lamp; 10 for the gas-light; 67 for the

t Lamps. (*Eng'g*, Sept. 1, 1893, p. 282.)—From srs. Siemens and Halske, Berlin, it appears that scent lamps at different expenditure of watts per

wer.......	1.5	2	2.5	3	3.5
....	45	200	450	1000	1000

y Tests of Lamps. (P. G. Gossler, *Elec.* mps burning at a voltage above that for which greater illuminating power than 16 candles, but is very considerably shortened. It has been ob- from the factory do not average the same candle- fferent invoices; that is, lamps which are received quite uniform throughout that lot, but they vary nade at other times.

ow the different illuminating-powers of a 16.c.p., arious voltages from 25 to 80 volts:

50	52.5	55.6	59.5	62	68.2	72.5	80
1.055	1.097	1.161	1.226	1.29	1.419	1.484	1.58
15.8	20.5	28.4	39.3	50.7	74.5	103.2	141
52.75	57.57	64.55	72.92	79.98	96.78	[illegible]	[illegible]
3.34	2.81	2.30	1.96	1.58	[illegible]	[illegible]	[illegible]

. Robinson, M.I.C.E., Eng'g News, *arc-lamp is the most economical.*

uminating-power of an arc. It is now gener-
0 amperes and not less than 45 volts between
The quality of the carbons will determine
ended in obtaining the most light or not, or
maximum intensity at one angle or another
ger the current passing in an arc, the less is
arcs with 4 amperes will have about 11 ohms,
l with 100 amperes .45 ohm.
m 50 to 60 lights in a series, each demanding
of, say, 3000 volts. In going beyond this the
reatly increased.

Electric Lighting.—Noll, How to Wire
ntinental Electric-light Central Stations, $6.00;
t Transformers in Theory and Practice, 2 vols.,
Electric Lighting, $1.50; Algave and Boulard,
roduction, and Application, $5.00.

CTRIC WELDING.

erally used consists of an alternating-current
atively high-potential current to the primary coil
ransformer, the secondary of which is made so
ort in length as to supply to the work currents
volts, and of very large volume or rate of flow.
ttached to the secondary terminals. Other forms
namos constructed to yield alternating currents
e to the welding-clamps, are used to a limited

eat of the metal to be welded has a decided influ-
in welding iron its comparatively low heat conduc-
aterially. (See papers by Sir F. Bramwell, Proc.
iii, p. 1; and Elihu Thomson, Trans. A. I. M. E., xix.

ge, Nov. 28, 1892, gives the following figures show-
required to weld axles and tires:

AXLE-WELDING.

	Seconds.
quires 25 H.P. for	45
quires 30 H.P. for	48
requires 35 H.P. for	60
requires 40 H.P. for	70
quires 75 H.P. for	95
quires 90 H.P. for	100

l time and power required for welding the square
he extra metal in it, but in part to the care which it
a perfect alignment.

TIRE-WELDING.

	Seconds.
quires 11 H.P. for	15
quires 23 H.P. for	25
quires 20 H.P. for	30
quires 23 H.P. for	40
ires 29 H.P. for	55
ires 42 H.P. for	62

for welding is of course that required for the actual
ent only, and does not include that consumed by
res in the machine, the removal of the upset and
s. From the data thus submitted, the cost of welding
for any locality where the price of fuel and cost of

cost of the fuel used under the boilers for produc-
velding is practically the same as the cost of fuel
me amount of work, taking into consideration the
el used in either case.
ly found that 2½-inch iron tubes ¼ inc
e net horse-power required at this sp
ower) per square inch of section. Be

size of arc-lamp at present manufactured requires a current of
amperes; but for steadiness and efficiency it is desirable to use not l
6 amperes. The candle-power of arc-lamps varies considerably, a
to the angle at which it is measured. The greatest intensity with con
current lamps is found at an angle of about 40° below the horizon
The following table gives the approximate candle-power at various
The height of the lamps should be arranged so as to give an angl
less than 7° to the most distant point it is intended to serve.

Lighting-power of Arc-lamps.

Current in Amperes.	Candle-power. Horizontal	At Angle of 7°.	At Angle of 10°.	At Angle of 20°.	Maxim Angle
6	92	175	207	322	4
8	156	300	350	546	7
10	220	420	495	770	11

The following data enable the coefficient of minimum lighting streets to be determined:

Let P = candle-power of lamps;
L = maximum distance from lamp in feet;
H = height of lamp in feet;
X = a coefficient.

The light falling on the unit area of pavement varies inversely as t of the distance from the lamp, and is directly proportional to the which it falls. This angle is nearly proportional to the height of divided by the distance. Therefore

$$X = \frac{P}{L^2} \times \frac{H}{L} \quad \text{or} \quad X = \frac{PH}{L^3}.$$

The usual standard of gas-lighting is represented by the amoun falling on the unit area of pavement 50 feet away from a 12-c.p. g feet high, which gives a coefficient as follows:

$$X = \frac{12 \times 9}{50^3} = 0.000864.$$

The minimum standard represents the amount of light on a u feet away from a 24-c.p. lamp, 9 ft. high, and gives the coefficient

Adopting the first of the above coefficients, Mr. Robinson calcu the before-mentioned sizes of arc-lights will give the same st light at the heights and distances stated in Table A. Table B corresponding distances, assuming the minimum standard to be

TABLE A.

Hgt. of Lamps.	20 ft.	25 ft.	30 ft.	35 ft.
Current in Amperes.	Max. distances served from lamp, in ft.			
6	160	175	190	202
8	185	202	220	235
10	205	225	243	260

TABLE B.

Height	20 ft.	25 ft.
Amperes.	Max. distanc from La	
6	130	144
8	150	165
10	170	190

The distances the lamps are apart would, of course, be doul tances mentioned in Tables A and B. One arc-lamp will take t from 3 to 6 gas-lamps, according to the locality, arrangement, an of light adopted. A scheme of arc-lighting, based on the substitu arc-light on the average for 3½ to 4 gas-lamps, would double the standard of light, while the average standard would be increas times.

Candle-power of the Arc-light. (Elihu Thomson, Feb. 28, 1891.)—With the long arc the maximum intensity of the li *to 50° downward from the horizontal.* The spherical candl *a fraction of the rated c.p., which is generally taken at the ... direction. For this reason the term 2000 c.*

-power of an arc. It is now gener-
s and not less than 45 volts between
ality of the carbons will determine
obtaining the most light or not, or
um intensity at one angle or another
current passing in an arc, the less is
ith 4 amperes will have about 11 ohms,
00 amperes .45 ohm.
60 lights in a series, each demanding
, 3000 volts. In going beyond this the
ncreased.

tric Lighting.—Noll, How to Wire
l Electric-light Central Stations, $6.00;
formers in Theory and Practice, 2 vols.,
ic Lighting, $1.50; Algave and Boulard,
on, and Application, $5.00.

C WELDING.

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perfect alignment.

TIRE-WELDING.	Seconds.
quires 11 H.P. for	15
quires 23 H.P. for	25
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f fuel used in either case.

nnedy found that 2½-inch iron tubes ¼ inch thick were
s, the *net horse-power* required at this speed being 23.4
rse-power) per square inch of section. Brass tubin

it at the radiator per pound of coal burned in the 0 + 2½ = 872 H.U. An ordinary steam-heating per lb. of coal for heating; hence the efficiency the efficiency of the steam-heating system as 872 *Eng'g News*, Aug. 9, '90; Mar. 30, '92; May 15, '93.)

ACCUMULATORS OR STORAGE-BATTERIES.

divided into two classes: viz., those in which the from the substance of the element itself, either ectro-chemical action, and those in which the lerated by the application of some easily reduci- of the former type are usually called Planté, and ," or "pasted."

ding a solution of acetate of lead found that per- d at the positive and metallic lead at the negative elements in a newly and fully charged Planté cell eroxide of lead, PbO_2, and spongy metallic lead, ositive and negative plates.

or if the cell be allowed to remain at rest, the sul- the solution enters into combination with the per- and partially converts it into sulphate. The acid cted from the electrolyte as the discharge proceeds, ion becomes less. In the charging operation this the reducible sulphates of lead which have been decomposed, the acid being reinstated in the liquid n increase in its density.

ntial developed by lead and lead peroxide immersed arly as may be, two volts.

gradually loses its electrical energy by local action, ying according to the circumstances of its prepara- f the cell. Various forms of both Planté and Faure in "Practical Electrical Engineering."

ted cells lead plates are coated with minium or ste with acidulated water. When dry these plates dilute H_2SO_4 and subjected to the action of the xide on the positive plate is converted into peroxide negative plate reduced to finely divided or porous

found that the initial electro-motive force of the volts, but after being allowed to rest some little bout 2.0 volts. The following tables show the size es of Faure cells, known as the E. P. S. cells. (Eng-

". S." Storage-cells, L Type.

Working Rate.		Capacity. Ampere hours.	Approximate External Dimensions.				Weight of Cell complete with Acid.
Charge	Dis-charge.		Length.	Width.	Height.	Height over all.	
Amper.	Amper.		in.	in.	in.	in.	lbs.
10 to 13	1 to 13	130	5¼	13¼	18¼	20½	74
10 " 13	1 " 13	130	5½	11½	13⅜	15¾	68
16 " 22	1 " 22	220	7½	13¼	18¼	20½	107
16 " 22	1 " 22	220	8	11½	13⅜	15¾	101
25 " 30	1 " 30	330	9½	13½	18¼	20½	143
25 " 30	1 " 30	330	9⅝	11¾	13⅝	15¾	[illegible]
38 " 46	1 " 46	500	14¾	13½	18¼	20½	
38 " 46	1 " 46	500	14¼	11¾	13¾	15⅞	
50 " 60	1 " 60	660	19¼	13¾	18¼	20½	
50 " 60	1 " 60	660	18½	12	13¾	15⅞	

"E. P. S." Cells, T Type.

Description of Cell.		Weight of Electrolyte.	Working Rate.		Capacity. Ampere hours.	Approx. External Dimensions.			
No. of Plates.	Material of Box.		Charge	Discharge.		Length.	Width.	Height.	Height over all
		lbs.	Amper.	Amper.		in.	in.	in.	in.
11	Wood (no lid)	10	16 to 20	1 to 20	66	6⅞	8¾	11⅝	13¼
	" (with lid)..	10	16 " 20	1 " 20	66	6⅞	8¾	11⅝	13¾
	Ebonite (no lid)..	10	16 " 20	1 " 20	66	6	7¾	11	12¾
15	Wood (no lid).....	14	24 " 28	1 " 30	95	8¼	8¾	11⅝	13¼
	" (with lid)..	14	24 " 28	1 " 30	95	8⅞	8¾	11⅝	13¾
	Ebonite (no lid)..	14	24 " 28	1 " 30	95	8	7¾	11	12¾
19	Wood (no lid)	18	30 " 35	1 " 40	120	11	8⅞	11⅝	13¼
	" (with lid)..	18	30 " 35	1 " 40	120	11	8⅞	11⅝	13¾
	Ebonite (no lid)..	18	30 " 35	1 " 40	120	10⅛	7¾	11	12¾
23	Wood (no lid)	22	38 " 42	1 " 50	145	13½	8⅞	11⅝	13¼
	" (with lid).	22	38 " 52	1 " 60	145	13¾	8⅞	11⅝	13¾
	Ebonite......... ..	22	38 " 42	1 " 50	145	12¼	7¾	11	12¾

For a very full description of various forms of storage batterie "Practical Electrical Engineering," part xii. For theory of the batter practice with the Julien battery, see paper on Electrical Accumulato P. G. Salom, Trans. A. I. M. E., xviii. 348.

Use of Storage-batteries in Power and Light Stati (*Iron Age*, Nov. 2, 1893.)—The storage-batteries in the Edison stati Fifty-third Street, New York, relieve the other stations at the hours of load, by delivering into the mains a certain amount of current that otherwise have to come, and at greater loss or "drop," from one or an of the stations connecting with the network of mains. Hence the loa be varied more or less arbitrarily at these stations according to the p tion of load that the larger stations are desired or able to carry.

The battery consists of 140 cells each of about 1000 ampere-hour cap weighing some 750 lbs., and of about 48 inches in length, 21 inches in and 15 inches in depth. The battery has a normal discharge rate of 200 amperes, but can be discharged, if necessary, at 500 amperes.

A test made when the station was running only 12 hours per day noon to midnight, showed that the battery furnished about 23.2% of the energy delivered to the mains. The maximum rate of discharge at by the battery was about 270 amperes. Thus, in this case, we have ample of a battery which is used for the purpose: 1. Of giving a l station machinery that would otherwise be idle. 2. Utilizing the energy to increase the rate of output of the station at the time of load, which would otherwise necessitate greater dynamo capacity.

The Working Current, or Energy Efficiency, of a se cell is the ratio between the value of the current or energy expended charging operation, and that obtained when the cell is discharged a specified rate.

In a lead storage cell, if the surface and quantity of active mate accurately proportioned, and if the discharge be commenced immed after the termination of the charge, then a current efficiency of as m 98% may be obtained, provided the rate of discharge is low and wel lated. In practice it is found that low rates of discharge are not econo and as the current efficiency always decreases as the discharge r creases, it is found that the normal current efficiency seldom excee and averages about 85%.

As the normal discharging electro-motive force of a lead seconda never exceeds 2 volts, and as an electro-motive force of from 2.4 to 2 is required at its poles to overcome both its opposing electro-motive and its internal resistance, there is an initial loss of 20% between the *required to charge it and that given out during its discharge.*

As the normal discharging potential is continually being reduced [illegible] discharge increases, it follows that an energy efficiency of [illegible]

of fact, a maximum of 75% and a mean of
of lead-sulphuric-acid storage-cells.

IICAL EQUIVALENTS.

Atomic Weight.†	Chemical Equivalent.	Electro-chemical Equivalent (milligrammes per coulomb).	Coulombs per gramme.	Grammes per ampere hour.
1.00	1.00	.010384	96293.00	0.03738
9.04	39.04	.40539	2467.50	1.45950
2.99	22.99	.23873	4188.90	0.85942
7.3	9.1	.09449	1058.30	3.40180
3.94	11.97	.12430	804.03	4.47470
6.2	65.4	.67911	1473.50	2.44480
7.66	107.66	1.11800	894.41	4.02500
3.00	31.5	.32709	3058.60	1.17700
3.00	63.00	.65419	1525.30	2.35500
9.8	99.9	1.03740	963.99	3.73450
9.8	199.8	2.07470	481.99	7.46900
7.8	29.45	.30581	3270.00	1.10090
7.8	58.9	.61162	1635.00	2.20180
5.9	18.64‡	.19356	5166.4	0.69681
5.9	27.95	.29035	3445.50	1.04480
8.6	29.3	.30425	3286.80	1.09530
4.9	32.45	.33696	2967.10	1.21330
6.4	103.2	1.07160	933.26	3.85780
5.96	7.98	.08286		
5.37	35.37	.36728		
6.53	126.53	1.31390		
9.75	79.75	.82812		
4.01	4.67	.04849		

atom-replacing power of an element com-
ency is unity.
of one atom of each element compared with
is unity.
aday's law showed that the electro-chemical
portional to its chemical equivalent. The
eight, and not to atomic weight ÷ valency,
italier, and others who have copied their
salt is an exception to Thompson's rule, as

CTROLYSIS.

ompound into its constituents by means of
ve the nomenclature relating to electroly-
be decomposed the Electrolyte, and the pro-
poles of the battery he called Electrodes.
essure exists he called the Anode, and the
oducts of decomposition he called Ions.
urrent of one ampere will deposit 0.017253
ver per second on one of the plates of a sil-
oyed being a solution of silver nitrate con-
t.
arly set free by a current of one

Knowing the amount of hydrogen thus set free, and the chemical equi lents of the constituents of other substances, we can calculate what wei of their elements will be set free or deposited in a given time by a gi current.

Thus the current that liberates 1 gramme of hydrogen will libera grammes of oxygen, or 107.7 grammes of silver, the numbers 8 and 1 being the chemical equivalents for oxygen and silver respectively.

To find the weight of metal deposited by a given current in a given ti find the weight of hydrogen liberated by the given current in the g time, and multiply by the chemical equivalent of the metal.

Thus: Weight of silver deposited in 10 seconds by a current of 10 amp = weight of hydrogen liberated per second × number seconds × cur strength × 107.7 = .00001038 × 10 × 10 × 107.7 = .11178 gramme.

Weight of copper deposited in 1 hour by a current of 10 amperes =

$$.00001038 \times 3600 \times 10 \times 31.5 = 11.77 \text{ grammes.}$$

Since 1 ampere per second liberates .00001038 gramme of hydro strength of current in amperes

$$= \frac{\text{weight in grammes of H. liberated per second}}{.00001038}$$

$$= \frac{\text{weight of element liberated per second}}{.00001038 \times \text{chemical equivalent of element}}.$$

The table on page 1057 (from "Practical Electrical Engineering") is culated upon Lord Rayleigh's determination of the electro-chemical equi lents and Roscoe's atomic weights.

ELECTRO-MAGNETS.

Units of Electro-magnetic Measurements.

C.G.S. unit of force = 1 dyne = 1.01936 milligrammes in localities in w the acceleration due to gravity is 981 centimetres, or 32.185 feet, per seco

C.C.S. unit of energy = 1 erg = energy required to overcome the re ance of 1 dyne at a speed of 1 centimetre per second. 1 watt = 10^7 ergs

Unit magnetism = that amount of magnetic matter which, if concentr in a point, will repel an equal amount of magnetic matter concentrate another point one centimetre distant with the force of one dyne.

Unit strength of field = that flow of magnetic lines which will exert mechanical force upon unit pole, or a density of 1 line per square ce metre.

The following definitions of practical units of the magnetic circuit given in Houston and Kennelly's "Electrical Engineering Leaflets."

Gilbert, the unit of magneto-motive force; such a M.M.F. as would produced by $\frac{10}{4\pi}$ or 0.7958 ampere-turn.

If an air-core solenoid or hollow anchor-ring were wound with 100 tu of insulated wire carrying a current of 5 amperes, the M.M.F. exerted wo be 500 ampere-turns = 628.5 gilberts.

Weber, the unit of magnetic flux; the flux due to unit M.M.F. when reluctance is one oersted.

Gauss, the unit of magnetic flux-density, or one weber per normal squ centimetre.

The flux-density of the earth's magnetic field in the neighborhood New York is about 0.6 gauss, directed downwards at an inclination of ab 72°.

Oersted, the unit of magnetic reluctance; the reluctance of a cubic cen metre of an air-pump vacuum.

Reluctance is that quantity in a magnetic circuit which limits the f under a given M.M.F. It corresponds to the resistance in the electric c cuit.

The reluctivity of any medium is its specific reluctance, and in the C.G system is the reluctance offered by a cubic centimetre of the body betw opposed parallel faces. The reluctivity of nearly all substances, other th the magnetic metals, is sensibly that of vacuum, is equal to unity, and independent of the flux density.

Permeability is the reciprocal of magnetic reluctivity.

gnetic circuit is
$\frac{\text{ilberts}}{\text{ersteds}}$;
rce ÷ magnetic reluctance.

oersteds = gilberts ÷ webers.
reasing the magnetic flux: 1. by in-
the reluctance.
In discussing magnetic and electrical
d that the attractions and repulsions
of a conductor upon iron filings are
e magnet or conductor. The "num-
of the forces acting. As the iron
ric circles, we may assume that the
ves or "loops of force." The follow-
the loops of force in a conductive

the conductor are parallel to the axis

o the conductor are proportional in
r, that is, a definite current generates
hese may be stated as the strength of

e are at right angles to the axis of

a point is equal at all points on the
ribed by a given radius about that
surface of 4π square centimetres. If
the number of lines of force emanat-
nagnetic matter,

$I = F \div 4\pi$.

roduct of strength of pole M and its
. Magnetic moment $= \frac{LF}{4\pi}$.

ugh each square centimetre of cross-
fic induction," and A = cross-section,

nt $= \frac{LAB}{4\pi}$.

magnetic field whose induction is H,
are all horizontal and at right angles
ill be pulled forward, that is, in the
the south pole will be pulled in the
ucing a torsional moment or torque,

÷ 4π, in dyne-centimetres.

nating from a point varies inversely
point. The law of inverse squares,
ism proceeds from a surface of ap-
are small, as in dynamo-electric
ation of Heat," page 467.)
net.—In an electric magnet made by
round a core of soft iron, the space
nce is called the magnetic field, and
ength of the field is proportional to
rce surrounding the magnet. Under
rent passing through a given number
agnetic loops will depend upon the
t as the current with a given press-
pon the resistance of the circuit.
ost important principles concerning

of an electro-magnet is nearly pro-
etizing current, provided the cor

(2) The magnetic strength is proportional to the number of turn in the magnetizing coil; that is, to the number of ampere turns.

(3) The magnetic strength is independent of the thickness or m the conducting wires.

These laws may be embraced in the more general statement strength of an electro-magnet, the size of the magnet being the proportional to the number of its ampere turns.

Force in the Gap between Two Poles of a Mag P = force exerted by one of the poles upon a unit pole in the gap, density of lines in the field (that is, that there are m absolute or C. on each square centimetre of the polar surface of the magnet), surface being large relative to the breadth of the gap, $P = 2\pi m$. force exerted upon the unit pole by both north and south pol magnet is $2P = 4\pi m$, in dynes $= B$, or the induction in lines of square centimetre. If S = number of square centimetres in ea surface, SB = total flow of force, or field strength $= F$; Sm = strength $= M$, spread over each of the polar surfaces. We then $4\pi M$, as before; that is, the total field is 4π times the total pole stre

Total attractive force between the two opposing poles of a magn the distance apart is small, $= \frac{SB^2}{8\pi}$, in dynes.

This formula may be used to determine the lifting-power of an magnet, thus:

A bent magnet provided with a keeper is 3 cm. square on each the induction B = 20,000 lines per square centimetre. The attract of each limb on the keeper in dynes $= \frac{9 \times 20000^2}{8 \times 3.14}$, or in kilogran both limbs, $\frac{9 \times 400 \times 10^6}{25.12 \times 981000} \times 2 = 292$ kilogrammes.

The Magnetic Circuit.—In the conductive circuit we have

$$\text{Current} = \frac{\text{electro-motive force}}{\text{resistance}} = \frac{\text{volts}}{\text{ohms}}.$$

In the magnetic circuit we have
Number of lines, or loops, of force, or magnetism

$$= \frac{\text{Current} \times \text{conductor turns}}{\text{Resistance of magnetic circuit}} = \frac{\text{Ampere turns}}{\text{Resistance of magnetic circ}}$$

Or, in the new notation, webers $= \frac{\text{gilberts}}{\text{oersteds}}$.

Let N = No. of lines of force, Rm = total magnetic resistanc ampere turns, then $N = \frac{At}{Rm}$.

The magnetic pressure due to the ampere turns $= \frac{4}{10}\pi TC =$ where T = turns and C = amperes, whence $N = \frac{.4\pi TC}{Rm} = \frac{1.257T}{Rm}$

If Rm = total magnetic resistance, and Ra, R_A, R_F the magneti ances of the air-spaces, the armature, and the field-magnets, resp

$$Rm = Ra + R_A + R_F; \text{ and } N = \frac{.4\pi TC}{Ra + R_A + R_F}.$$

Determining the Polarity of Electro-magnets.—If is wound around a magnet in a right-handed helix, the end at w current flows into the helix is the south pole. If a wire is wound ar ordinary wood screw, and the current flows around the helix in th tion from the head of the screw to the point, the head of the scre south pole. If a magnet is held so that the south pole is opposite th the observer, the wire being wound as a right-handed helix aroun current flows in a right-handed direction, with the hands of a clock

TRIC MACHINES.

electric machines, viz.:
al energy of rotation is converted into

ical energy of rotation is converted into

of a direct current is converted into

which the energy of one or more alter-
chanical energy of rotation.
duct of the potential difference and the
f the energy given off. With alternat-
nto current strength is greater than the
s the property of reacting upon itself,

c Machines as regards Man-
Electrical Dictionary.)
achine for the conversion of mechan-
means of magneto-electric induction.
he field-magnets are excited by more
than a single electric source.
ture-coils are grouped in sections con-
a collector, so as to be connected con-
t.
ure-coils, though connected to the suc-
not connected continuously in a closed

eld-magnets are excited by means of
distinct from those which furnish cur-

he field-magnet coils have no connec-
receive their current from a separate

ld-current and the external circuit are
re circuit, so that the entire armature
coils.
e armature-coils, the field, and the ex-
y increase in the resistance of the ex-
tro-motive force from the decrease in
se in the resistance of the external cir-
the electro-motive force from the in-
The use of a regulator avoids these

Compound-wound Dynamo.—There are
gnet cores, one of which is connected
the external circuit, and the other with
y excited.
ld-magnet coils are placed in a shunt
y a portion of the circuit generated
is, but all the difference of potential of
f the field-circuit.
rease in the resistance of the external
force, and a decrease in the resistance
electro-motive force. This is just the

uous balancing of the current occurs.
between the field and the external cir-
resistance of these circuits, if the resist-
es greater, a proportionately greater
gnets, and so causes the electro-motive
contrary, the resistance of the external
es through the field, and the electro-
eased.
ound-wound Dynamo.—The
ils, one of which is in se
and the other in shunt w
ound-wound machine.
Compound-wound Dyna

If P_k is taken in kilogrammes,

$$P_k = \frac{lCB}{9810000} = 10.1937lCB10^{-8} \text{ kilogram}$$

EXAMPLE.—The mean strength of field, B, of a dynam
a current of 100 amperes flows through a wire; the forc
metres of the wire $= 10.1937 \times 10 \times 100 \times 5000 \times 10^{-8} =$
In the "English" or Kapp's system of measuremen
C.G.S. lines is taken to equal one English line. Calling
English, or Kapp's, lines per square inch, and B the indu
per square centimetre, $B_E = B \div 930.04$; and taking l''
pounds, $P_p = 531Cl''B_E10^{-6}$ pounds.

Torque of an Armature.—P_p in the last formula
to move one wire of length l'', which carries a current of
the field whose induction is B_E English lines per square
through a drum-armature splits at the commutator
each half going through half of the wires or bars.
upon one of the wires under the influence of a pole-piec
number of wires under the pole-pieces, then the total fo
radius of the armature to the centre of the conductor
then the torque $= \frac{1}{2}P_ptr, = \frac{1}{2} \times 531 \times Cl''B_E \times 10^{-6}$
moment, or pounds acting at a radius of 1 foot.

EXAMPLE.—Let the length l of an armature = 20 in.,
.5 ft., number of conductors = 120, of which $t = 80$ are
of the two pole-pieces at one time, the average induct
through the armature-field $B_E = 5$ English lines per
current passing through the armature = 400 amperes;

$$\text{Torque} = \tfrac{1}{2} \times 531 \times 400 \times 20 \times 5 \times 80 \times .5 \times 1$$

The work done in one revolution = torque × circum
1 foot radius = 424.8 × 6.28 = 2670 foot-pounds.
Let the revolutions per minute = 500, then the horse-

$$= \frac{2670 \times 500}{33000} = 40.5 \text{ H.P.}$$

Electro-motive Force of the Armature C
horse-power, calculated as above, together with the am

-8 } for two-pole machines.
6

0^{-1} } for multipolar machines with series-wound armature.
-6

$1.615F\tau C10^{-10}$ } for two-pole machines.
$7.05Z\tau C10^{-6}$

$3.23F\tau cp\,10^{-10}$ } for multipolar machines.
$14.10Z\tau cp10^{-6}$

gth of armature $l = 20$ in., diameter
$= 240$ sq. in., induction per sq. in. $B_E =$
$= 240 \times 5 = 1200$; then

$120 \times 500 \times 10^{-4} =$ **72 volts.**

by Kapp is

$6\ ZNtn10^{-6}Ca$
$6\ 2abmNtn10^{-6}Ca.$

. per min., $2ab =$ sectional area of arm-
lines per sq. in. of armature-core, $Nt =$
nted all around the circumference, $t =$
ne plate in the commutator, $N =$ num-
ber of English lines of force.
own that the density of lines m in the
which is reached when the core is satu-
s reached when $m = 30$. A fair average
ors is $m = 20$, and the area ab must be
and not the gross area of the core. $2v$
G.S. lines per square centimetre. Sil-
visable in continuous-current machines
than $B = 17,000$ C.G.S. lines per square

e for the magnetic field in the gap-space
sq. cm., or 40,000 lines per sq. in., and
lb. for each ampere of current carried.
or $= \dfrac{\text{H.P.} \times 33,000}{\text{ft. per min.} \times C}$, in which C is the
rmature.
Field.—Kapp gives for the total num-
C.G.S. lines $\div$ 6000) in the magnetic cir-

$Z =$ number of magnetic lines, $X =$ the

turns $= .4\pi TC$, Ra, R_A, and R_F. = re-
spaces, the armature, and the field-mag-

l values of Ra, R_A, and R_F, for dynamos
rought iron, with a permeability of $\mu =$

$$_A = \frac{l}{ab};\quad R_F = 2\,\frac{L}{AB};$$

pan between armature-core and polar
neasured parallel to axis, $\lambda =$ length of
that $\lambda b =$ the polar area out of which
th of armature-core, so that $ab =$ sec-
y occupied by iron only being reckoned,
= length of magnetic cir
cuit in field magnet; a

For cast-iron magnets, $Z = \dfrac{0.8X}{1800\dfrac{2\delta}{\lambda b} + \dfrac{l}{ab} + \dfrac{8L}{AB}}$.

For double horse-shoe magnets of wrought iron,

$$\frac{Z}{2} = \frac{X}{1440\dfrac{2\delta}{\lambda b} + \dfrac{2l}{ab} + \dfrac{2L}{AB}};$$

and of cast iron, $\dfrac{Z}{2} = \dfrac{0.8X}{1800\dfrac{2\delta}{\lambda b} + \dfrac{2l}{ab} + \dfrac{8L}{AB}}$

These formulæ apply only to cases in which the intensity of magnet is not too great—say up to 10 Kapp's lines per square inch.

Silvanus P. Thompson gives the following method of calculati strength of the field, or the magnetic flux, MF, or the whole num magnetic lines flowing in the circuit in C.G.S. lines:

The magnetic resistance of any magnetic conductor is proportional ly to its length and inversely to its cross-section and its permeability.

Magnetic resistance $= \dfrac{L}{S\mu}$, in which $L =$ length of the magnetic passing through any piece of iron, $S =$ section of the magnetic passing through any piece of iron, $\mu =$ permeability of that piece of

In a dynamo-machine in which the resistances are three, viz.: 1. Th magnet cores; 2. The armature-core; 3. The gaps or air-spaces b them,—

let Lm, Sm, μm refer to the field-magnet part of the circuit;
Las, Sas, μas refer to the air-space part of the circuit;
La, Sa, μa refer to the armature part of the circuit;

the lengths across each of the air-spaces being Las, and the exposed polar surface at either pole being Sas.

Total magnetic resistance $= \dfrac{Lm}{Sm\mu m} + \dfrac{Las}{Sas\mu as} + \dfrac{La}{Sa\mu a}$.

Magnetic flux, or total number of magnetic lines, =

$$MF = \frac{1.257\,TwC}{\dfrac{Lm}{Sm\mu m} + \dfrac{Las}{Sas\mu as} + \dfrac{La}{Sa\mu a}}.$$

$Tw =$ turns of wires, or number of turns in the spiral;
$C =$ current in amperes passing through spiral.

Application to Designing of Dynamos. (S. P. Thom Suppose in designing a dynamo it has been decided what will be a ient speed, how many conductors shall be wound upon the armatu what quantity of magnetic lines there must be in the field, it then b necessary to calculate the sizes of the iron parts and the quantity of tion to be provided for by the field-magnet coils. It being known w is to be, the problem is to design the machine so as to get the r value. Experience shows that in every type of dynamo there is m leakage; also, that it is not wise to push the saturation of the armatu to more than 16,000 lines to the square centimetre at the most high rated part, and that the induction in the field-magnet ought to greater than this, even allowing for leakage. Leakage may amou of the whole: hence, if the magnet-cores are made of same quality as the armature-cores, their cross-section ought to be at least 5/4 as that of the armature-core at its narrowest point. If the field- are of cast iron, the section ought to be at least twice as great.

Now, Ba (the induction in the armature-core) $= Ma \div Sa$ (or magn through armature $\div$ cross-sectional area of the armature; hence is fixed at 16,000 lines per centimetre of cross-section, we at once $Ma \div Ba$. This fixes the cross-section of the armature-core. (Exa $Ma = 4{,}000{,}000$ of lines, then there must be a cross-section equa square centimetres for $\dfrac{4{,}000{,}000}{16{,}000} = 250$.)

Circuit.—The size of wires on the arma-
mperes which it must carry without risk.
urrent (in ring or drum armatures) passes
number is supposed to have been fixed be-
he quantity of copper that must be put on
tates that the core should be made so large
winding does not exceed 1/6 of the radial
tles the size of the armature-core, from
rage length of path of the magnetic lines

Area of Air-space.—Experience further
and the advantage of making the pole-
e machines) of at least 135° each, so as to
tles *Las* and *Sas*.
ores, etc.—As shown above, the minimum
and materials; *Lm* therefore remains to
agnet-cores must be long enough to allow
ls, but should not be longer. As a rule,
ly in the yoke part, that they do not add
of the circuit, then a little extra length as-
ot matter much. It now only remains to
-turns of excitation for which it will be

t to rewrite the formula of the magnetic

$$\frac{\frac{Lm}{Sm\mu m}+2\frac{Las}{Sas.\mu as}+\frac{La}{Sa.\mu a}\Big\}}{1.257};$$

assing through the field-magnet coils;
gnet wire;
ay 5/4).

$$Ma\frac{\lambda Rm+Ras+Ra}{1.257}.$$

$$7\frac{A\times Tmw}{\lambda Rm+Ras+Ra},$$

e magnetic resistance of magnets, air-
.
yet, because the values of μ in it depend on
on in the various parts. These have to be
, given below; and, indeed, it is preferable
nce more, by dividing it into its separate
y the ampere-turns requisite to force the
es through the separate parts, and then

$$\text{nagnet-cores} = \lambda\frac{Ma}{Sm}\times\frac{Lm}{\mu m}\div 1.257.$$

$$\text{ir-spaces} = \frac{Ma}{Sas}\times 2\frac{Las}{\mu as}\div 1.257.$$

$$\text{armature-core} = \frac{Ma}{Sa}\times\frac{La}{\mu a}\div 1.257.$$

he magnet-cores, and reference to the table
the corresdonding value of μm must be.
o μa. When the total number of ampere-
certained, the size and length of
le rise of temperature, and t
ther in series, or as a shunt

Permeability.—Materials differ in regard to the resistance der to the passage of lines of force; thus iron is more permeable than air. permeability of a substance is expressed by a coefficient μ, which its relation to the permeability of air, which is taken as 1. If H = of magnetic lines per square centimetre which will pass through space between the poles of a magnet, and B the number of lines that pass through a certain piece of iron in that space, then $\mu = B \div H$. permeability varies with the quality of the iron, and the degree of tion, reaching a practical limit for soft wrought iron when B = about and for cast iron when B = about 10,000 C.G.S. lines per square cent

The following values are given by Thompson as calculated from H son's experiments:

Annealed Wrought Iron.			Gray Cast Iron.		
B	H	μ	B	H	
5,000	2	2,500	4,000	5	
9,000	4	2,250	5,000	10	
10,000	5	2,000	6,000	21.5	
11,000	6.5	1,692	7,000	42	
12,000	8.5	1,412	8,000	80	
13,000	12	1,083	9,000	127	
14,000	17	823	10,000	188	
15,000	28.5	526	11,000	292	
16,000	52	308			
17,000	105	161			
18,000	200	90			
19,000	350	54			

Permissible Amperage and Permissible Depth of W ing for Magnets with Cotton-covered Wire. (Walter *El. Engineer*, Dec. 21, 1892.)—The tables on pp. 1068, 1069, abridged those of Mr. Dix, are calculated from the formula

$$C = \sqrt{\frac{12 \times W}{\frac{\omega_m f}{M} \times T \times L}},$$

where C = current;
W = emissivity in watts per square inch;
$\omega_m f$ = ohms per mil-foot ;
M = circular mils ;
T = turns per linear inch;
L = number of layers in depth.

The emissivity is taken at .4 watt per sq. in. for stationary magn rise of temperature of 35° C. (63° F.). For armatures, according to experiments, it is approximately correct to say that .9 watt per sq be dissipated for a rise of 35° C.

The insulation allowed is .007 inch on No. 0 to No. 11 B. & S.; on No. 12 to No. 24; and .0045 inch on No. 25 to No. 31 single; twi values for insulation of double-covered wires. Fifteen per cent is for imbedding of the wires.

The standard of resistance employed is 9.612 ohms per mil-foot at running temperature of tables is taken at 25° + 35° = 60° C. The giving the depth for one layer is the diameter over insulation.

Formulæ of Efficiency of Dynamos.

(S. P. Thompson in "Munro and Jamieson's Pocket-Book.")

Total Electrical Energy (per second) of any dynamo (expressed is the product of the whole E.M.F. generated by armature-coils whole current which passes through the armature.

Useful Electrical Energy (per second), or useful output of the ma the product of the useful part of the E.M.F. (i.e., that part which able ... *minals of the machine*) into the useful part of the (i.e. ... *the current which flows from the terminals into th* [illegible] Double Cotton-covered Wire.

		Diam. Bare, inches.	Circular Mils.	Ohms per foot at 60°C.	Lbs. per foot.		Turns per linear inch.	Layers.							
					Bare.	Cover'd		1		5		10		20	
								Amp.	Depth.	Amp.	Depth.	Amp.	Depth.	Amp.	Depth.
	2	.284	80656	.0001495	.244		3.36	97.8	.298	43.7	1.334	31.0	2.63	21.9	5.23
	3	.259	67081	.000180	.203		3.66	85.4	.273	38.1	1.222	27.1	2.41	19.1	4.79
2		.2576	66373	.000182	.201		3.68	84.6	.2716	37.7	1.216	26.8	2.40	18.9	4.76
	4	.238	56644	.000213	.172		3.97	75.4	.252	33.6	1.128	23.9	2.21	16.8	4.42
3		.2294	52634	.000229	.159		4.11	71.5	.2434	31.9	1.060	22.7	2.15	16.0	4.27
	5	.22	48400	.000249	.146		4.27	67.2	.234	30.0	1.016	21.3	2.07	15.0	4.10
4		.2043	41743	.000289	.126	.159	4.58	60.3	.2183	26.9	.980	19.1	1.93	13.5	3.84
	6	.203	41209	.000292	.125		4.61	59.8	.217	26.7	.974	18.9	1.92	13.4	3.80
5		.1819	33102	.000365	.100	.122	5.10	50.8	.196	22.7	.880	16.1	1.73	11.4	3.44
	7	.18	32400	.0002725	.0979		5.16	50.0	.194	22.3	.871	15.8	1.71	11.2	3.40
	8	.165	27225	.000413	.0825		5.59	41.1	.179	19.7	.804	13.9	1.58	9.85	3.14
6		.162	26251	.000459	.0791	.0942	5.68	42.9	.176	19.1	.791	13.6	1.55	9.64	3.09
	9	.148	21904	.000551	.0663		6.17	37.6	.162	16.8	.727	11.9	1.43	8.41	2.84
7		.1443	20817	.0005795	.0628	.0796	6.32	36.2	.1583	16.1	.712	11.5	1.40	8.10	2.78
	10	.134	17956	.000672	.0544		6.76	32.5	.148	14.5	.665	10.3	1.31	7.27	2.59
8		.1285	16510	.000731	.0501	.0617	7.02	30.6	.1425	13.6	.640	9.70	1.26	6.84	2.50
	11	.12	14400	.000838	.0436		7.46	27.7	.134	12.3	.602	8.78	1.18	6.20	2.35
9		.1144	13094	.000922	.0393	.0505	7.79	25.7	.1284	11.5	.577	8.14	1.13	5.75	2.25

Permissible Amperage and Permissible Depth of Winding for Magnets with Single Cotton-covered Wire.

Gauge.		Diam. Bare, inches.	Circular Mils.	Ohms per foot at 60 deg. C.	Lbs. per foot.		Turns per linear inch.	Layers.																											
								1		2		3		4		6		8		10		12		14		16		18	
B.& S.	Bir.				Bare.	Cov'd		Amp.	Depth.	Amp.	Depth.	Amp.	Depth.	Amp.	Depth.	Amp.	Depth	Amp.	Depth.	Amp.	Depth.	Amp.	Depth.	Amp.	Depth.	Amp.	Depth.	Amp.	Depth.
	2	.284	80656	.000150	.244		3.44	96.7	.291	68.3	.56	55.8	.80	48.4	1.05	39.5	1.55	34.2	2.06	30.7	2.57	27.9	3.08	25.8	3.58	24.2	4.08	22.8	4.60
	3	.259	67081	.000180	.203		3.76	84.3	.266	59.6	.50	48.6	.73	42.2	.96	34.4	1.42	29.8	1.89	26.7	2.35	24.3	2.81	22.6	3.28	21.1	3.74	19.9	4.20
2		.2576	66373	.000182	.201		3.78	83.6	.2646	59.1	.50	48.2	.73	41.8	.96	34.1	1.41	29.6	1.87	26.5	2.34	24.1	2.80	22.4	3.26	20.9	3.72	19.7	4.18
	4	.238	56644	.000213	.172		4.08	74.3	.245	52.5	.46	42.8	.67	37.1	.89	30.3	1.31	26.1	1.74	23.6	2.16	21.4	2.59	19.9	3.02	18.6	3.44	17.5	3.87
3		.2294	52634	.000229	.159		4.23	70.4	.2364	49.8	.44	40.6	.65	35.2	.86	28.8	1.26	24.9	1.67	22.3	2.09	20.3	2.50	18.8	2.91	17.6	3.32	16.6	3.73
	5	.22	48400	.000249	.146		4.41	66.2	.227	46.8	.43	38.2	.62	33.1	.82	27.0	1.21	23.4	1.61	21.0	2.00	19.1	2.40	17.7	2.80	16.6	3.19	15.6	3.58
4		.2043	41743	.000289	.126	.143	4.73	59.3	.2113	41.9	.40	34.2	.58	29.6	.76	24.2	1.13	21.0	1.50	18.8	1.86	17.1	2.23	15.8	2.60	14.8	2.97	14.0	3.33
	6	.203	41209	.000292	.125		4.76	58.8	.210	39.5	.39	33.9	.58	29.9	.76	24.0	1.12	20.8	1.49	18.6	1.85	16.9	2.22	15.7	2.59	14.7	2.95	13.9	3.32
5		.1819	33102	.000365	.100	.111	5.29	49.9	.189	35.3	.36	28.8	.52	24.9	.68	20.4	1.01	17.7	1.34	15.8	1.67	14.4	2.00	13.3	2.33	12.5	2.86	11.8	2.99
	7	.18	32400	.000273	.0979		5.35	49.1	.187	34.7	.35	28.3	.51	24.5	.67	20.0	1.00	17.3	1.33	15.5	1.65	14.1	1.98	13.1	2.30	12.27	2.63	11.6	2.95
	8	.165	27225	.000443	.0825		5.81	43.2	.172	30.5	.32	24.9	.47	21.6	.62	17.6	.92	15.3	1.22	13.7	1.52	12.5	1.82	11.5	2.12	10.80	2.42	10.2	2.72
6		.162	26251	.000459	.0794	.0869	5.92	42.1	.169	29.7	.32	24.2	.46	21.1	.61	17.2	.90	14.9	1.20	13.3	1.49	12.1	1.79	11.2	2.08	10.52	2.38	9.94	2.67
	9	.148	21904	.000551	.0663		6.45	36.8	.155	26.0	.29	21.2	.43	18.4	.56	15.0	.83	13.0	1.10	11.65	1.37	10.6	1.64	9.83	1.91	9.20	2.18	8.68	2.45
7		.1443	20817	.000580	.0628	.0714	6.61	35.4	.1513	25.0	.28	20.4	.42	17.7	.55	14.4	.81	12.5	1.07	11.21	1.33	10.2	1.60	9.46	1.86	8.85	2.13	8.35	2.39
	10	.134	17956	.000672	.0544		7.09	31.7	.141	22.4	.26	18.3	.39	15.8	.51	12.9	.75	11.2	1.00	10.04	1.24	9.15	1.49	8.48	1.74	7.92	1.98	7.48	2.23
8		.1285	16510	.000731	.0501	.0571	7.38	29.9	.1355	21.2	.25	17.2	.37	14.9	.49	12.2	.72	10.6	.96	9.48	1.20	8.63	1.43	7.99	1.67	7.47	1.90	7.06	2.14
	11	.12	14400	.000838	.0436		7.87	27.0	127	19.1	.24	15.55	.35	13.5	.46	11.0	.68	9.56	.90	8.56	1.12	7.79	1.34	7.22	1.56	6.75	1.79	6.37	2.00
9		.1144	13094	.000922	.0393	.0454	8.24	25.1	.1214	17.8	.23	14.45	.33	12.57	.44	10.2	.65	8.90	.86	7.95	1.07	7.25	1.28	6.71	1.50	6.27	1.71	5.92	1.92
	12	.109	11881	.001015	.0359		8.62	23.4	.116	16.6	.22	13.48	.32	11.70	.42	9.55	.62	8.28	.82	7.44	1.02	6.75	1.23	6.25	1.43	5.85	1.63	5.52	1.83
10		.1019	10382	.001161	.0314	.0370	9.18	21.3	.1089	15.1	.204	12.26	.30	10.63	.39	8.20	.58	7.54	.77	6.75	.96	6.15	1.15	5.70	1.34	5.32	1.53	5.03	1.72
	13	.095	9025	.001336	.0273		9.80	19.2	.102	13.56	.191	11.05	.28	9.60	.37	7.84	.55	6.80	.72	6.08	.90	5.54	1.08	5.13	1.26	4.80	1.43	4.53	1.61
11		.0907	8234	.001465	.0250	.0294	10.24	17.9	.0977	12.65	.183	10.60	.27	8.95	.35	7.31	.57	6.34	.69	5.67	.86	5.17	1.03	4.78	1.20	4.97	1.37	4.22	1.54
	14	.083	6889	.00175	.0208		11.36	15.37	.088	10.85	.165	8.86	.24	7.68	.32	6.27	.47	5.44	.62	4.87	.78	4.43	.93	4.10	1.08	3.84	1.24	3.62	1.39
12		.0808	6530	.001845	.0198	.0238	11.66	14.93	.0858	10.55	.161	8.62	.236	7.46	.31	6.10	.46	5.28	.61	4.73	.76	4.31	.91	3.99	1.06	3.73	1.21	3.52	1.35
13	15	.072	5184	.002325	.0157	.0182	12.99	12.61	.077	8.92	.144	7.28	.212	6.30	.28	5.15	.41	4.47	.55	4.00	.68	3.64	.82	3.37	.95	3.15	1.08	2.98	1.22
	16	.065	4225	.00286	.0128		14.29	10.84	.07	7.66	.131	6.26	.192	5.42	.25	4.43	.38	3.84	.50	3.44	.62	3.12	.74	2.90	.86	2.71	.98	2.56	1.10
14		.0641	4107	.002935	.0124	.0147	14.47	10.63	.0691	7.52	.129	6.14	.190	5.31	.25	4.34	.37	3.76	.49	3.37	.61	3.07	.73	2.84	.85	2.66	.97	2.51	1.09
15		.0571	3257	.00370	.00986	.0115	16.10	8.98	.0621	6.34	.116	5.18	.170	4.49	.224	3.66	.33	3.18	.44	2.85	.55	2.59	.66	2.40	.77	2.24	.87	2.12	.98
16		.0508	2583	.004665	.00783	.00909	17.92	7.58	.0558	5.36	.105	4.37	.153	3.79	.202	3.10	.30	2.68	.40	2.40	.49	2.19	.59	2.03	.69	1.89	.79	1.79	.88
17		.0453	2048	.00580	.00620	.00714	19.88	6.41	.0503	4.53	.094	3.69	.138	3.21	.181	2.62	.27	2.27	.36	2.08	.44	1.85	.53	1.71	.62	1.60	.71	1.51	.80
18		.0403	1624	.00742	.00493	.00571	23.08	5.41	.0453	3.82	.085	3.12	.124	2.71	.164	2.21	.24	1.91	.32	1.71	.40	1.56	.48	1.44	.56	1.35	.64	1.27	.72
19		.0359	1288	.00935	.00389	.00455	24.45	4.59	.0409	3.24	.077	2.65	.112	2.29	.148	1.87	.22	1.62	.29	1.46	.36	1.32	.43	1.23	.51	1.15	.58	1.08	.65
20	21	.032	1024	.0118	.00310	.00357	27.03	3.88	.037	2.74	.069	2.24	.102	1.94	.137	1.58	.198	1.37	.26	1.26	.33	1.12	.39	1.04	.46	.970	.52	.916	.58
21		.0285	810	.0149	.00246	.00278	29.85	3.28	.0335	2.32	.063	1.89	.092	1.64	.121	1.34	.179	1.16	.24	1.04	.30	.946	.35	.877	.41	.820	.47	.774	.52
22		.0253	642	.01874	.00194	.00220	33.00	2.79	.0303	1.97	.057	1.61	.082	1.39	.109	1.14	.162	.988	.22	.885	.27	.805	.32	.746	.37	.697	.43	.658	.48
23		.0226	509	.02365	.00154	.00175	36.23	2.37	.0276	1.67	.052	1.365	.076	1.184	.100	.968	.148	.839	.20	.752	.24	.684	.29	.633	.34	.592	.39	.559	.44
	24	[illegible]	[illegible]	[illegible]	.00146		37.04	2.29	.027	1.616	.051	1.330	.074	1.142	.098	.935	.144	.810	.19	.720	.24	.661	.29	.619	.33	.572	.38	.540	.43

Gauge. B. & S.	Gauge. Bir.	Diam. Bare, inches.	Circular Mils.	Ohms per foot at 60°C.	Lbs. per foot. Bare.	Lbs. per foot. Cover'd	Turns per linear inch.	1 Amp.	1 Depth.	Amp.	Depth.	Amp.	Depth.		
	2	.284	80656	.0001495	.244		3.36	97.8	.298	43.7	1.334	31.0	2.63	21.9	5.23
	3	.259	67081	.000180	.203		3.66	85.4	.273	38.1	1.222	27.1	2.41	19.1	4.79
2		.2576	66373	.000182	.201		3.68	84.6	.2716	37.7	1.216	26.8	2.40	18.9	4.76
	4	.238	56644	.000213	.172		3.97	75.4	.252	33.6	1.128	23.9	2.21	16.8	4.42
3		.2294	52634	.000229	.159		4.11	71.5	.2434	31.9	1.090	22.7	2.15	16.0	4.27
	5	.22	48400	.000249	.146		4.27	67.2	.234	30.0	1.046	21.3	2.07	15.0	4.10
4		.2043	41743	.000289	.126	.159	4.58	60.3	.2183	26.9	.980	19.1	1.93	13.5	3.84
	6	.208	41209	.000292	.125		4.61	59.8	.217	26.7	.974	18.9	1.92	13.4	3.80
5		.1819	33102	.000365	.100	.122	5.10	50.8	.196	22.7	.880	16.1	1.73	11.4	3.44
	7	.18	32400	.0002725	.0979		5.16	50.0	.194	22.3	.871	15.8	1.71	11.2	3.40
	8	.165	27225	.000443	.0825		5.59	44.1	.179	19.7	.804	13.9	1.58	9.85	3.14
6		.162	26251	.000459	.0794	.0942	5.68	42.9	.176	19.1	.791	13.6	1.55	9.61	3.09
	9	.148	21904	.000551	.0663		6.17	37.6	.162	16.8	.727	11.9	1.43	8.41	2.84
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		.1285	16510	.000731	.0501	.0617	7.02	30.6	.1425	13.6	.640	9.70	1.26	6.84	2.50
	11	.12	14400	.000838	.0436		7.46	27.7	.134	12.3	.602	8.78	1.18	6.20	2.35
		.1144	13094	.000922	.0393	.0505	7.79	25.7	.1284	11.5	.577	8.14	1.13	5.75	2.25

Alternating Currents, Multiphase Currents, Transformers, etc.—The proper discussion of these subjects would take more space than can be afforded in this work. Consult S. P. Thompson's "Dynamo-Electric Machinery," Bedell and Crehore on "Alternating Currents," Fleming on "Alternating Currents," and Kapp on "Dynamos, Alternators and Transformers."

The Electric Motor.—The electric motor is the same machine as the dynamo, but with the nature of its operation reversed. In the dynamo mechanical energy, such as from a belt, is converted into electric current; in the motor the current entering the machine is converted into mechanical energy, which may be taken off by a belt. The difference in the action of the machine as a dynamo and as a motor is thus explained by Prof. F. B. Crocker, (*Cassier's Mag.*, March, 1895):

In the case of the dynamo there exists only one E.M.F., whereas in the motor there must always be two.

One kilowatt dynamo, $C = E \div R$; 10 amperes = 100 volts ÷ 10 ohms.

One kilowatt motor, $C = \dfrac{E - e}{R_1}$; 10 amperes $= \dfrac{\text{100 volts} - \text{90 volts}}{\text{1 ohm}}$.

C is the current; E, the direct E.M.F.; e, the counter E.M.F.; R, the total resistance of the circuit; R_1, the resistance of the armature. The current and direct E.M.F. are the same in the two cases, but the resistance is only one tenth as much in the case of the motor, the difference being replaced by the counter E.M.F., which acts like resistance to reduce the current. In the case of the motor the counter E.M.F. represents the amount of the electrical energy converted into mechanical energy. The so-called electrical efficiency or conversion factor = counter E.M.F. ÷ direct E.M.F. The actual or commercial efficiency is somewhat less than this, owing to friction, Foucault currents, and hysteresis.

For full discussions of the theory and practice of electric motors see S. P. Thompson's "Dynamo-Electric Machinery," Kapp's "Electric Transmission of Energy," Martin and Wetzler's "The Electric Motor and its Applications," Cox's "Continuous Current Dynamos and Motors," and Crocker and Wheeler's "Practical Management of Dynamos and Motors."

ES QUOTED IN THIS BOOK.

ce or a few times only, the page or pages writers of text-books, who are quoted fre-s" affixed in place of the page-number. both as to names and page numbers.

INDEX.

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CLASSIFIED INDEX TO ADVERTISEMENTS.

The C. W. HUNT COMPANY,

Business Established in 1872. • **45 BROADWAY, NEW YORK,**

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"STEVEDORE" ROPE. TRADEMARK.

This rope is intended to be used for **Rope Driving and Hoisting only**, and is sold under the trade name of **"Stevedore."** We guarantee that **more work** can be done with it, in proportion to its cost, than with any other rope in the market, without any exception whatever, and will gladly refund the difference in price, if it is not all we claim. **You cannot lose** by making a trial of this rope, and are sure to reduce your expense account. This guarantee goes with every rope we make. We do not manufacture transmission pulleys, shafting, or hangers, neither do we make ordinary rope.

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This system of cars and tracks, 21½ inches gauge, is especially designed for use in manufacturing establishments. Our narrow gauge railroad track is made up complete, ready to lay, with steel cross ties securely riveted to the rails, and, with switches, curves, crossings, and turntables, makes a perfect permanent way. The cars run around a curve of 12 feet radius as easily as a wagon turns a corner. The advantage of this will be appreciated by every one who has had experience with rigid wheel-base cars.

A system of cars and track in a factory is as much a "machine" as a "lathe," a "steam hammer," or a "loom," and should be judged in the same way.

COAL HANDLING MACHINERY.

For hoisting Coal, Ore, and Stone from vessels, of all varieties, from the simple and cheap Mast and Gaff hoisting from Canal Boats, to the most elaborate plant that will unload a 3,000 ton vessel in ten hours; Hoisting Engines, Automatic and Cable Railways for handling and distributing Coal and Bulk Materials in Coal Yards, on Wharves, and at Manufacturing Establishments, Coal Tubs, Steam Shovels, Mast and Gaff Fittings, Wheel Barrows, and Hoisting Blocks for both Manila and Wire Rope.

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Noiseless Conveyors, which carry the aterial without shock, breakage, or violence any direction, in which every bearing can be pt thoroughly lubricated, and the whole chine is as durable as an ordinary machine tool. We print a alogue describing the application of this machine to large eam Generating Plants, Locomotive Coaling Stations, Gas orks, and Coal Yards.

ADVERTISEMENTS.

5 BROADWAY,

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This rope is intended to be used r **Rope Driving and Hoisting** nly, and is sold under the trade ame of "**Stevedore.**" We guarntee that **more work** can be done with it, in proportion to its cost, than with any other rope in the market, without any exception whatever, and will gladly refund the difference in price, if it is not all we claim. **You cannot lose** by making a trial of this rope, and are sure to reduce your expense account. This guarantee goes with every rope we make. We do not manufacture transmission pulleys, shafting, or ke ordinary rope.

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A system of cars and track in a factory is as much a "machine" as a "lathe," a "steam hammer," or a "loom," and should be judged in the same way.

tables, anent

G MACHINERY.

, and Stone from vessels, of all varieties, eap Mast and Gaff hoisting from Canal orate plant that will unload a 3,000 ton loisting Engines, Automatic and Cable nd distributing Coal and Bulk Materials. es, and at Manufactur- l Tubs, Steam Shovels, Wheel Barrows, and Manila and Wire Rope.

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, which carry the breakage, or violence h every bearing can be cated, and the whole s an ordinary machine tool. We print a e application of this machine to large ts, Locomotive Coaling Stations, Gas

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Foundry and Machine Work.

THE TAUNTON LOCOMOTIVE MANUFACTURING CO.,

Taunton, Mass.